Verification and validation manual

Axomek 0.3.3

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2371 of 2371 cases within their tolerance, in 117 groups over 21 chapters.

Scope and use

What this manual covers

Every calculation module of Axomek 0.3.3, one chapter each: the methods it implements, with their standards' numbers and clauses; the cases it is validated against, each with its reference, Axomek's value, the deviation and the tolerance; and the methods' stated limits.

How it is made

Every number in the validation tables was written by the tests themselves when they compared it, in this version's test run. Continuous integration fails when this manual and the code disagree: a case missing or new, a value, tolerance or reference that differs, or a case outside its tolerance. A release is built only from a commit that passed it.

How to read a case

- A case compares Axomek's value with a reference: a worked example printed in a standard or book, a standard's table, the standard's own formulae worked by hand (which shows they are implemented as printed, not that they hold), an exact closed-form solution, a calculation of the same case by another program or method, or a consistency check of a module against its own parts (which shows no more than that they agree). Its tolerance is the one the test enforces: usually the reference's printed rounding, wider where the reference is itself approximate, as the group's note says. A relative tolerance and deviation are given in % (below 0.001 % as a power of ten), an absolute one in the value's unit. Values are kept to eight significant digits and shown to seven, so a deviation below about 10^{-7} of the value shows as 0.
- A passing case shows that the method is implemented as published for that case, within the tolerance. It does not show that the method fits a particular design, nor that cases far from those checked behave the same. The methods' limits are stated with each chapter.
- Standards and books are cited by number and clause, table or page only. Reference values from other programs or manufacturers are cited without their names.

Methods not validated against a published reference

Their cases, where there are any, are the standards' formulae worked by hand, closed forms of mechanics or consistency checks.

- Shafts: deflection, critical speeds and thermal growth: Unbalance response, and the torsional and axial modes of a system of shafts.
- Rolling bearings: The four-component friction model, and Palmgren's idle factor f0: DIN 732's for an oil bath, where the AGMA ISO 14179-1 example compared takes a larger f0 by how deep the bearing dips (the load term M1 and the rest of M0 are compared with that example); The elastic outer ring and the high-speed ball model.
- Lubrication and friction: Film thickness and the viscosity ratio κ; The four-component bearing friction and oil drag.
- Limits, fits and press fits: DIN 7190 transmission capacity and slip, the stresses in hub and shaft against their permissible joint pressure, the press-in force, the joining temperature and turning joints: no worked example is compared (the cases are Lamé's closed forms).

- Cylindrical gears: ISO 6336: Loaded tooth contact analysis and the tooth root finite elements (compared only with ISO 6336 itself in the tests); The mesh loss's coefficient of friction μmz: compared only with the AGMA ISO 14179-1 example's, which another formula gives (the engine's is 29 % higher at Rz 3 μm); Gearbox efficiency over its stages beyond the mesh loss's path of contact, and thermal rating.
- Bevel gears: ISO 10300: No worked example of ISO 10300 is compared: the cases are an independent evaluation of its formulae; Tooth root stress (rated by ISO 6336-3 on the virtual pair; ISO 10300-3 is not implemented).
- Planetary gear sets: No worked planetary rating: the cases are the load sharing factor's source tables and the stage's statics; The planet carrier's finite elements.
- Racks and rack meshes: The rack and pinion mesh rating and the rack duty: no worked example is compared; Rack member strength at the teeth.
- Keys and splines: DIN 6892's K_λe: compared with Annex D.2's reading only; D.1 and D.3 print 1.07 where the figures give 1.087 and 1.097, so their pressures are compared at the printed value; DIN 6892's friction factor K_R, load direction change factor f_W and peak load factor f_L, which the engine does not have; Spline load capacity (flank pressure and shear), and the splines' estimated K_λ.
- Helical springs: EN 13906: The stainless wire's solid length: no worked example of stainless wire is compared (its d_max takes EN 10270-3 Table 5's class T12); Permissible stresses, buckling, the growth of the outside diameter, the initial tension that can be coiled and extension spring hooks: no worked example is compared; Fatigue (a Goodman-type line in place of the standard's diagrams).
- Welds: EN 1993-1-8 and EN 1993-1-9: Butt and partial-penetration butt welds, and weld groups in torsion: no worked example is compared (the cases are the formulae worked by hand); Fatigue damage of a weld group: only the fatigue strength curves are compared, and not those of details below category 71, which EN 1993-1-9:2025 changed.
- Linear guides: ISO 14728: No worked example of ISO 14728 is compared: the cases are the statics of a rigid table and the life formula.
- Belt and chain drives: No standard's or maker's rating of belts and chains: only the mechanics of the strands are validated.
- Housings: Built-in housing walls and the planet carrier's finite elements: compared in the tests with plate theory, mesh refinement and imported meshes, not with an external benchmark.
- Steering systems: No published worked steering system: only the closed forms of the column, the assist balance and the ball nut are validated; The steering duty and the load cases it derives.

Your responsibility

You remain responsible for your design and for checking that the methods you use, their assumptions and their stated limits fit it. This manual shows how Axomek's methods compare with their references; it does not approve a design.

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Summary by chapter

2371 of 2371 cases lie within their tolerance. “Largest share used” is the case nearest its tolerance: 50 % means half the allowed deviation.

Chapter	Groups	Cases within tolerance	Largest share used
Shafts: deflection, critical speeds and thermal growth	9	61 / 61	39 %
Shaft strength: DIN 743 and the FKM guideline	4	154 / 154	92 %
Rolling bearings	21	513 / 513	97 %
Hertz contact and subsurface stress	3	44 / 44	46 %
Lubrication and friction	1	4 / 4	85 %
Limits, fits and press fits	7	177 / 177	86 %
Cylindrical gears: ISO 6336	28	593 / 593	97 %
Bevel gears: ISO 10300	1	28 / 28	4 %
Worm gears: DIN 3996, ISO/TR 14521 and VDI 2736-3	8	314 / 314	100 %
Plastic cylindrical gears: VDI 2736-2	4	61 / 61	88 %
Planetary gear sets	2	14 / 14	0 %
Racks and rack meshes	1	8 / 8	0 %
Bolted joints: VDI 2230	3	143 / 143	99 %
Keys and splines	5	82 / 82	83 %
Helical springs: EN 13906	5	59 / 59	95 %
Welds: EN 1993-1-8 and EN 1993-1-9	6	61 / 61	92 %
Ball screws: ISO 3408	3	29 / 29	43 %
Linear guides: ISO 14728	1	6 / 6	< 1 %
Belt and chain drives	1	6 / 6	24 %
Housings	2	8 / 8	50 %
Steering systems	2	6 / 6	< 1 %

Shafts: deflection, critical speeds and thermal growth

The shaft as a beam of finite elements on its bearings: deflection, slope, support loads, critical speeds, unbalance response and the axial growth of a located shaft.

Methods and standards

Shaft: finite elements with Timoshenko beam elements (optional shear deformation), supports as linear springs; stresses from bending, axial force and torsion combined by von Mises.

Shear deformation (a setting of the shaft, on by default) takes Cowper's shear coefficient of a hollow circular section, $k = 6(1 + \nu)(1 + m^2)^2 / ((7 + 6\nu)(1 + m^2)^2 + (20 + 12\nu)m^2)$ with $m = d_i/do$ ($k = 6(1 + \nu)/(7 + 6\nu)$, 0.886 for steel, when solid).

Supports at one node share its reaction among those rigid in that freedom (a spring there carries its own $k \cdot u$, which is zero at a rigid freedom).

A support carrying a self-aligning bearing on its own (self-aligning ball, spherical roller) is free to tilt: a tilt stiffness given to it is ignored, with a warning.

A part with a length spreads its mass over it (checked against the shaft's ends; a diametral inertia below the spread's $m \cdot L^2/12$ is raised to it, with a warning).

Stress concentrations at shoulders and grooves are not included here (the strength check takes them).

The largest deflection, bending moment and stress are found inside the elements too (the deflection exactly as the element's cubic), and the charts mark them where the results give them.

Shaft–bearing coupling (optional): each support linked to a bearing becomes a spring with the bearing's secant stiffness (in series with the support's own stiffness as the housing) plus a force and moment offset; the offsets are solved by Newton's method with the bearings' tangent compliance from finite differences so that each bearing's displacement and tilt under its support load equal the shaft's deflection and slope at the support in both planes.

Bearing moments follow from the support moments.

The reported coupled stiffness is the bearing's own secant: its load over its own displacement and tilt (the shaft's motion less the housing's compliance under the load and the seat's ground motion from the housing's other seats or imposed loads).

The vibration analyses use the tangent stiffness at the coupled state instead: per bending plane, radially and in tilt, the inverse of the diagonal of the bearing-and-housing compliance $\partial u/\partial F$ by forward differences at the coupled loads.

Critical speeds: lowest bending natural frequencies of the shaft in both bending planes (Timoshenko elements with the Timoshenko-consistent mass matrix, translational and rotary parts with the shear parameter $\varphi = 12EI/(\kappa GAl^2)$ after Przemieniecki and Nelson; the parts at the loads as point masses with their diametral inertia, by default a thin disc's $Ip/2$, a part with a length spread over it at three Gauss points; supports rigid or with their stiffness, coupled bearings with their tangent stiffness at the coupled state), non-rotating; the speed dependence of bearing stiffness is not included.

Mode shapes by inverse iteration at each critical speed.

Unbalance response: $(K + 2i\zeta \cdot K_d - \omega^2 M) \cdot u = U \cdot \omega^2$ per bending plane, structural damping ζ on the shaft, supports and housings (K_d; the negative magnetic springs undamped), the unbalances turning with the shaft, solved directly at each speed; the orbit radius is the larger plane's amplitude.

Gyroscopic whirl: for whirl at ω with the shaft at Ω = s·ω, the rotary inertia becomes Id − s·Ip (shaft: ρI − s·2ρI in the element's rotary matrix, which is also the gyroscopic matrix's form; parts: Id − s·Ip), exact for round supports; the generalized eigenproblem is solved through K's Cholesky factor (M may then be indefinite), s = ±1 giving the forward and backward synchronous critical speeds and a sweep of s the Campbell diagram; the unbalance response uses s = 1.

On supports that differ between the planes both planes are solved together, coupled by the gyroscopic matrix G (the polar part, in the rotary inertia's form): (K − ω²M + i·s·ω²·[[0, G], [−G, 0]])·q = 0, a Hermitian problem K·q = ω²·(M − i·s·C)·q, solved in the lowest 24 non-rotating modes of each plane (K-Cholesky, as its real symmetric double); a whirl is forward when its forward circular part (Y + iZ)/2 outweighs its backward one (mass-weighted), and s = 1 gives every synchronous crossing, forward and elliptic backward, at once.

The minimum critical speed check uses the first synchronous whirl critical speed that unbalance excites when any part has polar inertia (the rotor meets that one as it runs up): the first forward one on round supports, on supports that differ between the planes any whirl with a forward share of at least 5 %, elliptic backward ones included; else the first non-rotating bending critical speed; its message names which.

Torsional and axial modes: the shaft as a rod of linear elements (G·Ip/l with consistent polar inertia ρ·Ip, or E·A/l with ρ·A), the parts' polar inertia or mass at their nodes; torsion free at both ends; axial held at the locating support by the linked bearing's axial tangent stiffness at its load (rigid rings), or rigidly, with a note, when no bearing gives one.

Axial thermal growth (optional): free growth of the shaft against the housing (α_s·ΔT_s − α_h·ΔT_h)·L from the assembly temperature; a floating bearing's axial travel is the shaft's displacement against the housing from the locating bearing (with its axial spring and the shaft's stretch N/(E·A)).

Two or more locating bearings (a cross-located X or O pair) make the axial problem statically indeterminate: the shaft is a bar on the bearings' axial springs (their given stiffness, else the linked bearing's axial tangent stiffness at its load, iterated), so the growth changes the pair's axial force by ΔL/(1/k_1 + 1/k_2 + L/(E·A)) for two bearings, adding to the preload face to face and taking it off back to back.

A bearing that holds the shaft one way only (each bearing of the X or O pair, a single angular contact or tapered roller bearing, a thrust bearing, an NJ or NF) is a one-sided contact: it pushes but never pulls, so where the linear spring would pull it lifts off, carries no axial load and an axial gap opens, the contact set found by active-set iteration (least-index pivoting, every set tried should it not settle); bearings that hold both ways (deep groove, four-point, double-row, spherical, NUP, matched pairs) keep the linear spring.

A mounted preload P of the X or O pair is the bearings' force as mounted, the shaft's stretch between them included (a prestress of the bar), so the pair keeps P plus or minus the thermal force until the growth takes it all and then opens a play of (F − P)·(1/k_1 + 1/k_2 + L/(E·A)).

Shaft weight (optional): gravity in the shaft frame turns the shaft's own weight ρ·A·g into a distributed load (consistent nodal loads q·l/2, ±q·l²/12 per element, exact at the nodes; internal forces less the elements' fixed-end forces) and the parts' masses into point loads.

Stiffness at a load: its radial force over the shaft deflection there, with all loads acting (a spindle's nose stiffness when the cutting force is the only radial load).

Electric machine rotors: unbalanced magnetic pull as a negative radial spring k = π·D·L·B²/(2μ₀·g) (RMS flux density B, uniform gap, times a reduction factor for saturation and parallel paths), spread over the stack length at the three Gauss–Legendre points (weights 5/18, 4/9, 5/18 at 0 and ±0.387·L) in the shaft and critical speed models, so that a tilting stack meets the distributed pull's k·L²/12; the rotor's mass and inertia (a solid cylinder: polar m·D²/8, diametral m·(3r² + L²)/12) are spread at the same points.

The static eccentricity adds the pull k·e.

Rotor deflection is checked against a share of the air gap.

Magnetic stability: the largest ν with K_s − ν·K_m singular (K_s the shaft and its supports as in the critical speeds, K_m the pull's springs), from the flexibility at the springs' nodes; at ν ≥ 1 a mode diverges (a negative eigenvalue) and the check fails with a warning.

Validation

61 of 61 cases within their tolerance, in 9 groups.

Closed-form solutions

Shaft: axial thermal growth and one-sided bearing contact

Kind of reference	Closed-form solutions
Implements	Axial bar with springs
Reference source	Closed-form solutions
Cases	7 of 7 within tolerance

Ø40 × 500 mm steel shaft (E = 210 000 MPa, α = 12·10⁻⁶ 1/K) on two locating bearings 400 mm apart with axial springs k_1 = 200 and k_2 = 300 N/µm, the housing at the 20 °C assembly temperature, the shaft warmer: free growth ΔL = α·ΔT·L. Two-way springs press in series: F = ΔL/(1/k_1 + 1/k_2 + L/(E·A)). In an X or O pair each bearing pushes the shaft one way only and lifts off rather than pull: closed forms of the bar with the lifted bearing removed (statics), and of the pair once the growth has taken its mounted preload P.

Case	Reference	Axomek	Deviation	Tolerance	
Two-way springs, ΔT = 40 K: thermal force ΔL/(1/k_1 + 1/k_2 + L/(E·A))	19494.18 N	19494.18 N	6.0×10^{-15}	10^{-9}	**pass**
X pair under Fa = 80 kN: the left bearing lifts off, its axial load	0 N	0 N	0 N	1×10^{-6} N	**pass**
X pair under Fa = 80 kN: the left gap Fa·(1/k_2 + 200/(E·A)) − ΔL	0.1352971 mm	0.1352971 mm	1.2×10^{-14}	10^{-9}	**pass**
X pair under Fa = 80 kN: the right bearing takes Fa	80000 N	80000 N	0 %	10^{-9}	**pass**
O pair, P = 2 kN, ΔT = 40 K: the play (F − P)·(1/k_1 + 1/k_2 + L/(E·A))	0.1723018 mm	0.1723018 mm	1.1×10^{-14}	10^{-9}	**pass**
O pair, P = 2 kN, ΔT = 40 K: preload in operation	0 N	0 N	0 N	1×10^{-6} N	**pass**
O pair, P = 2 kN, ΔT = 2 K: bearing load P − F	1025.291 N	1025.291 N	5.8×10^{-15}	10^{-9}	**pass**

Shaft: beam theory

Kind of reference	Closed-form solutions
Implements	Euler–Bernoulli and Rayleigh beam theory
Reference source	Closed-form deflections, moments and critical speeds of uniform beams
Cases	9 of 9 within tolerance

Euler–Bernoulli closed forms (shear deformation off, 1 mm elements) and Rayleigh-beam critical speeds: steel E = 210 000 MPa, ρ = 7850 kg/m³, Ø40 mm solid shaft.

Case	Reference	Axomek	Deviation	Tolerance	
Simply supported, central load: max deflection $F \cdot L^3/(48 \cdot E \cdot I)$	0.2526269 mm	0.2526269 mm	2.1×10^{-8}	0.1 %	**pass**
Simply supported, central load: max moment $F \cdot L/4$	500000 N·mm	500000 N·mm	2.5×10^{-8}	10^{-6}	**pass**
Simply supported, central load: max bending stress M/W	79.57747 MPa	79.57747 MPa	2.5×10^{-8}	10^{-6}	**pass**
Simply supported, central load: support reaction F/2	2500 N	2500 N	2.9×10^{-8}	10^{-6}	**pass**
Simply supported, first bending critical speed (Rayleigh beam)	30373.2 rpm	30373.21 rpm	2.6×10^{-8}	0.2 %	**pass**
Simply supported, second bending critical speed (Rayleigh beam)	120390.7 rpm	120390.8 rpm	4.2×10^{-7}	0.2 %	**pass**
Overhung load: tip deflection $F \cdot a^2 \cdot (L + a)/(3 \cdot E \cdot I)$	0.05052538 mm	0.05052538 mm	5.2×10^{-9}	0.1 %	**pass**
Overhung load: reaction at the near support $F \cdot (L + a)/L$	1333.333 N	1333.333 N	2.0×10^{-8}	10^{-6}	**pass**
Torsion: von Mises stress $\sqrt{3} \cdot 16 \cdot T/(\pi \cdot d^3)$	13.78322 MPa	13.78322 MPa	0 %	10^{-6}	**pass**

Shaft–bearing coupling: beam theory

Kind of reference	Closed-form solutions
Implements	Euler–Bernoulli beam on elastic supports
Reference source	Closed-form solutions
Cases	9 of 9 within tolerance

Ø40 mm shaft, bearings 6208 at x = 20 and 380 mm (span 360 mm), central load, shear deformation off, coupling on. Reference: Euler–Bernoulli mid-span deflection of the span with end moments plus each bearing's displacement from the bearing solver (itself checked against the reference bearing solver) under the coupled support load.

Case	Reference	Axomek	Deviation	Tolerance	
F = 2 kN: support load F/2	1000 N	1000 N	3.8×10^{-10}	10^{-6}	**pass**
F = 2 kN: shaft deflection at the support = bearing displacement	0.02180754 mm	0.02180754 mm	1.7×10^{-10}	0.001 %	**pass**
F = 2 kN: mid-span deflection	0.09546031 mm	0.09546031 mm	7.4×10^{-10}	0.1 %	**pass**
F = 5 kN: support load F/2	2500 N	2500 N	2.3×10^{-9}	10^{-6}	**pass**
F = 5 kN: shaft deflection at the support = bearing displacement	0.03260009 mm	0.03260009 mm	3.7×10^{-10}	0.001 %	**pass**
F = 5 kN: mid-span deflection	0.2166388 mm	0.2166388 mm	6.7×10^{-10}	0.1 %	**pass**

Case	Reference	Axomek	Deviation	Tolerance	
F = 15 kN: support load F/2	7500 N	7500 N	5.9×10^{-9}	10^{-6}	**pass**
F = 15 kN: shaft deflection at the support = bearing displacement	0.05769825 mm	0.05769825 mm	7.0×10^{-11}	0.001 %	**pass**
F = 15 kN: mid-span deflection	0.6088153 mm	0.6088153 mm	3.8×10^{-9}	0.1 %	**pass**

Critical speeds of short and slender shafts against the exact Timoshenko beam

Kind of reference: Closed-form solutions
Implements: —
Reference source: The exact Timoshenko frequencies of a pinned uniform beam (shear and rotary inertia, Cowper's κ): Ø100 mm steel shaft, L/D 3 and 15
Cases: 4 of 4 within tolerance

Tolerance 0.1 % for L/D 3 and 0.03 % for L/D 15: the discretisation error of the finite elements.

Case	Reference	Axomek	Deviation	Tolerance	
Ø100 × 300 mm on pins: mode 1	12639.26 rad/s	12639.76 rad/s	0.004 %	0.1 %	**pass**
Ø100 × 300 mm on pins: mode 2	40449.82 rad/s	40465.85 rad/s	0.04 %	0.1 %	**pass**
Ø100 × 1500 mm on pins: mode 1	564.167 rad/s	564.1682 rad/s	2.1×10^{-6}	0.03 %	**pass**
Ø100 × 1500 mm on pins: mode 2	2221.736 rad/s	2221.807 rad/s	0.0032 %	0.03 %	**pass**

Motor bearings' minimum load where the rotor's pull leaves them least loaded

Kind of reference: Closed-form solutions
Implements: —
Reference source: Closed form (statics): a rigid shaft on two rigid supports, a load W and the rotor's pull at mid-span: each support carries (W ± t·F_p)/2 at the share t of the static eccentricity, least at t = 0 along the load and at t = W/F_p against it
Cases: 8 of 8 within tolerance

A rigid, weightless shaft (E 1000 × steel's) between a 6312 and a 6212 bearing 380 mm apart, a 1 000 N load and a rotor (Ø170 × 170 mm, 0.6 mm gap, 0.65 T, 10 % static eccentricity) at mid-span, bearing coupling off. The minimum load check takes the bearing's radial load where it is least between the rotor centred and at its eccentricity.

Case	Reference	Axomek	Deviation	Tolerance	
near the bound, BR1	0 N	0 N	0 N	0.001526281 N	**pass**
near the bound, BR2	0 N	0 N	0 N	0.001526281 N	**pass**
apart, BR1	0 N	0 N	0 N	0.001526281 N	**pass**
apart, BR2	0 N	0 N	0 N	0.001526281 N	**pass**
pull against the load, BR1	0 N	0 N	0 N	0.001 N	**pass**
pull against the load, BR2	0 N	0 N	0 N	0.001 N	**pass**
pull along the load, BR1	500 N	500.044 N	0.0088 %	0.1 %	**pass**
pull along the load, BR2	500 N	500.044 N	0.0088 %	0.1 %	**pass**

The shaft's slope at a load

Kind of reference	Closed-form solutions
Implements	—
Reference source	Closed form: the elastic curve of a simply supported uniform beam under a point load F at a (b = L − a), whose slope there is θ = F·b·(L² − b² − 3a²)/(6·E·I·L); Ø40 × 400 mm steel, F 5000 N at 100 mm
Cases	1 of 1 within tolerance

A result to read (a hub's or a gear's tilt): no limit checks it in 0.3.0. The slope is the section's rotation, both planes combined; on simple supports the shear deformation adds none to it.

Case	Reference	Axomek	Deviation	Tolerance	
Ø40 × 400 mm, F 5000 N at 100 mm on simple supports: slope at the load	3.256752 arcmin	3.256752 arcmin	1.2×10^{-11}	10^{-6}	**pass**

Speeds along a train of shafts and the bearing lives at them

Kind of reference	Closed-form solutions
Implements	—
Reference source	Closed forms: a belt's speed ratio d_1/d_2 and the basic rating life L10 = (C/P)^p·10⁶/(60·n) at unchanged loads, which goes as 1/n
Cases	6 of 6 within tolerance

The speed typed on an item (a drive's driver, a rotor) is its train's: the shafts and bearings turn at it.

Case	Reference	Axomek	Deviation	Tolerance	
belt drive Ø125 → Ø250: fan shaft speed for a driver speed of 1460 rpm	730 rpm	730 rpm	0 %	10^{-12}	**pass**
belt drive BR3: L10 at 730 rpm, from its L10 at 900 rpm	299362.3 h	299362.3 h	0 %	10^{-9}	**pass**
belt drive BR4: L10 at 730 rpm, from its L10 at 900 rpm	4.1016×10^{7} h	4.1016×10^{7} h	0 %	10^{-9}	**pass**
belt drive Ø140 → Ø224, driver speed typed 1460 rpm before the pulleys changed: fan shaft speed	912.5 rpm	912.5 rpm	0 %	10^{-12}	**pass**
electric motor BR1: L10 at 3000 rpm, from its L10 at 1460 rpm	2.2318×10^{6} h	2.2318×10^{6} h	2.1×10^{-16}	10^{-9}	**pass**
electric motor BR2: L10 at 3000 rpm, from its L10 at 1460 rpm	642125.6 h	642125.6 h	3.6×10^{-16}	10^{-9}	**pass**

The same cases calculated by other programs or methods

Shaft: critical speeds against an open-source rotordynamics code

Kind of reference	The same cases calculated by other programs or methods
Implements	Timoshenko beam finite elements
Reference source	An independent open-source rotordynamics program (version 3.0) on the same models
Cases	15 of 15 within tolerance

Bending natural frequencies at 0 rpm against an independent open-source rotordynamics program (version 3.0) on the same models: Timoshenko elements of 5 mm with shear and rotary inertia, steel $E = 210\,000$ MPa, $\nu = 0.3$, $\rho = 7850$ kg/m³, linear bearings (rigid as 10^{14} N/m), parts as disks with negligible inertia. The first mode agrees within 10^{-3}; higher modes within 0.5 % (mesh and shear coefficient).

Case	Reference	Axomek	Deviation	Tolerance	
Uniform Ø40 × 800 mm, rigid supports: mode 1	7593.7 rpm	7593.724 rpm	3.2×10^{-6}	0.1 %	**pass**
Uniform Ø40 × 800 mm, rigid supports: mode 2	30105.4 rpm	30105.89 rpm	0.0016 %	0.5 %	**pass**
Uniform Ø40 × 800 mm, rigid supports: mode 3	66769.7 rpm	66775.34 rpm	0.0084 %	0.5 %	**pass**
Uniform Ø40 × 800 mm, supports 20 N/µm: mode 1	7229.8 rpm	7229.787 rpm	1.8×10^{-6}	0.1 %	**pass**
Uniform Ø40 × 800 mm, supports 20 N/µm: mode 2	24737.4 rpm	24737.69 rpm	0.0012 %	0.5 %	**pass**
Uniform Ø40 × 800 mm, supports 20 N/µm: mode 3	44125.6 rpm	44127.21 rpm	0.0037 %	0.5 %	**pass**
Ø30 × 600 mm with a 20 kg disk at mid-span: mode 1	2790 rpm	2789.976 rpm	8.6×10^{-6}	0.1 %	**pass**
Ø30 × 600 mm with a 20 kg disk at mid-span: mode 2	40140.5 rpm	40141.18 rpm	0.0017 %	0.5 %	**pass**
Ø30 × 600 mm with a 20 kg disk at mid-span: mode 3	63013.9 rpm	63016.38 rpm	0.0039 %	0.5 %	**pass**
Stepped motor shaft, overhung 2.5 kg pulley, supports 100 N/µm: mode 1	36319.2 rpm	36319.2 rpm	1.2×10^{-7}	0.1 %	**pass**
Stepped motor shaft, overhung 2.5 kg pulley, supports 100 N/µm: mode 2	81554.1 rpm	81554.88 rpm	9.6×10^{-6}	0.5 %	**pass**
Stepped motor shaft, overhung 2.5 kg pulley, supports 100 N/µm: mode 3	123142.5 rpm	123145 rpm	0.002 %	0.5 %	**pass**
Stepped gearbox shaft, 6 kg gear and 0.8 kg pinion, supports 200 N/µm: mode 1	31217.7 rpm	31217.66 rpm	1.1×10^{-6}	0.1 %	**pass**
Stepped gearbox shaft, 6 kg gear and 0.8 kg pinion, supports 200 N/µm: mode 2	114114 rpm	114114.5 rpm	4.3×10^{-6}	0.5 %	**pass**
Stepped gearbox shaft, 6 kg gear and 0.8 kg pinion, supports 200 N/µm: mode 3	292540.3 rpm	292560.2 rpm	0.0068 %	0.5 %	**pass**

Motor rotor on ball bearings: first critical speed with the magnetic pull

Kind of reference	The same cases calculated by other programs or methods
Implements	—
Reference source	Closed form: a rigid rotor of mass m centred on two equal supports whirls in its softer plane at ω = √(k/m), k the net stiffness there: the bearings' tangent stiffness at their loads less the magnetic pull's k_m (k from the static solve by a central difference of ±20 N across the rotor)
Cases	2 of 2 within tolerance

A rigid, near-massless shaft (E 1000 × steel's, density 10^{-6} of it) between two 6312 bearings, a 25 kg rotor in a 0.6 mm gap at 0.65 T. The static eccentricity is a bound: its pull's preload of the bearings is not credited, so the reference at 10 % eccentricity is the centred rotor's. The stiffness k is the engine's own static solve's: the comparison checks the modes against the statics and the state they stand on, not the bearing's stiffness itself (the bearing solver has its own validation).

Case	Reference	Axomek	Deviation	Tolerance	
centred rotor	11859.3 rpm	11885.87 rpm	0.22 %	1 %	**pass**
10 % static eccentricity	11859.3 rpm	11885.87 rpm	0.22 %	1 %	**pass**

Not validated against a published reference

- Unbalance response, and the torsional and axial modes of a system of shafts

Stated limits

Method and limits (help: modules/shaft.md)

- Timoshenko beam elements (with shear deformation); supports rigid or linear springs. Stresses from bending, axial force and torsion combined by von Mises.
- Nominal stresses only: stress concentrations at shoulders and grooves are not in the stress result or the stress check. The strength check applies notch factors.
- Without coupling, bearings see only forces; the supports are rigid or the springs you give.
- Not checked: shoulder heights (that each bearing and hub has a shoulder high enough to sit against) and the assembly order (that each part slides on from an end past the seats before it). A shaft made lighter by **Make it pass**, the design search or the Assistant keeps its checks, not its shoulders: check that each seat still has one and that the shaft can be assembled.

Method and limits (help: modules/shaft-vibration.md)

- Timoshenko beam elements with the consistent translational and rotary mass, supports rigid or with their stiffness. Mode shapes by inverse iteration.
- The bending critical speeds listed first are non-rotating. Gyroscopic stiffening enters only the whirl results, the Campbell diagram, the unbalance response and the governing critical speed when a part has polar inertia.
- Supports and bearings are linear springs at one operating point. The speed dependence of bearing stiffness is not included, and neither are oil film bearings, seals or fluid forces.
- Damping is structural, one ratio for the shaft, supports and housings. There is no stability (logarithmic decrement) analysis.
- The coupled-plane whirl on supports that differ between the planes is solved in the lowest 24 non-rotating modes of each plane.

- Torsional modes are for the shaft alone, free at both ends: the inertia of a motor or a driven machine on the far side of a coupling is not included unless you model it as a part.
- The critical speeds are checked against ROSS 3.0, an independent open-source rotordynamics program, in the verification suite.

Limits of the model (help: theory/shaft-dynamics.md)

- The vibration analyses take each bearing's stiffness at its load, without its speed dependence.
- Damping is structural and proportional; fluid-film bearings and seals are not modelled.
- `criticalSpeeds` are always the non-rotating values; the whirl results add the gyroscopic effect.
- A housing enters as springs, or as the seat stiffness of a housing modelled by finite elements.

Known limits

- Units (R-387, R-931): the app and axomek read the same units typed, with the same factors, and show the same US customary units; a test holds them to it (`units-parity.spec.ts`), also for a field shown in another unit than its value's. It lists what still differs, on purpose or refused (never read wrongly): a design search's bound on a deflection is typed in µm (mil) in the app, as its check shows it, and in mm (in) by axomek `optimize --require`, which refuses nm there; a bolt's expansion coefficient α_S is typed in 10^{-6}/K in the app, in 1/K in axomek. A bare number in a bearing clearance is in µm in the app (the unit it shows) and in mm in axomek without `--units imperial` (base units, as every bare argument; mil with it, as in the app, but mm again with `--json`, which always reads base units). The bolt's expansion field refuses its own unit typed out (`12 10⁻⁶/K` reads as two numbers): type `12e-6 1/K`, or a bare 12 with metric units shown (with US units shown a bare number is in 10^{-6}/°F). The text output and the report keep a few units metric that the app shows in US units (m, J, GPa, √MPa, mm³, a steering's feel: `SCREEN_ONLY` in `crates/axm-model/src/units.rs`). Not compared: fields the app shows in % of a plain fraction (a torque amplitude, a modulus share), and the change lists of the Assistant and `axomek diff`, which give a clearance in mm.
- A motor bearing's minimum load and gyroscopic slip over its rotor's static eccentricity (user test rr1-motor; see "From 0.3.0"). While any rotor in the project pulls, their pass (or a warning for a bearing its axial load holds) counts only in the case rated in full, where all of these hold: - an uncoupled shaft alone: Couple linked bearings off, in no group (a rigid coupling, a bearing between shafts) and in no housing whose seats move each other; - no load cases; - the minimum load checked on the radial load F_r alone (not the equivalent load P of angular contact, tapered roller and thrust bearings), one bearing on its support; - no axial load on the shaft (an axial load shares out anew and can hold or let go of a bearing between the states); - the least loaded state (the least radial load, linear in the pulls without coupling) solved exactly, and the bearing loaded at the eccentricity. Every other such pass or held warning is not decided, saying why: "enter the eccentricity 0 to see it" (the Electric motor example, coupled, with an eccentricity entered, is one; it ships centred). Every bearing's gyroscopic slip margin is not decided in every case while any rotor pulls: the least radial load does not bound its balls' gyroscopic margin (a later release bounds the margin itself). A later release: a bound on the coupled shaft's least bearing load between centred and the eccentricity (the bearings' load–deflection laws bound how far load moves between them), solved with the groups' and housings' own equilibrium in each state and per load case; P, the axial holding and the balls' loads minimised themselves.

- With a rotor's pull, a one-way bearing's axial direction check can still pass in a narrow window between the states, as its 5 % threshold scales with the radial load; and a shaft's own checks (static and fatigue strength, deflection) are rated at the eccentricity as given, where a pull against the weight bends the span less than centred. A later release rates them at both states.
- A test over every example and five more cases (a hybrid, a four-point contact bearing, a designation its geometry contradicts, the electric motor pulled off centre, a life below κ 0.1 failing beside a determined fail) replaces every figure the engine withholds and checks that no display shows it and no verdict or ranking moves with it (the "poison" harness). What it still finds is listed, each with its reason, in `crates/axm-cli/tests/poison/expected-fail.json`. Beside the limits above it found: the report's appendix of calculations gives a figure not determined as a number (a contradicted bearing's life, the pulled motor's critical speed); `axomek diff --json` and Compare projects give the check and result rows of such a life their numbers without an `outsideMethod` flag; the design search's `--json` gives a candidate's `objective` with no flag of its own where it is such a figure or one not determined (its `undetermined` names the metric; the search does not choose by it); `axomek find-bearing --json` lists a candidate's checks in an order its outside-method life's share moves (the candidates' order does not); and `axomek solve --json` gives `shaft.governingCriticalSpeed` with no flag where the check `minCriticalSpeed` is not determined (read that check's `undetermined`). A bearing's contact stress shows as a number where the static safety that goes by it is not determined: in the Overview's key results, `axomek solve`'s `p max` and the report's Max contact stress (read the static safety's "not determined" beside it). The app's checks panel picks µm or mm for the limit of a length check below 1 mm by the value as modelled, also where the check says "not determined" (no fixture has such a check yet). The harness itself does not tell a leak that moves to the same words on another line, a further difference inside a line whose wording already differs, or a leak moving to another item at the same place of a list from the one its table lists; nor, in the report's decisions, items whose ids are numbers alone ("1", "2").
- A gear's mass is warned of only far off its geometry's (above twice a solid disc, below a quarter of the disc less the shaft): a mass left from a gear up to about twice smaller, or from one only a little larger, is not, and the critical speed then reads somewhat too low or too high. Check each gear's mass after resizing an example.

- An electric motor's critical speed and magnetic stability with a static eccentricity on coupled bearings (review of ut4-motor-critical-speed; see "From 0.3.0"). In 0.3.0: - A pull not along a bearing's other loads (more than 1° apart: a belt or a gear's force across the weight) leaves both not rated: the bearing's load turns as the eccentricity grows, and its stiffness along y or z can be least in between, which neither the states taken nor a bound with each bearing equally stiff every way at its least load bounded (a reviewer's case read 9 988 rpm where a state inside the bound gives 9 604 rpm). This blocks a belt- or gear-driven motor with an eccentricity from the coupled critical speed, and 0.3.0 names no way around it: no stiffness entered on the supports (with coupling off) is known to be safe, as the least one lies somewhere between centred and pulled (keep the eccentricity: its pull counts in the bearing loads and the air gap; 0 would drop it from them). - A state the coupling cannot settle exactly (bearings loose in their clearance, e.g. the Electric motor example without its wave spring, centred) leaves a passing result not decided (a warning). A motor whose bearings carry no load with the rotor centred (the weight off, a vertical motor without preload) has both not rated, with a hint (a horizontal motor: turn the weight on; a vertical one, or one held only by a preload, 0.3.0 cannot rate). The hints of the not decided and not determined results name no stiffness to enter: none is known to be safe there (review: a recipe of the bearings' stiffness without the pull could pass a motor that fails). - A coupled bearing under an axial load (a spring preload, as in the Electric motor example, or the shaft's own axial loads) or preloaded from within (an X or O pair's preload, a negative operating clearance) leaves a passing result not decided (a warning): its radial stiffness need not grow with its radial load, so the states taken bound nothing between them (a reviewer's case: e = 0.2 passed 15 000 rpm at 15 908 rpm where e = 0.15 gives 13 878 rpm). - A spring preload is refused on any shaft joined to another by a bearing between them (shafts solved together do not carry it), also where the spring would act on its own housing. - The unbalance response is the given eccentricity's, with a warning: at a smaller one it can be larger. - Up to four pulling rotors on a shaft; more have both not rated. The axial modes of a state keep the locating bearings' axial stiffness and the housings' seat motions of the pulled state. A critical speed or magnetic stability that is not rated fails, shown as "not rated" in place of its placeholder value (0 rpm, a share of 100 %), and What matters most leaves it out, saying why. Planned for a later release: the rotor pulled within its bearings' clearance to where they hold it (it never sits centred there), in place of the centred state; and a sweep of the eccentricity with a proven bound between its steps, for a pull across the bearings' loads, for bearings under an axial load (a spring preload) and for the unbalance response.
- Rotating loads, for a later release (user test ut4, Codex review of ut4-small): a bearing's loads are taken as fixed in direction. A load that turns with the shaft (an unbalance, an exciter) has no setting of its own: **Outer ring rotates** gives its load zone and life, but also the cage and element speeds, the friction and the high-speed model's centrifugal and gyroscopic forces of a turning outer ring, which the bearing does not have. The app and the help say to check the life both ways and keep the shorter. A load of unknown direction, or both senses of rotation, is not a study of its own: give each direction worth checking as a load case (a negative factor reverses a load).
- A mounted ball screw that does not derive the load cases (one step, or linked to them) is the shaft and turns at its speed: entered at another speed (0 too) it is refused, not rated at its own, whose critical speed, d·n and life would be at a speed it does not turn at; rating it at the shaft's speed is for a later release.

- With gravity on, a shaft placed by a bearing in another shaft (seated in it) is refused when the two are solved together and its frame is turned by what places that shaft (a universal joint, a worm pair, a rack mesh): the model of shafts solved together leaves the bearings between them out, and with them that placement (0.3.0). Place the shaft by another connection or at coordinates, or switch gravity off.
- The unbalance response without shear deformation, read within 1/100 of an element from a node (through the element's shape functions), omits structural damping on the in-element bending term, so it reads high there: up to about 8 % at $\zeta = 0.2$.
- A deflection, tilt or critical speed on rigid supports (bearings linked, "Couple linked bearings" off) leaves out the bearings' give; the results, checks and "Show the calculation" say so, and the results offer "Check with the bearings' stiffness". Coupling linked bearings by default is for a later release (user test ut4).

Shaft strength: DIN 743 and the FKM guideline

Fatigue and static strength of shafts at their notches by DIN 743 and by the FKM guideline, with damage accumulation over a duty cycle.

Methods and standards

Shaft strength: DIN 743-1/-2:2012 with nominal stresses from the shaft solution.

Technological size factor K1(deff) by DIN 743-2 eqs. (10)–(14) for the material class (quenched and tempered, case-hardening, Cr-Ni-Mo case-hardening, structural or nitriding steel) at the largest shaft diameter, the tensile-strength K1 for the fatigue limits and the yield-strength K1 for the support number and yield points; material fatigue limits 0.4·σ_B (tension), 0.5·σ_B (bending), 0.3·σ_B (torsion).

Notch factors: shoulders and round grooves β = α/n (α by eq. (9) and Table 3, n from the related stress gradient of Table 2, eq. (5) or eq. (6) for hard layers, n ≤ α); cross bores by 5.2.3 (Bild 12: ασ = 3 − 2r/d in tension, 3 + 1.4·(2r/d) − 2.8·√(2r/d) in bending, ατ = 2.023 − 1.125·√(2r/d), stresses on the net section, G' = 2.3/r, 2.3/r + 2/d and 1.15/r + 2/d); keyways and press fits from Table 1 with σ_B(d) = K1·σ_B, scaled from 40 mm by K3 (eq. (3)), KF = 1, two keyways with both factors times 1.15; retaining-ring grooves by 4.2.4 (structural radius, corrected radius, width correction, caps 4 and 2.5, scaled from 30 mm) and V-notches by 4.2.3 (β_σ = 0.109·σ_B(d)/100 + 1.074 in tension, 0.0923·σ_B(d)/100 + 0.985 in bending, βτ = 0.8·β_σ,b at 15 mm, scaled by K3) with KF = KF(R_z)/KF(20 µm).

Geometric size factor K2(d), roughness factor KF (eqs. (18), (19)), surface strengthening KV (lower bounds of Table 4: rows "a" for unnotched parts and β = α/n, rows "b" for measured notch factors when β exceeds them; the d ≤ 25 mm value up to 25 mm, linear to the 25–40 mm value at 40 mm and on with that slope down to 1, as clause 8 describes the trend of Bilder 17 to 19; KV = 1 in tension of a plain section); K = (β/K2 + 1/KF − 1)/KV.

Mean stress sensitivity ψ = σWK/(2·K1·σ_B − σWK); case 2 (constant ratio, default) or case 1 (constant mean), each limited by the component yield point; compressive normal mean stresses give σ_mv = sign(H)·√|H| with H = −(σzdm + σbm)² + 3τm² and τ_mv = 0 when σ_mv < 0, and beyond σ_mv,grenz (case 1) or (σ_mv/σa)grenz (case 2) the amplitude strength follows Annex A Table A.1 with σdFK = σzd,bFK; S = 1/√((σzda/σzdADK + σba/σbADK)² + (τta/τtADK)²).

Static safety by eq. (25) with the magnitudes and σFK = K1·K2F·γF·σS: K2F 1.2 (hollow 1.1 bending, 1.0 torsion), γF from ασ (β_σ for ring grooves and V-notches) at circumferential notches; hard layers take K2F = γF = 1 and add the incipient crack check of 5.1.2 (eq. (28) with the local stresses α·σmax against σBRand: 2300 MPa for case hardening and 1700 MPa for nitriding by 5.3 unless given; none without a value).

The material fatigue limits are those of DIN 743-3:2024 eqs. (1) to (3): 0.5·σ_B in bending, 0.4·σ_B in tension and compression, 0.3·σ_B in torsion.

With a duty cycle the worst load case governs.

Shoulders are checked on the smaller diameter at every diameter step, with the forces and bore of that smaller section.

Where a gear's teeth cut into the shaft end at a step up (where its teeth end), that step is rated instead of a shoulder by DIN 743-2 4.2.2's measured notch factors of a shaft with involute teeth (Figure 1 curve 2: β* = exp(4.2·10⁻⁷·σ_B(d)²), βτ = 1 + 0.75·(β* − 1), β_σ = 1 + 0.49·(β* − 1) at 29 mm, scaled by eq. (3), KF = 1; β = 1 and KV = 1 when case hardened) at the root diameter, and as the closest eq. (9) notch (a shoulder whose fillet is the hob's run-out radius, 25 mm unless given, or a round groove whose depth is taken from the larger section), the lower safety governing: no published method for a hob's run-out is used. The toothed length takes the toothed shaft's factors too (its automatic section is the toothed section).

Shaft strength (method FKM): FKM guideline, 7th edition (2020), nominal stress approach for non-welded rod-shaped components (chapter 1 static, chapter 2 fatigue) with the same nominal stresses and notch positions as DIN 743.

Component strengths R_m = Kd,m·K_A·R_m,N and Rp = Kd,p·K_A·Rp,N with the technological size factor Kd = (1 − 0.7686·ad·lg(deff/7.5 mm))/(1 − 0.7686·ad·lg(deff,N/7.5 mm)) (eq. 1.2.8) at the largest shaft diameter, held at 250 mm for rolled steel, none for stainless steel; deff,N and ad of the grade from the material tables of section 5.1 (e.g. C45 0.16/0.36, 42CrMo4 0.32/0.43, 16MnCr5 0.44/0.44 at 16 mm; S355 0.08 at 3 mm/0.23 at 16 mm), else the group's default or the input.

Aluminium: wrought Kd = 1 (the strengths of the product's size are entered), cast 1.1·(deff/7.5 mm)^−0.2 between 12 and 150 mm, 0.6 above (eqs. 1.2.10 to 1.2.12).

Material fatigue strengths σW,zd = f_W,σ·R_m and τW,s = f_W,τ·σW,zd (Table 2.2.1: f_W,σ 0.45 quenched and tempered and structural steel, 0.40 case-hardening and stainless steel, 0.34 cast steel, 0.25 + 70 MPa/R_m,N for GJS, 0.30 aluminium; f_W,τ 0.577, GJS 0.65, cast aluminium 0.75).

Fatigue notch factors Kf = Kt/(n(r)·n(d)) (eq. 2.3.2) with Kt from the formulas of section 5.2 (the same as DIN 743-2), the notch's own stress gradient G(r) (shoulder 2.3·(1 + φ)/r, groove 2·(1 + φ)/r, shear 1.15/r and 1/r, Table 2.3.5) and the nominal gradient G(d) = 2/d for bending and torsion, each with the support number of Siebel and Stieler (eqs. 2.3.6 to 2.3.8, aG and bG of Table 2.3.1, for shear with f_W,τ·R_m); keyways (two keyways times 1.15), press fits, ring grooves and V-notches use the measured notch factors of DIN 743-2 divided by n(d), with KR = 1; cross bores Kt of DIN 743-2 Bild 12 (section 5.2 has none for round bars) on the net section with G(r) = 2.3/r (Table 2.3.5) and 1.15/r in shear.

Roughness factor KR,σ = 1 − aR,σ·lg(R_z)·lg(2Rm/R_m,N,min), KR,τ with f_W,τ (Table 2.3.6); surface treatment factor KV the lower bound of Table 2.3.8 (notched or unnotched, steel or cast iron, the small-specimen value at d ≤ 15 mm); design factor KWK = (Kf + 1/KR − 1)/KV.

Mean stress factor KAK (section 2.4.2) for overload case F2 (constant stress ratio, default) or F1 (constant mean) with Mσ = aM·10⁻³·R_m + bM (Table 2.4.1: steel 0.35/−0.1, cast steel 0.35/0.05, GJS 0/0.4; at least 0) and Mτ = f_W,τ·Mσ, at the equivalent mean stress q·Sm,v,NH + (1 − q)·Sm,v,GH (Tm,v = f_W,τ·Sm,v) when the normal mean stresses are not negative, unless a single mean stress is larger.

Variable amplitude factor for N constant-amplitude cycles: (ND/N)^(1/k) below ND = 10⁶, beyond it 1 on the S-N curve type I (steel, cast iron: k 5 normal, 8 shear; 15 and 25 when surface hardened) and (ND/N)^(1/kII) to ND,II = 10⁸, fII = (ND/ND,II)^(1/kII) beyond and for infinite life on type II (aluminium, austenitic steel: kII 15 and 25, Table 2.4.3).

SBK = KBK·KAK·SWK, at most 0.75·Rp·KT,p·npl·KAK (tension without npl, eq. 2.4.29).

With a duty cycle and $\bar{N}$ cycles the fatigue check accumulates damage (section 2.4.3.5.2): the cycles are shared by the cases' time shares (times their speeds when every case sets one); each case's amplitudes become fully reversed ones by its F2 mean stress factor (eq. 2.4.5); each stress type's spectrum gives KBK by Miner consistent (iterated, eqs. 2.4.43 to 2.4.49, default) or Miner elementary (eqs. 2.4.39, 2.4.42) with the effective Miner sum D_m = min(max(2/A^¼, D_m,min), 1) (eq. 2.4.41; D_m,min 0.3 for steel, cast steel and aluminium, 1 for GJS, Table 2.4.4) unless given, at least 1 (type I) or the single-step value (type II) and at most SBK,max/SWK with KAK = 1; aBK = Sa,W,1·j_D/(KBK·SWK) per stress type, combined as below; worked example 6.2's spectrum is reproduced in the verification suite.

Cyclic degrees of utilisation aBK = Sa·j_D/SBK, combined by q·aNH + (1 − q)·aGH with q from f_W,τ (0 for steel and cast steel: √((aBK,zd + aBK,b)² + aBK,t²); 1 for surface-hardened parts).

Static check: SSK = Rp·npl, TSK = fτ·Rp·npl with npl = min(√(Rp,max/Rp), fR·Kp), fR = ½·(1 + R_m/Rp), Rp,max 1150 MPa for steel and cast steel (750 for GJS) at Kt ≤ 3, else E·εertr/Kt² (Table 1.3.1), Kp 1, 1.70, 1.33 for a round bar (tube by eq. 1.3.6); aSK combined as for fatigue.

Safety factors: j_D = jF·jG/KT,D with jF 1.5, 1.4 or 1.3 for high, medium or low consequences without inspection and 1.35, 1.25 or 1.2 with regular inspection (Table 2.5.1); j_ges = jG·max(jm/KT,m·Rp/R_m, jp/KT,p) with jm/jp from the consequences and the probability of the peak load (Table 1.5.1); the cast factor jG 1.4, 1.25 or 1.0 (Table 2.5.2) for cast steel, GJS and cast aluminium.

Temperature factors above normal temperature (100 °C; aluminium 50 °C, non-age-hardenable 100 °C for the static factor): steel KT,D = 1 − 1.4·10⁻³·(T − 100), KT,m = KT,p = 1 − 1.7·10⁻³·(T − 100); cast steel 1.2 and 1.5·10⁻³; GJS 1 − 1.6·(10⁻³T)² and 1 − 2.4·(10⁻³T)²; aluminium 1 − 1.2·10⁻³·(T − 50) and 1 − 4.5·10⁻³·(T − 50 or 100) (eqs. 2.2.6, 1.2.26 to 1.2.32), up to 500 °C (aluminium 200 °C); none for stainless steel above 100 °C; the creep values KTt are not used.

The reported safeties are j_D/aBK and j_ges/aSK, checked against j_D and j_ges, so a check passes exactly when aBK ≤ 1 and aSK ≤ 1.

Coating and residual stress factors are 1; the material-mechanics support number and the local stress approach are not used; with a duty cycle but no number of cycles the worst load case governs (infinite life).

Worked example 6.1 of the guideline is reproduced in the verification suite.

Duty cycle: Palmgren–Miner rule, 1/L = Σ t_i/L_i with each case's life at its own speed.

System life: L = (Σ L_i^−e)^−1/e with Weibull slope e = 10/9 (balls) or 9/8 (rollers).

Validation

154 of 154 cases within their tolerance, in 4 groups.

Worked examples of standards and books

Shaft strength: DIN 743 Beiblatt 1 examples

Kind of reference	Worked examples of standards and books
Implements	DIN 743-1/-2:2012; DIN 743-3:2024
Reference source	DIN 743 Beiblatt 1:2024-07, examples 5.2, 5.3 and 5.4
Cases	78 of 78 within tolerance

The three worked examples of DIN 743 Beiblatt 1:2024-07 (DIN 743-1/-2:2012, material values of DIN 743-3:2024 as the examples print them): 5.2 shoulder in bending and torsion (case 1), 5.3 keyway with rotating bending and steady torsion (case 1, also solved through the shaft model), 5.4 retaining-ring groove in an axle (case 2). Every printed intermediate and final value is compared; the examples carry rounded intermediate values, so the tolerance is 0.2 % (1 % for values printed to two figures).

Closest to its tolerance: The fatigue safeties of 5.3 and 5.4 (1.2677 and 1.1878 against the printed 1.27 and 1.19) use 92 % of the 0.2 % tolerance, which is tighter than half the printed digit (0.4 %); both round to the printed values.

Case	Reference	Axomek	Deviation	Tolerance	
5.2 shoulder: K_1(d_eff) tensile	0.871	0.871339	0.039 %	0.2 %	**pass**
5.2 shoulder: K_1(d_eff) yield	0.832	0.831751	0.03 %	0.2 %	**pass**
5.2 shoulder: σ_S(d)	665.6 MPa	665.4008 MPa	0.03 %	0.2 %	**pass**
5.2 shoulder: α_σ bending	1.557	1.557282	0.018 %	0.2 %	**pass**
5.2 shoulder: α_τ torsion	1.283	1.282877	0.0096 %	0.2 %	**pass**
5.2 shoulder: G' bending	0.542 1/mm	0.5424711 1/mm	0.087 %	0.2 %	**pass**

Case	Reference	Axomek	Deviation	Tolerance	
5.2 shoulder: G' torsion	0.23 1/mm	0.23 1/mm	1.2×10^{-16}	0.2 %	**pass**
5.2 shoulder: n bending	1.04	1.040053	0.0051 %	0.2 %	**pass**
5.2 shoulder: n torsion	1.026	1.02608	0.0078 %	0.2 %	**pass**
5.2 shoulder: β_σ bending	1.497	1.49731	0.021 %	0.2 %	**pass**
5.2 shoulder: β_τ torsion	1.25	1.25027	0.022 %	0.2 %	**pass**
5.2 shoulder: K_2(d)	0.885	0.8849853	0.0017 %	0.2 %	**pass**
5.2 shoulder: KFσ	0.902	0.9017146	0.032 %	0.2 %	**pass**
5.2 shoulder: KFτ	0.944	0.9434859	0.054 %	0.2 %	**pass**
5.2 shoulder: K_σ bending	1.8	1.800902	0.05 %	0.2 %	**pass**
5.2 shoulder: K_τ torsion	1.472	1.472657	0.045 %	0.2 %	**pass**
5.2 shoulder: σbWK	241.9 MPa	241.9174 MPa	0.0072 %	0.2 %	**pass**
5.2 shoulder: τtWK	177.5 MPa	177.5035 MPa	0.0019 %	0.2 %	**pass**
5.2 shoulder: ψbσK	0.1613	0.1611965	0.064 %	0.2 %	**pass**
5.2 shoulder: ψτK	0.1135	0.1134081	0.081 %	0.2 %	**pass**
5.2 shoulder: σmv	529.1 MPa	529.1503 MPa	0.0095 %	0.2 %	**pass**
5.2 shoulder: τmv	305.5 MPa	305.5051 MPa	0.0017 %	0.2 %	**pass**
5.2 shoulder: σ_bADK (case 1)	156.6 MPa	156.6202 MPa	0.013 %	0.2 %	**pass**
5.2 shoulder: τ_tADK (case 1)	142.8 MPa	142.8567 MPa	0.04 %	0.2 %	**pass**
5.2 shoulder: fatigue safety S	2.62	2.61697	0.12 %	0.2 %	**pass**
5.2 shoulder: γ_F bending (from α_σ)	1.05	1.05	0 %	exact	**pass**
5.2 shoulder: σbFK	838.7 MPa	838.405 MPa	0.035 %	0.2 %	**pass**
5.2 shoulder: τtFK	461.1 MPa	461.0032 MPa	0.021 %	0.2 %	**pass**
5.2 shoulder: static safety S	1.4	1.400464	0.033 %	0.2 %	**pass**
5.3 keyway: σ_B(d) = K_1·σ_B(d_B)	958.1 MPa	958.4729 MPa	0.039 %	0.2 %	**pass**
5.3 keyway: β_σ(d_BK = 40 mm)	2.95	2.952036	0.069 %	0.2 %	**pass**
5.3 keyway: K_3(d)	0.94	0.9404568	0.049 %	0.2 %	**pass**
5.3 keyway: K_3(d_BK)	0.947	0.9474604	0.049 %	0.2 %	**pass**
5.3 keyway: β_σ(d)	2.972	2.97402	0.068 %	0.2 %	**pass**
5.3 keyway: K_2(d)	0.873	0.8733452	0.04 %	0.2 %	**pass**
5.3 keyway: K_σ (K_F = 1)	3.404	3.40532	0.039 %	0.2 %	**pass**
5.3 keyway: σba	97.8 MPa	97.7848 MPa	0.016 %	0.2 %	**pass**
5.3 keyway: τtm	122.2 MPa	122.231 MPa	0.025 %	0.2 %	**pass**
5.3 keyway: σmv	211.7 MPa	211.7103 MPa	0.0049 %	0.2 %	**pass**
5.3 keyway: σbWK	140.7 MPa	140.7317 MPa	0.023 %	0.2 %	**pass**
5.3 keyway: ψbσK	0.0792	0.07923127	0.039 %	0.2 %	**pass**
5.3 keyway: σ_bADK (case 1)	123.9 MPa	123.9576 MPa	0.047 %	0.2 %	**pass**
5.3 keyway: fatigue safety S	1.27	1.267658	0.18 %	0.2 %	**pass**
5.3 keyway: σ_bFK (γ_F = 1)	898.6 MPa	898.2911 MPa	0.034 %	0.2 %	**pass**
5.3 keyway: τtFK	518.8 MPa	518.6286 MPa	0.033 %	0.2 %	**pass**
5.3 keyway: static safety S	2.57	2.567993	0.078 %	0.2 %	**pass**

Case	Reference	Axomek	Deviation	Tolerance	
5.3 keyway in the shaft model: σba	97.8 MPa	97.7848 MPa	0.016 %	0.2 %	**pass**
5.3 keyway in the shaft model: τtm	122.2 MPa	122.231 MPa	0.025 %	0.2 %	**pass**
5.3 keyway in the shaft model: β_σ(d)	2.972	2.97402	0.068 %	0.2 %	**pass**
5.3 keyway in the shaft model: Kσ	3.404	3.40532	0.039 %	0.2 %	**pass**
5.3 keyway in the shaft model: σbWK	140.7 MPa	140.7317 MPa	0.023 %	0.2 %	**pass**
5.3 keyway in the shaft model: σbADK	123.9 MPa	123.9576 MPa	0.047 %	0.2 %	**pass**
5.3 keyway in the shaft model: fatigue safety S	1.27	1.267658	0.18 %	0.2 %	**pass**
5.3 keyway in the shaft model: static safety S	2.57	2.567993	0.078 %	0.2 %	**pass**
5.4 ring groove: K_1(d_eff) tensile	1	1	0 %	exact	**pass**
5.4 ring groove: K_1(d_eff) yield	0.897	0.8965356	0.052 %	0.2 %	**pass**
5.4 ring groove: σ_S(d)	300.5 MPa	300.3394 MPa	0.053 %	0.2 %	**pass**
5.4 ring groove: ρ*	0.107 mm	0.1070247 mm	0.023 %	0.2 %	**pass**
5.4 ring groove: rf	0.56 mm	0.5603717 mm	0.066 %	1 %	**pass**
5.4 ring groove: β_σ(d_BK = 30 mm), m/t ≥ 1.4	2.744	2.7437	0.011 %	0.2 %	**pass**
5.4 ring groove: K_3(d)	0.932	0.9320375	0.004 %	0.2 %	**pass**
5.4 ring groove: K_3(d_BK)	0.959	0.9594314	0.045 %	0.2 %	**pass**
5.4 ring groove: β_σ(d)	2.823	2.824341	0.047 %	0.2 %	**pass**
5.4 ring groove: K_2(d)	0.845	0.8449536	0.0055 %	0.2 %	**pass**
5.4 ring groove: K_F(R_z 25)	0.86	0.8601139	0.013 %	1 %	**pass**
5.4 ring groove: K_F(R_zB 20)	0.87	0.8698113	0.022 %	1 %	**pass**
5.4 ring groove: K_Fσ = K_F(R_z)/K_F(R_zB)	0.989	0.9888512	0.015 %	0.2 %	**pass**
5.4 ring groove: Kσ	3.352	3.353873	0.056 %	0.2 %	**pass**
5.4 ring groove: σbm	182 MPa	182.0144 MPa	0.0079 %	0.2 %	**pass**
5.4 ring groove: σba	56.9 MPa	56.87951 MPa	0.036 %	0.2 %	**pass**
5.4 ring groove: σbmax	238.9 MPa	238.8939 MPa	0.0025 %	0.2 %	**pass**
5.4 ring groove: σbWK	85 MPa	84.97638 MPa	0.028 %	0.2 %	**pass**
5.4 ring groove: ψbσK	0.0806	0.08054453	0.069 %	0.2 %	**pass**
5.4 ring groove: σ_bADK (case 2)	67.6 MPa	67.56262 MPa	0.055 %	0.2 %	**pass**
5.4 ring groove: fatigue safety S	1.19	1.18782	0.18 %	0.2 %	**pass**
5.4 ring groove: γ_F (from β_σ)	1.1	1.1	0 %	exact	**pass**
5.4 ring groove: σbFK	396.7 MPa	396.448 MPa	0.064 %	0.2 %	**pass**
5.4 ring groove: static safety S	1.66	1.659515	0.029 %	0.2 %	**pass**

Shaft strength: FKM damage accumulation

Kind of reference	Worked examples of standards and books
Implements	FKM guideline, 7th edition (2020), 2.4.3 (variable amplitudes)
Reference source	FKM guideline, 7th edition (2020), worked example 6.2 and Table 6.2.1
Cases	8 of 8 within tolerance

FKM guideline, 7th edition (2020), worked example 6.2 (cast bearing block, EN-GJL-250): overload case F2 with Mσ = 0.5 for the amplitudes 47 and 12 MPa at mean stresses 13 and 4 MPa (σWK 115 and 69 MPa), and the spectrum of Table 6.2.1 (Sa,i/Sa,1 = 1, 0.8, 0.5, 0.2 with 10^3, $4{\cdot}10^4$, $6.8{\cdot}10^5$ and $1.4{\cdot}10^6$ cycles) on the S-N curve type I with k = 5, ND = 10^6 and D_m = D_m,min = 1. Miner elementary as printed; Miner consistent against the note below the example, which prints 1.97 (the iteration of eqs. 2.4.43 to 2.4.46 gives 1.977; the note's σBK of 199 and 116 MPa follow from 1.97, so it truncates).

Case	Reference	Axomek	Deviation	Tolerance	
Mean stress factor KAK,1	0.879	0.8785047	0.056 %	0.1 %	**pass**
Mean stress factor KAK,2	0.857	0.8571429	0.017 %	0.1 %	**pass**
Sum Σ (n_i/N̄)·(Sa,i/Sa,1)^k	0.01688	0.01688128	0.0076 %	0.1 %	**pass**
Distance Aele	59.2	59.23721	0.063 %	0.1 %	**pass**
Effective Miner sum D_m	1	1	0 %	10^{-9}	**pass**
Variable amplitude factor KBK (elementary)	1.946	1.946305	0.016 %	0.05 %	**pass**
Variable amplitude factor KBK (consistent)	1.97	1.976991	0.35 %	0.4 %	**pass**
Component strength σBK,1 = KBK·σAK (elementary)	197 MPa	196.5768 MPa	0.21 %	0.3 %	**pass**

Shaft strength: FKM guideline

Kind of reference	Worked examples of standards and books
Implements	FKM guideline, 7th edition (2020), nominal stresses (chapters 1 and 2)
Reference source	FKM guideline, 7th edition (2020), worked example 6.1
Cases	45 of 45 within tolerance

FKM guideline, 7th edition (2020), worked example 6.1 (shaft shoulder), nominal stress approach: 41Cr4 quenched and tempered (R_m,N 1000, R_p,N 800 MPa at d_eff,N 16 mm; ad,m and ad,p from the material table), shoulder D 50 / d 42 mm with r 5 mm, R_z 10 µm, rotating bending ±150 MPa with torsion 50 ± 80 MPa in phase, overload case F2, medium consequences with regular inspection; support numbers after Siebel and Stieler. References are the example's printed (rounded) values.

Case	Reference	Axomek	Deviation	Tolerance	
Size factor Kd,m	0.895	0.8954224	0.047 %	0.1 %	**pass**
Size factor Kd,p	0.841	0.8405086	0.058 %	0.1 %	**pass**
Tensile strength R_m	895 MPa	895.4224 MPa	0.047 %	0.1 %	**pass**
Yield strength R_p	672 MPa	672.4069 MPa	0.061 %	0.1 %	**pass**
Stress concentration Kt,b	1.557	1.557282	0.018 %	0.1 %	**pass**
Stress concentration Kt,t	1.283	1.282877	0.0096 %	0.1 %	**pass**
Material fatigue strength σW,zd	403 MPa	402.9401 MPa	0.015 %	0.2 %	**pass**
Material fatigue strength τW,s	232 MPa	232.4964 MPa	0.21 %	0.4 %	**pass**
Nominal stress gradient G(d)	0.0476 1/mm	0.04761905 1/mm	0.04 %	0.2 %	**pass**
Notch stress gradient Gσ(r)	0.542 1/mm	0.5424711 1/mm	0.087 %	0.2 %	**pass**

Case	Reference	Axomek	Deviation	Tolerance	
Notch stress gradient Gτ(r)	0.23 1/mm	0.23 1/mm	1.2×10^{-16}	0.2 %	**pass**
Support number nσ(d)	1.022	1.022189	0.019 %	0.1 %	**pass**
Support number nσ(r)	1.109	1.10853	0.042 %	0.1 %	**pass**
Support number nτ(d)	1.031	1.03065	0.034 %	0.1 %	**pass**
Support number nτ(r)	1.098	1.097613	0.035 %	0.1 %	**pass**
Mean stress sensitivity Mσ	0.213	0.2133978	0.19 %	0.3 %	**pass**
Mean stress sensitivity Mτ	0.123	0.1231306	0.11 %	0.3 %	**pass**
Fatigue safety factor j_D	1.25	1.25	0 %	10^{-9}	**pass**
Total static safety factor j_ges	1.4	1.4	0 %	10^{-9}	**pass**
Fatigue notch factor Kf,b	1.374	1.374322	0.023 %	0.1 %	**pass**
Fatigue notch factor Kf,t	1.133	1.13403	0.091 %	0.1 %	**pass**
Roughness factor KR,σ	0.857	0.8567805	0.026 %	0.1 %	**pass**
Roughness factor KR,τ	0.917	0.9173623	0.04 %	0.1 %	**pass**
Design factor KWK,b	1.541	1.541482	0.031 %	0.1 %	**pass**
Design factor KWK,t	1.224	1.224112	0.0092 %	0.1 %	**pass**
Component fatigue limit SWK,b	261 MPa	261.3979 MPa	0.15 %	0.3 %	**pass**
Component fatigue limit TWK,t	190 MPa	189.9307 MPa	0.036 %	0.3 %	**pass**
Equivalent mean stress Sm,v	86.6 MPa	86.60254 MPa	0.0029 %	0.1 %	**pass**
Equivalent mean stress Tm,v	50 MPa	50 MPa	0 %	0.1 %	**pass**
Mean stress factor KAK,b	0.89	0.8903092	0.035 %	0.1 %	**pass**
Mean stress factor KAK,t	0.928	0.9285425	0.058 %	0.1 %	**pass**
Component fatigue strength SAK,b	232 MPa	232.7249 MPa	0.31 %	0.5 %	**pass**
Component fatigue strength TAK,t	176 MPa	176.3587 MPa	0.2 %	0.3 %	**pass**
Upper limit SBK,max,b	588 MPa	587.1744 MPa	0.14 %	0.3 %	**pass**
Upper limit TBK,max,t	354 MPa	353.349 MPa	0.18 %	0.3 %	**pass**
Degree of utilisation aBK,b	0.81	0.8056721	0.53 %	1 %	**pass**
Degree of utilisation aBK,t	0.57	0.5670262	0.52 %	1 %	**pass**
Combined degree of utilisation aBK,v	0.99	0.9852037	0.48 %	1 %	**pass**
Plastic notch factor npl,b	1.31	1.307775	0.17 %	0.3 %	**pass**
Plastic notch factor npl,t	1.31	1.307775	0.17 %	0.3 %	**pass**
Static strength SSK,b	880 MPa	879.3565 MPa	0.073 %	0.3 %	**pass**
Static strength TSK,t	508 MPa	507.3887 MPa	0.12 %	0.3 %	**pass**
Static utilisation aSK,b	0.24	0.238811	0.5 %	1 %	**pass**
Static utilisation aSK,t	0.36	0.3586993	0.36 %	1 %	**pass**
Combined static utilisation aSK,v	0.43	0.4309245	0.21 %	1 %	**pass**

The same cases calculated by other programs or methods

Shaft strength: DIN 743 against a published example

Kind of reference	The same cases calculated by other programs or methods
Implements	DIN 743-1/-2:2012
Reference source	A published DIN 743 calculation of a shouldered shaft
Cases	23 of 23 within tolerance

Shoulder D 50 / d 40 mm, r 3 mm, Rz 5 µm, 36CrNiMo4 +QT (σB 1100, σS 900 MPa at 16 mm, K1 = 0.87), stresses σzd 200 ± 50, σb 300 ± 60, τt 100 ± 40 MPa, constant mean stress: a published worked DIN 743 calculation by a calculation program’s maker (DIN 743:2000, with one K1 for both strengths).

Case	Reference	Axomek	Deviation	Tolerance	
Stress concentration α_σ, tension	1.97	1.968407	0.081 %	0.6 %	pass
Stress concentration α_σ, bending	1.8	1.801377	0.076 %	0.6 %	pass
Stress concentration α_τ, torsion	1.4	1.404984	0.36 %	0.6 %	pass
Stress gradient G', bending	0.87 1/mm	0.8736836 1/mm	0.42 %	1 %	pass
Stress gradient G', torsion	0.38 1/mm	0.3833333 1/mm	0.88 %	1.5 %	pass
Notch factor β_σ, tension	1.9	1.902301	0.12 %	1 %	pass
Notch factor β_σ, bending	1.74	1.740881	0.051 %	1 %	pass
Notch factor β_τ, torsion	1.37	1.373371	0.25 %	1 %	pass
Total influence factor K_σ, tension	2.02	2.019055	0.047 %	1 %	pass
Total influence factor K_σ, bending	2.08	2.07667	0.16 %	1 %	pass
Total influence factor K_τ, torsion	1.61	1.610127	0.0079 %	1 %	pass
Fatigue limit σ_zdWK	190 MPa	189.5936 MPa	0.21 %	1 %	pass
Fatigue limit σ_bWK	231 MPa	230.4169 MPa	0.25 %	1 %	pass
Fatigue limit τ_tWK	179 MPa	178.3089 MPa	0.39 %	1 %	pass
Equivalent mean stress σ_mv	529 MPa	529.1503 MPa	0.028 %	0.2 %	pass
Amplitude strength σ_zdADK	132 MPa	131.415 MPa	0.44 %	1.5 %	pass
Amplitude strength σ_bADK	157 MPa	157.9969 MPa	0.63 %	1.5 %	pass
Amplitude strength τ_tADK	147 MPa	146.9242 MPa	0.052 %	1.5 %	pass
Yield limit σ_zdFK	823 MPa	822.15 MPa	0.1 %	0.5 %	pass
Yield limit σ_bFK	988 MPa	986.58 MPa	0.14 %	0.5 %	pass
Yield limit τ_tFK	543 MPa	542.4783 MPa	0.096 %	0.5 %	pass
Fatigue safety S	1.24	1.23838	0.13 %	1 %	pass
Static safety S	1.4	1.394639	0.38 %	1 %	pass

Stated limits

Standards and scope (help: modules/shaft-strength.md)

- **DIN 743-1/-2:2012**, nominal stresses from the shaft solution. Notch factors from DIN 743-2 (formulas for shoulders, grooves and cross bores; measured factors for keyways, press fits, retaining-ring grooves and V-notches), geometric size factor K_2, roughness factor K_F, surface factor K_V from the lower bounds of Table 4, and mean stress cases 1 and 2. Compressive mean stresses follow Annex A of part 1. The static check uses the component yield point.
- **FKM guideline, 7th edition (2020)**, nominal stress approach for non-welded rod-shaped parts: chapter 1 (static) and chapter 2 (fatigue), with the size factor from the FKM material tables, Kf from the stress gradients, K_R, K_V, the mean stress factor, the variable amplitude factor and damage accumulation over a duty cycle.
- The DIN 743 group of `axomek verify` recomputes the three worked examples of DIN 743 Beiblatt 1:2024-07. The FKM group reproduces the guideline's worked examples 6.1 and 6.2.

What is not covered:

- **Material data.** DIN 743-3 material tables are not built in beyond the eight preset steels. The material fatigue limits are those of DIN 743-3:2024 eqs. (1) to (3): 0.5·σ_B in bending, 0.4·σ_B in tension and compression and 0.3·σ_B in torsion. Enter σ_B and σ_S from your material certificate for other steels.
- **DIN 743 life.** DIN 743 here is an endurance-limit check. With a duty cycle the worst load case governs; there is no damage accumulation. When the shaft cannot be solved for a load case (for example an axial load that no locating support takes), the fatigue and static safety checks are left out, since the worst case is not known, and the results are marked as not completely computed. Use FKM with **Load cycles** for finite life and spectra.
- **Local stresses.** Only the nominal stress approach. The local stress approach of FKM and its material-mechanics support number are not used. The shaft's own stress results (and the stress limit) contain no notch effects.
- **FKM factors set to 1.** Coating and residual stress factors are 1, and creep (KTt) is not considered. Cross bores take Kt from DIN 743-2 Bild 12, as FKM section 5.2 has none for round bars. Stainless steel gets no temperature factor above 100 °C.
- **Surface treatment.** K_V is always the lower bound of the tables.
- **Geometry limits.** Cross bores up to 2r/d = 0.3, V-notches for t/d from 0.05 to 0.2.

What is not covered (help: theory/shaft-strength.md)

- Splined and serrated shafts (DIN 743-2, 4.2.2) are not among the notch kinds; its involute teeth's factors rate a pinion's teeth cut into the shaft (see above).
- A hob's run-out has no method of its own in the sources at hand: it is rated as above.
- DIN 743 with a duty cycle takes the worst load case; damage accumulation is offered by the FKM method only.
- In the FKM method, coating and residual stress factors are 1, the local stress approach and the material-mechanics support number are not used, and the creep range of the temperature factors is not covered.

`axomek verify` recomputes the worked examples of DIN 743 Beiblatt 1 and examples 6.1 and 6.2 of the FKM guideline.

Known limits

- A shaft's DIN 743 fatigue check over a steering duty is an endurance check at the duty's worst state, however few times it occurs: a shaft loaded hardest only when parking at the lock (60 000 strokes in user test RR1, SH5 at 1.162 against 1.2) fails it where a finite life could pass. On the safe side; a finite-life shaft rating over the strokes is for a later release (RR1 U-A26).
- A teeth end (`teethEnd`) is rated by DIN 743-2's notch factors of a shaft with involute teeth without a hub (4.2.2) and an eq. (9) shoulder or groove of its shape, not by a published method for a hob's run-out, which no source at hand gives (see "Shaft strength references"); the mesh's own load on the teeth is the rack mesh's or gear pair's tooth-root rating. A worm's thread end takes none (a strength notch instead). The Shaft module's 3D draws a rack pinion's teeth cut into the shaft; a gear pair's pinion cut into its shaft is drawn there at its root diameter as before (the Assembly and Steering views draw its teeth). The 3D views draw a single toothed section's ends; teeth over several sections at the root diameter are rated over all of them but drawn on the one at the mesh.
- A placed key's keyway (see "From 0.3.0") is listed in the shaft's strength results at every point it is assessed, each mesh station along the key and the middle between two, which makes the list long for a shaft with long keys (the Drive shaft example's two keys give some 120 points); its lowest safety is the one the checks take. Between two sampled points the shaft's own weight may lift the moment by up to $q \cdot h^2/32$ (q its weight per length, h an element's length): a strength check with a placed keyway point whose safety that bound could take below its limit (through the lower of the bending amplitude strength and fatigue limit, the yield strength or, for a hard layer, its cracking criterion) is not determined (a warning, never a pass), saying so; on a shaft that does not rotate, where the missed stress is a mean stress whose effect is not bounded, a fatigue check with a placed keyway under the shaft's weight is never determined (take the key off the shaft and enter its keyway as a notch); refine the mesh (`settings.maxElementLength`) or enter a keyway notch where the moment along the key is largest. The exact search between the points comes in a later release.
- Over steering-derived load cases a shaft's FKM damage check (with a number of cycles) is undecided where it passes when the shaft's loads follow the speed (0.3.0): every shaft of a mounted stage (a gear or worm pair, a rack mesh, a drive), whose mesh efficiency is rated at each case's speed and enters a member's torque, and whose belt's pull rises with the square of the speed, and every shaft joined to one by a coupling or a bearing between shafts. The cases' chord split bounds that damage only where the stresses follow the force alone. In belt-drive with SH1's FKM over 800000 cycles and one-way strokes over a 126–130 mm/rev lead, the cases passed 6.681 at 6.681179 where the strokes give 6.680947.
- Over a steering system's duty states (0.3.0), all on the side of saying more or of reporting less, for a later release: a rack buckling span that exists in some of a span's states and not in others (the bar moves under the supports) draws no marginal band over that change; a shaft standing still at the system's `state` takes FKM's checks over a number of cycles at the worst state for all its cycles instead of their Miner sum; a state that gives the ball nut no thrust can leave its rating over the duty refused; the system's own state, kept among the worst-case checks, is rated with its mean and amplitude stresses where the duty's states take theirs fully reversed (a state heavier than every duty point is rated more leniently than they are); a refused duty marks every drive's checks, also those of a drive off the steering's train; a bearing's band for its checks other than life and static safety is drawn from those quantities' bands rather than its own; `atState` does not list bearings off the shafts or planet bearings (rated at their own loads), nor the couplings.

- A parallel key's keyway edge chamfers s1 and s2 (DIN 6892 eqs. (4) and (5)) are 0 unless entered: DIN 6885-1 gives none, the drawing does, and each lowers the depth the key bears by its size. Enter them from the drawing.
- A press fit's yield strengths are sized for steels only (DIN 743-2's K1 at the hub's outside diameter and the shaft's seat diameter): the cast irons and aluminium of the list keep their listed values, and a yield strength entered by hand is taken as the part's. A shaft larger elsewhere than at its seat is weaker than its seat diameter gives (DIN 743-2 takes its largest diameter), which only the shaft strength check takes. The size classes the app holds end at 100 mm for C45 +QT, 160 mm for the alloy steels and 250 mm for S235 and S355 (EN 10025-2:2019, Table 6, p. 27; its class over 250 up to 400 mm is for flat products only, Clause 1, and is not taken for a fit's round parts), and it holds none for E335 and the case-hardening steels' cores: beyond them a passing yield check is not determined until the guaranteed yield strength at that size is entered. The class is taken at the part's finished diameter, not at the size of the bar or forging it is turned from, whose class the steel's guarantee follows and which can be lower (an S355 hub of Ø40 from Ø45 bar: 335 MPa, where Ø40 takes 345; review of 0.3.2): for a part near its yield limit, enter the yield strength of the stock's size. E335's classes (Table 7) wait for an input of the stock's size for the same reason. A fit is not linked to a shaft, so its torque is not compared with the torque its seat passes (a placed key's is): a later release.
- The shaft strength check (DIN 743) takes DIN 743-2's K1 alone, as the standard prescribes where the strength at the part's size is not known (6.2): for an alloy steel inside a size class it can lie above the class's guaranteed minimum (42CrMo4 +QT at Ø50: 748.6 MPa against EN 10083-3's 650 MPa), as in any calculation after DIN 743. Enter the guaranteed strengths ("strengths as given for the size") for a shaft near its limit. Press fits hold it to the class (see "From 0.3.0"). Keys of form AB, C to J and lift-out bevels are not in the app, nor a keyway shorter than its key: the key's length stands for both, where DIN 6892 takes the shorter for l_tr and K_λ. A spline's K_λ is still the estimate of 1 up to a length of one diameter, + 0.4 per further diameter: DIN 6892 is for parallel keys only, and DIN 5466-1's factors are not taken.
- A key's and a spline's yield strength (shaft, hub and key) is the material's value at 16 mm, whatever the part's size, as in 0.2.7. A larger part's can be lower (the release-candidate test's keyhub: an S355 hub of Ø90, 355 → about 314 MPa by DIN 743-2's K1), and the safeties shown are then optimistic. For a part over 16 mm, enter the supplier's guaranteed yield strength at that size. The app's Materials lists hold the steels' 16 mm values and have no field for it, so set it in the file or with `echo '{"op":"updateKey","id":"K1","patch":{"hubMaterial":{"name":"S355 at 90 mm","yieldStrength":314}}}' | axomek apply <file> -` (piped, as PowerShell 5.1 drops the quotes of an argument; `shaftMaterial`, `keyMaterial` likewise), under a name of its own: a material that keeps a listed name is shown in the app with the list's Re, though the file's value is the one rated. Taking it at size, as press fits do, is for a later release.

Rolling bearings

Load ratings, load distribution, rating lives, static safety, friction and thermal speeds of rolling bearings, by the bearing solver ported from the original solver.

Methods and standards

Bearings: rolling element loads from the equilibrium of the inner ring with Hertz contact (load–deflection Q = K·δ^1.5 for balls, slicing with lamina for rollers, the approach of ISO 16281:2025, Annex A, the lamina constant 35948·L^(8/9) (Formula (43)) for steel scaled by the materials, [(ηi^0.9 + ηe^0.9)/(2·ηs^0.9)]^(−10/9) with η = Σ(1 − ν²)/E and steel E = 207 000 MPa, ν = 0.3; a spherical roller bearing's rollers as ISO 16281:2025, A.3.3: each roller's operating contact angle and approach from the curvature centres of the outer and its row's inner raceway, its laminae less the crown's gap (Rp = 0.97·re, B.13), every force through the bearing centre, so that it has no tilt stiffness, an element's reported load the sum of its laminae at its operating contact angle; a self-aligning ball bearing's rows D_pw·tan α apart, both on the one sphere; a ball's force along the line through the groove curvature centres, the inner one moved with the ring exactly and the moment taken about the inner ring's centre with the arm ℜi = D_pw/2 + (ri − D_w/2)·cos α_0); contact stresses by Hertz theory.

The equilibrium is solved by Levenberg–Marquardt from a geometric guess; when that does not converge (a bearing with clearance under a light load, whose solution lies just past the clearance), by continuation in load: first at a multiple of the load, then down to the load in halving steps, each from the last solution.

A bearing on a shaft support that the shaft leaves without load is unloaded: no contact stress and no fatigue (life and static safety unlimited), its ratings and minimum load from a solve at a 100 N reference load.

A bearing on its own has its inner ring held at zero tilt (tilt fixed, the default without a moment load, as ISO 281 rates a bearing), the moment that holds it reported; with tilt free, or on a shaft support, the ring tilts under the moments.

A self-aligning bearing on its own (self-aligning ball, spherical roller) takes no moment load and has its tilt released (held at zero, the moments left unbalanced; the self-aligning ball bearing under the standard rules, the original rules keeping its given row distance free to tilt), and on a shaft support it is coupled radially only, passing the shaft's slope without a moment: its support's moment is not passed on (coupled or not, in every load case and the duty cycle), and the vibration analyses take only its radial tangent stiffness (the 2×2 radial block of the bearing's and housing's compliance inverted), no tilt stiffness.

A ball pair's preload becomes the axial interference at which Z·K·δ^1.5·sin α equals it, α the contact angle the interference turns the balls to.

Bearing stiffness: the 5×5 tangent matrix of the inner ring at the operating point by central finite differences of the element reactions (steps 10^{-5}·D_w and the same arc for tilt, rigid rings, made symmetric; tilts right-handed like the moments, the ball solver's own tilt about z turned round), and in the shaft–bearing coupling the inner ring tilting with the shaft's slope (θz = slope_xy, θy = −slope_xz), and the radial, axial and tilt stiffness from a quarter to twice the operating load.

Cage and element speeds for rolling without slip at the contact centres, balls with outer raceway control from the most loaded ball's inner and outer contact angles (Harris: ωm/ω = (1 − γ·cos αi)/(1 + cos(αi − αo)), spin axis tan β = sin αo/(cos αo + γ)); rib loads of flanged cylindrical roller bearings (the axial load shared by the rollers in proportion to their radial loads; an NUP holds it both ways, an NJ or NF only the way its single rib backs the rollers, the other way being refused) and tapered roller bearings (2·Q·sin ψ on the cone rib, ψ from an apex design), with the roller ends' sliding on the ribs taken 0.15·D_w above the raceway.

Roller profile: logarithmic crowning k·D_w·ln(1/(1 − (2x/L)²)) per raceway (the form of the reference profiles of ISO 16281:2025, Annex B.5 and B.6), its coefficient searched to minimise the peak lamina stress under the operating load (scan of ln k and golden-section refinement).

Grease relubrication interval by the manufacturers' rule t_f = k·(14·10⁶/(n·√d) − 4·d) h (k = 10 radial ball, 5 cylindrical and needle, 1 spherical, tapered and thrust bearings), halved for every 15 K above 70 °C. Minimum load after the bearing makers' rules: radial ball bearings Frm = k_r·(ν·n/1000)^(2/3)·(dm/100)² kN with the operating viscosity, cylindrical roller bearings Frm = k_r·(6 + 4·n/nr)·(dm/100)² kN with the reference speed (estimated 550 000/dm unless given), needle and tapered roller bearings 0.02·C, spherical roller bearings 0.01·C_0, against the radial load and for every bearing of a pair; thrust bearings P ≥ 0.01·C (ball) or 0.02·C (roller); k_r typical of the type unless the catalogue's is given; a bearing held by a preload or an axial load of at least the minimum passes with a warning.

Raceway shoulders of ball bearings (deep groove, angular contact, four-point contact): each loaded ball's Hertz contact ellipse reaches up its groove (radius r = f·D_w) to α + asin(a/r) from the groove bottom, α its operating contact angle and a the semi-major axis, against the shoulder edge at cos θe = 1 − h/r (four-point: √(1 − [((r − D_w/2)/r) sin α_0]²) − h/r, α_0 the free contact angle, with each groove centre (r − D_w/2) sin α_0 axially from the cusp plane and h measured above the cusp) for a shoulder h above the groove bottom (the given heights, else estimated as 0.2·D_w for deep groove and 0.3·D_w for angular and four-point contact ball bearings, fallback proportions without catalogue shoulder data); a reach past the edge is a truncated contact, which ISO 281:2007 and ISO 76:2006 (clause 1 of each) exclude from satisfactory ratings, so with both heights given the check warns, naming the ball, the ring and over a duty cycle the load case of the least margin.

Estimated shoulder heights give information only ("estimated, not a check"), without pass/fail or a contribution to the check counts; enter both drawing heights to enable the check.

Ratings and life: basic dynamic load rating after ISO 281:2007 (Eq. (1)–(19)) with fc from ISO/TR 1281-1:2008, Eq. (15), (20), (25), (35), (39), (43) at the actual groove radii (they reproduce ISO 281, Tables 2, 4, 7 and 10 at the tables' radii, 0.52/0.52·D_w radial, 0.53·D_w self-aligning inner, 0.535·D_w thrust; the usual 0.52/0.53 grooves rate 0.3 % to 9 % below Table 2, tighter ones above it); thrust roller rows combined by ISO 281, Eq. (19).

Static load rating from the element load at the ISO 76:2006 contact stress limit (4200 MPa balls, 4600 MPa self-aligning ball, 4000 MPa rollers) by Hertz at the actual groove radii, with the standards' E = 207 000 MPa and ν = 0.3 for steel bearings (every modulus within 15 % of 207 000 MPa; this gives ISO 76, Table 1 and Eq. (7), (11)); hybrid bearings with their own moduli (ISO 20056, the hybrid bearing standard, was not available).

Fatigue load limit after ISO 281, Annex B.3.2 (1500 MPa point contact, the same elastic constants; rollers by Hertz line contact at 1250 MPa, the mean stress of 1500 MPa peak after ISO/TR 1281-2, 9.2), the self-aligning bearing's outer ring contact 1.6 times higher (B.3.2.1.2).

Basic and modified reference rating life after ISO 16281:2025 from the element loads: ball bearings by 5.2 (Formulae (1)–(16), a multi-row bearing's rows each with its own a_ISO, Formulae (14), (15)), roller bearings by laminae after 5.3 (Formulae (17)–(35), with the edge stress load correction of Annex C.2, Formula (C.4), by each lamina's position) with a local a_ISO for every lamina from its dynamic equivalent load Psk (Formulae (31)–(35), the default) or, with Life factor a_ISO: One for the bearing, one a_ISO for the bearing from Pref = C/L_10r^(3/10), the load at which ISO 281's (C/P)^(10/3) gives L_10r (the reference solver's method; ISO 16281:2025 defines no reference load for roller bearings).

a_ISO after ISO 281, Eq. (31)–(42) (50 for eC·C_u/P > 5, κ = 4 above 4, below κ = 0.1, where the standard gives no value, its value at κ = 0.1 with a warning, and eC from κ = 0.1 as well; a κ that is not positive or not a number gives a_ISO 0, with a warning); a double-row bearing's rows and a system's positions each with the life of their own element loads, combined by Weibull (L = (Σ Lj^−e)^−1/e, e = 10/9 balls, 9/8 rollers, consistent with the ratings' i^0.7 and i^(7/9)), κ = ν/ν1

with ν1 from Eq. (28), (29), eC from Annex A (κ ≤ 4); reliability factor a1 from ISO 281, Table 12 (90 % to 99.95 %; ISO/TR 1281-2, Eq. (12)); static safety s_0 from the contact stress against the limits of ISO 76.

Rings or rolling elements softer than 58 HRC: the dynamic rating times (HRC/58)^3.6, and under the standard rules the static contact stress limit (hence C_0 and s_0) times HV/HV(58 HRC), the Vickers hardness from ISO 18265, which for ball bearings is ISO 3408-5's static hardness factor fH0 = (HV/654)³ on C_0.

Also given: ISO 281's dynamic equivalent load P = X·Fr + Y·Fa with its basic rating life (C/P)^p (p = 3 balls, 10/3 rollers) and modified rating life, and, for comparison only, ISO 76's static equivalent load P0 = X0·Fr + Y0·Fa (Tables 2 and 5, at least Fr; thrust 2.3·Fr·tan α + Fa) with C_0/P0; the static check goes by the contact stress.

The life check of a roller bearing goes by the modified reference rating life of ISO 16281:2025 where its internal geometry is entered, else by the lower of that and ISO 281's modified rating life; a ball bearing's, and one ISO 281 gives no life (four-point contact ball bearings), by the modified reference rating life.

ISO 16281:2025 defines a_ISO for steel bearings only (5.2.5.1): a hybrid bearing's life check that passes only by a_ISO, its basic life below the minimum, is not determined.

Above n·dm = 1 000 000 mm/min the rolling elements' centrifugal forces change the load distribution (ISO 16281:2025, A.4), which the static equilibrium leaves out unless the high-speed ball model is on: a note says so.

ISO 281 rates a bearing of about 45° contact angle differently as a radial or as a thrust bearing, so ISO 281's ratings and lives jump there (ISO 16281:2025, Annex F).

A bearing with rules "original" follows the reference solver the bearing module was ported from instead (thrust ball bearings rated with groove radii of at least 0.525·D_w, C_0 and C_u with the materials' moduli, a linear viscosity, the inner ring always free to tilt with the ball moments taken at the pitch circle, the steel roller constant for every material, a self-aligning ball bearing's given row distance, and the rows' loads summed as one raceway); these fixes apply under both rules: a spherical roller bearing's laminae on the arc with its tilt released, a_ISO below κ = 0.1 at its κ = 0.1 value (eC likewise), and moments refused by a self-aligning bearing on its own.

A four-point contact ball bearing is rated as a double-row angular contact ball bearing (ISO 16281:2025, 5.1: Cr with i = 2, each contact pair a row with its own a_ISO); where its balls run in four-point contact (a radial load) the standard gives no rating, and a passing life check is not determined.

Operating clearance: ISO 286 limit deviations (grade-4 holes K4 to S4 with Δ = IT4 − IT3, also as a bearing's housing tolerance) and ISO 492 normal-class bearing tolerances; on a shaft support the fit takes the shaft's material (modulus, Poisson's ratio, expansion) and bore and the linked housing's material wherever the bearing's mounting leaves them unset; ring expansion and contraction from thick-walled cylinder theory (plane stress) with the moduli and Poisson's ratios of shaft, housing and bearing steel (equal materials: I·d/F for a solid shaft, I·E/D for a thick housing); fit pressure and the hoop stress at the inner ring bore; smoothing 0.8·(R_z ring + R_z seat) with R_z 2.5 µm for the ring when a seat roughness is given (DIN 7190-1); thermal expansion of rings, rolling elements, shaft and housing from 20 °C.

Elastic outer ring (when switched on, in a housing): the housing's radial compliance between 36 sectors round the bore (unit loads spread over each sector, the sectors' mean displacement) and a thin ring's ($c(\Delta) = (1/\pi R)[R^2/2EA + \Sigma_{n\geq 2}\, R^4 \cdot s_n^2 \cdot \cos n\Delta/(EI(n^2-1)^2)]$, sector averaging s_n) are each inverted without the first harmonic, which the shaft solve has as the seat's motion, and their stiffnesses added; the element loads spread onto the sectors move the raceway outwards, which reduces each element's approach (by u, or u·cos α for rollers), repeated (Anderson mixing of the sector displacements) until the two agree to 10^{-9}.

The same elastic ring is used in the shaft–bearing coupling, every load case and the duty cycle.

High-speed ball model (when switched on): every ball's centre balances the inner and outer contact forces (Hertz, separate contact constants) and its centrifugal force m·(dm/2)·ωc², with the cage at ω/2·(1 ∓ D·cos α/dm), so the inner and outer contact angles and loads differ (Jones, Harris); the gyroscopic moment Mg = J·ωR·ωc·sin β (J = m·D²/10, pure-rolling spin, outer raceway control attitude tan β = sin α/(cos α + D/dm) at the nominal angle) is held by friction 2·Mg/D at the outer contact, at most 0.1·Qo, in the sense of the precession torque ωc·x̂ × J·ωb (Harris' signed Mg: the ball spins the other way relative to the cage with the outer ring turning, so the friction reverses).

Stresses, the life and the film of each ring use that ring's loads.

Power loss: bearing friction after Palmgren (M0 from f0·(ν·n)^(2/3)·dm³, M1 = f1·P1·dm, by bearing type and lubrication method) plus a given seal torque; gear mesh losses after ISO/TR 14179-2 (Niemann and Schlenk), P = P_in·μmz·HV with μmz = 0.048·(Fbt/b/(vΣC·ρredC))^0.2·η^−0.05·Ra^0.25 (mineral oil, η = 0.88·ν at the pair's oil temperature, ν on the Walther line (ASTM D341) of the oil's viscosity index (95 unless given), at a load case's temperature in that case for a mounted pair that takes the cases', at 40 °C when none is given, Ra = R_z/6; at the working pitch circle: Fbt = Ftw/cos αwt = Ft/cos αt, vΣC = 2·vtw·sin αwt) and HV from the addendum contact ratios; a mounted wheel that drives loses P1·(1/η − 1), in the pair's own mesh loss as in the power loss.

The overall efficiency is 1 − the total loss over the power into the gear train (the driving members of the pairs and drives whose shaft no other pair or drive drives).

No-load gear losses (churning, windage) and seal losses not entered are not included.

Validation

513 of 513 cases within their tolerance, in 21 groups.

Worked examples of standards and books

Rolling bearing friction: Palmgren's load and idle terms of tapered roller bearings

Kind of reference	Worked examples of standards and books
Implements	Palmgren's friction torque M = M0 + M1: M1 = f1·P1·dm with f1 0.0004 and P1 = 2·Y·Fa of a tapered roller bearing (DIN 732:2010, Table A.1), M0 = 10^{-7}·f0·(ν·n)^(2/3)·dm³ for ν·n ≥ 2000
Reference source	AGMA ISO 14179-1:2004, Annex C, C.2.1.1, bearings A and B of Table C.1 (pp. 22 to 23); eqs. 9, 10 and 26, Table 2
Cases	6 of 6 within tolerance

Single tapered roller bearings at 1780 rpm with the contact angle of their axial load factor (Y = 0.4·cot α: 1.6 and 2.0) and dm = (d + D)/2 as eq. 10 gives it (112.7 and 76.2 mm), in oil of 22.6 mm²/s at 95 °C (ν100 chosen so that the Walther line gives it), at the example's loads: A Fr 1064 N with Fa 3112.5 N from its load case 2a), B Fr 4566 N with Fa 1141.5 N. The idle term's f0 differs by source: the engine's for a tapered roller bearing in an oil bath is 3 (DIN 732, series 302), while the example takes f0 between 4 (rollers clear of the oil) and 8 (submerged) by the oil's height (eq. 27, Table 5), 5.26 and 4.92 for its bearings, so the engine's M0 is 43 % and 39 % lower than the example's. M0 is compared at the example's f0 (the engine's × f0/3), which checks the rest of the term, (ν·n)^(2/3)·dm³. Bearing B's M0 (0.2556 N·m) is printed rounded to 0.256, near the edge of the half digit its tolerance allows.

Case	Reference	Axomek	Deviation	Tolerance	
Bearing A: friction load P1 = 2·Y·Fa	9960 N	9960 N	0 N	0.5 N	**pass**
Bearing A: load torque M1 (eq. 9)	0.449 N·m	0.4489968 N·m	3.2×10^{-6} N·m	0.0005 N·m	**pass**

Case	Reference	Axomek	Deviation	Tolerance	
Bearing A: idle torque M0 at f0 5.26 (eq. 26)	0.884 N·m	0.8838932 N·m	0.0001068078 N·m	0.0005 N·m	pass
Bearing B: friction load P1 = 2·Y·Fa	4566 N	4566 N	0 N	0.5 N	pass
Bearing B: load torque M1 (eq. 9)	0.139 N·m	0.1391717 N·m	0.00017168 N·m	0.0005 N·m	pass
Bearing B: idle torque M0 at f0 4.92 (eq. 26)	0.256 N·m	0.2555292 N·m	0.0004708128 N·m	0.0005 N·m	pass

Standards’ tables and printed values

Bearings: ISO 281 load ratings

Kind of reference	Standards’ tables and printed values
Implements	ISO 281:2007 (basic dynamic load ratings); ISO/TR 1281-1:2008
Reference source	ISO 281:2007 Tables 2, 4, 7 and 10; Annex C.5.2 and C.5.3
Cases	45 of 45 within tolerance

The factor fc of the solver against ISO 281:2007, Tables 2, 4, 7 and 10 (printed to 0.1, tolerance half a digit, 0.12 % on Table 7, whose last digits differ from the equation by up to 0.09). The standard rules take fc from the fc equations of ISO/TR 1281-1:2008 at the actual groove radii, which give the tables at the radii the tables are computed with (Table 1 there): deep groove, double row and angular contact bearings at 0.52/0.52·D_w, self-aligning inner 0.53·D_w, thrust ball 0.535·D_w (the usual 0.52/0.53 radial grooves rate 0.3 % to 9 % below Table 2). Thrust roller bearings with α ≠ 90° (no bearing type has them) from the equation (ISO/TR 1281-1, Eq. (39)). Then the worked examples of ISO 281, Annex C.5.2 (α = 40°, D_w = 7.5 mm, Z = 27, D_w/D_pw = 0.091, Cr = 18 651 N from fc = 51.1 at γ rounded to 0.07, the table's radial conformities) and C.5.3 (α = 60°, thrust conformities, fc = 61.12, C_a = 28 663 N), and the tandem rule of thrust roller bearings (8.1.3.1, Eq. (19): two bearings 2^(7/9)·C_a).

Closest to its tolerance: Table 2 at γ = 0.3 (55.951 against 56.0) and Table 10 at α = 65°, γ = 0.01 (107.051 against 107.1) round to the printed values with less than a tenth of the last digit to spare; the tolerance is half that digit.

Case	Reference	Axomek	Deviation	Tolerance	
Table 2, single row (deep groove), γ = 0.01	29.1	29.06029	0.14 %	0.17 %	pass
Table 2, double row radial contact, γ = 0.01	27.5	27.5308	0.11 %	0.18 %	pass
Table 2, self-aligning, γ = 0.01	9.9	9.861292	0.39 %	0.51 %	pass
Table 2, single row (deep groove), γ = 0.05	46.7	46.68898	0.024 %	0.11 %	pass
Table 2, double row radial contact, γ = 0.05	44.2	44.23166	0.072 %	0.11 %	pass
Table 2, self-aligning, γ = 0.05	17.3	17.30488	0.028 %	0.29 %	pass
Table 2, single row (deep groove), γ = 0.1	55.5	55.54441	0.08 %	0.09 %	pass
Table 2, double row radial contact, γ = 0.1	52.6	52.62102	0.04 %	0.095 %	pass
Table 2, self-aligning, γ = 0.1	23.4	23.38066	0.083 %	0.21 %	pass
Table 2, single row (deep groove), γ = 0.2	59.9	59.9249	0.042 %	0.083 %	pass
Table 2, double row radial contact, γ = 0.2	56.8	56.77096	0.051 %	0.088 %	pass
Table 2, self-aligning, γ = 0.2	33.5	33.50723	0.022 %	0.15 %	pass
Table 2, single row (deep groove), γ = 0.3	56	55.95107	0.087 %	0.089 %	pass
Table 2, double row radial contact, γ = 0.3	53	53.00628	0.012 %	0.094 %	pass
Table 2, self-aligning, γ = 0.3	40.3	40.29005	0.025 %	0.12 %	pass
Table 2, single row (deep groove), γ = 0.4	48.4	48.36797	0.066 %	0.1 %	pass

Case	Reference	Axomek	Deviation	Tolerance	
Table 2, double row radial contact, γ = 0.4	45.8	45.82229	0.049 %	0.11 %	**pass**
Table 2, self-aligning, γ = 0.4	40.4	40.43051	0.076 %	0.12 %	**pass**
Table 4, α = 90°, γ = 0.01	36.7	36.72147	0.059 %	0.14 %	**pass**
Table 4, α = 45°, γ = 0.01	42.1	42.07484	0.06 %	0.12 %	**pass**
Table 4, α = 60°, γ = 0.01	39.2	39.15867	0.11 %	0.13 %	**pass**
Table 4, α = 90°, γ = 0.05	59.5	59.5129	0.022 %	0.084 %	**pass**
Table 4, α = 45°, γ = 0.05	67.3	67.30197	0.0029 %	0.074 %	**pass**
Table 4, α = 60°, γ = 0.05	62.6	62.63732	0.06 %	0.08 %	**pass**
Table 4, α = 90°, γ = 0.1	73.3	73.26897	0.042 %	0.068 %	**pass**
Table 4, α = 45°, γ = 0.1	79.7	79.68172	0.023 %	0.063 %	**pass**
Table 4, α = 60°, γ = 0.1	74.2	74.15904	0.055 %	0.067 %	**pass**
Table 4, α = 90°, γ = 0.2	90.2	90.20468	0.0052 %	0.055 %	**pass**
Table 4, α = 45°, γ = 0.2	85.4	85.43471	0.041 %	0.059 %	**pass**
Table 4, α = 60°, γ = 0.2	79.5	79.5133	0.017 %	0.063 %	**pass**
Table 7, γ = 0.01	52.1	52.09095	0.017 %	0.12 %	**pass**
Table 10, α = 90°, γ = 0.01	105.4	105.3554	0.042 %	0.047 %	**pass**
Table 10, α = 65° (equation), γ = 0.01	107.1	107.0512	0.046 %	0.047 %	**pass**
Table 7, γ = 0.1	84.2	84.18221	0.021 %	0.12 %	**pass**
Table 10, α = 90°, γ = 0.1	175.7	175.7434	0.025 %	0.028 %	**pass**
Table 10, α = 65° (equation), γ = 0.1	171.4	171.3551	0.026 %	0.029 %	**pass**
Table 7, γ = 0.2	88.7	88.65373	0.052 %	0.12 %	**pass**
Table 10, α = 90°, γ = 0.2	205	205.0098	0.0048 %	0.024 %	**pass**
Table 10, α = 65° (equation), γ = 0.2	179.3	179.3314	0.017 %	0.028 %	**pass**
Table 7, γ = 0.3	83.8	83.71081	0.11 %	0.12 %	**pass**
Table 10, α = 90°, γ = 0.3	224.3	224.3396	0.018 %	0.022 %	**pass**
Annex C.5.2: Cr, α = 40°, radial conformities	18651 N	18629.84 N	0.11 %	0.2 %	**pass**
Annex C.5.3: fc, α = 60°, thrust conformities	61.12	61.03115	0.15 %	0.2 %	**pass**
Annex C.5.3: C_a	28663 N	28621.17 N	0.15 %	0.3 %	**pass**
Two thrust roller bearings in tandem: C_a,2/C_a = 2^(7/9)	1.714488	1.714488	1.3×10^{-16}	10^{-9}	**pass**

Bearings: ISO 281 modified rating life

Kind of reference	Standards’ tables and printed values
Implements	ISO 281:2007, Table 12 and Eqs. (31) to (42); ISO 16281:2025
Reference source	ISO 281:2007 Table 12; ISO/TR 1281-2:2008 Eq. (12) and 5.7
Cases	23 of 23 within tolerance

The life modification factor for reliability a1 of ISO 281:2007, Table 12 (90 % to 99.95 %), as used by the reliability setting, against ISO/TR 1281-2:2008, Eq. (12) (three-parameter Weibull, slope 1.5, asymptote 0.05), within the table's rounding. The a_ISO equations (31) to (42) at the borders of their κ ranges, where ISO/TR 1281-2, 5.7 says they meet: rollers within 10^{-5}, balls within 0.2 % (the published constants are rounded). ISO 16281:2025 for a ball bearing without clearance under radial load, which should reduce to ISO 281 (Pr = Fr) within the approximations of the load integrals (2 %),. No worked a_ISO or eC examples are published; the eC equations of Annex A are taken as printed (the diagrams cannot be read from the available copy).

Closest to its tolerance: a1 for 96 % from ISO/TR 1281-2’s equation is 0.5549, which Table 12 prints as 0.55: within half its last digit, near the edge.

Case	Reference	Axomek	Deviation	Tolerance	
a1 for 90 % (L10m)	1	1	0 %	0.5 %	**pass**
a1 for 95 % (L5m)	0.64	0.6379117	0.33 %	0.78 %	**pass**
a1 for 96 % (L4m)	0.55	0.5548951	0.89 %	0.91 %	**pass**
a1 for 97 % (L3m)	0.47	0.4653531	0.99 %	1.1 %	**pass**
a1 for 98 % (L2m)	0.37	0.3658964	1.1 %	1.4 %	**pass**
a1 for 99 % (L1m)	0.25	0.2483317	0.67 %	2 %	**pass**
a1 for 99.2 % (L0.8m)	0.22	0.2208023	0.36 %	2.3 %	**pass**
a1 for 99.4 % (L0.6m)	0.19	0.1908997	0.47 %	2.6 %	**pass**
a1 for 99.6 % (L0.4m)	0.16	0.1574546	1.6 %	3.1 %	**pass**
a1 for 99.8 % (L0.2m)	0.12	0.1176469	2 %	4.2 %	**pass**
a1 for 99.9 % (L0.1m)	0.093	0.09260068	0.43 %	0.54 %	**pass**
a1 for 99.92 % (L0.08m)	0.087	0.0867097	0.33 %	0.57 %	**pass**
a1 for 99.94 % (L0.06m)	0.08	0.08030117	0.38 %	0.63 %	**pass**
a1 for 99.95 % (L0.05m)	0.077	0.07683227	0.22 %	0.65 %	**pass**
a_ISO radial ball continuous at κ = 0.4 (eC·C_u/P = 0.5)	0.768114	0.7686704	0.072 %	0.3 %	**pass**
a_ISO radial ball continuous at κ = 1 (eC·C_u/P = 0.5)	50	50	0 %	0.3 %	**pass**
a_ISO radial roller continuous at κ = 0.4 (eC·C_u/P = 0.5)	0.2312615	0.2312628	5.7×10^{-6}	0.01 %	**pass**
a_ISO radial roller continuous at κ = 1 (eC·C_u/P = 0.5)	1.714109	1.714109	2.2×10^{-9}	0.01 %	**pass**
a_ISO thrust ball continuous at κ = 0.4 (eC·C_u/P = 0.5)	0.3914959	0.3916788	0.047 %	0.3 %	**pass**
a_ISO thrust ball continuous at κ = 1 (eC·C_u/P = 0.5)	5.068126	5.068126	2.7×10^{-9}	0.3 %	**pass**
a_ISO thrust roller continuous at κ = 0.4 (eC·C_u/P = 0.5)	0.1773377	0.1773384	3.8×10^{-6}	0.01 %	**pass**
a_ISO thrust roller continuous at κ = 1 (eC·C_u/P = 0.5)	0.6505069	0.6505069	1.4×10^{-9}	0.01 %	**pass**

Case	Reference	Axomek	Deviation	Tolerance	
ISO 16281:2025, ball bearing without clearance: Pref = Fr	3000 N	3011.836 N	0.39 %	2 %	pass

Bearings: ISO 76 static load ratings

Kind of reference	Standards’ tables and printed values
Implements	ISO 76:2006 (basic static load ratings); ISO/TR 1281-2:2008 Eq. (33)
Reference source	ISO 76:2006 Table 1, Eq. (7) and (11), Annex A.5.2 and A.5.3
Cases	18 of 18 within tolerance

C_0 of the solver from the largest element load at the contact stress limit (4200 MPa, self-aligning ball 4600 MPa, rollers 4000 MPa; C0r = 0.2·i·Z·Q0·cos α, C_0a = Z·Q0·sin α), with the standard's E = 207 000 MPa and ν = 0.3, against ISO 76:2006: f0 of Table 1 (radial ball 0.52/0.53·D_w, self-aligning, thrust ball 0.54·D_w at α = 60° and 90°), printed to 0.1, tolerance half a digit (0.12 for the thrust column, whose 41.9 at 0.25 is 41.95); the roller constants 44 and 220 of Eq. (7) and (11) within 0.5 %; and Annex A.5.2 (α = 40°, D_w = 7.5 mm, Z = 27, D_w/D_pw = 0.091, f0 = 16.1: C0r = 18 731 N) and A.5.3 (α = 60°, f0 = 57.82: C_0a = 76 049 N). Then ISO/TR 1281-2:2008, Eq. (33): C_0/C_u of roller bearings from 4000 and 1250 MPa Hertz line contact and the load zones 0.2 and 0.2453 (printed 8.31; the equation's arithmetic gives 8.35).

Case	Reference	Axomek	Deviation	Tolerance	
Table 1, radial ball, D_w·cos α/D_pw = 0.05	15.7	15.72945	0.19 %	0.32 %	pass
Table 1, self-aligning ball, 0.05	2.1	2.143495	2.1 %	2.4 %	pass
Table 1, thrust ball (α = 60°), 0.05	57.5	57.52167	0.038 %	0.1 %	pass
Table 1, radial ball, D_w·cos α/D_pw = 0.1	16.4	16.39522	0.029 %	0.3 %	pass
Table 1, self-aligning ball, 0.1	2.4	2.352498	2 %	2.1 %	pass
Table 1, thrust ball (α = 60°), 0.1	53.5	53.53207	0.06 %	0.11 %	pass
Table 1, radial ball, D_w·cos α/D_pw = 0.2	14	13.95409	0.33 %	0.36 %	pass
Table 1, self-aligning ball, 0.2	2.8	2.799667	0.012 %	1.8 %	pass
Table 1, thrust ball (α = 60°), 0.2	45.7	45.74364	0.095 %	0.13 %	pass
Table 1, radial ball, D_w·cos α/D_pw = 0.3	11.6	11.61514	0.13 %	0.43 %	pass
Table 1, self-aligning ball, 0.3	3.3	3.285721	0.43 %	1.5 %	pass
Table 1, thrust ball (α = 60°), 0.3	38.2	38.2369	0.097 %	0.16 %	pass
Table 1, thrust ball α = 90° (first row)	61.6	61.57013	0.048 %	0.081 %	pass
Eq. (7): radial roller constant 44	44	44.19477	0.44 %	0.5 %	pass
Eq. (11): thrust roller constant 220 (α = 90°)	220	220.9739	0.44 %	0.5 %	pass
Annex A.5.2: C0r, α = 40°	18731 N	18761.11 N	0.16 %	0.3 %	pass
Annex A.5.3: C_0a, α = 60°	76049 N	76133.01 N	0.11 %	0.3 %	pass
ISO/TR 1281-2, Eq. (33): C_0/C_u of a roller bearing	8.31	8.348961	0.47 %	0.5 %	pass

The standards’ formulae worked by hand

Bearing heat balance: ISO 15312 and DIN 732

Kind of reference	The standards’ formulae worked by hand
Implements	DIN ISO 15312:2004; DIN 732:2010
Reference source	DIN ISO 15312:2004 and DIN 732:2010 formulae, worked by hand
Cases	9 of 9 within tolerance

Heat-emitting reference surface A_r = π·B·(D + d) of radial bearings (DIN ISO 15312:2004, Eq. (1)) and π/2·(D² − d²) of thrust bearings (the two washer seat faces of definition 3.3; the 2004 edition prints Eq. (3) with D² + d²); reference heat flow density q_r at 50 K (5.3.2) for radial bearings 0.016 W/mm² up to 50 000 mm² and 0.016·(A_r/50 000)^−0.34 above, for axial bearings 0.020 W/mm² and 0.020·(A_r/50 000)^−0.16; oil 28.5 W per l/min and K (DIN 732:2010, Eq. (12), from 1.88 kJ/(kg·K) and 0.91 kg/dm³ of mineral oil, Eq. (10), (11)). Neither standard has a worked example. The balance itself against a linear friction law with a closed-form solution.

Case	Reference	Axomek	Deviation	Tolerance	
6208 reference surface π·B·(D + d)	6785.84 mm²	6785.84 mm²	0 %	10^{-12}	**pass**
Radial, A_r ≤ 50 000 mm²: q_r	0.016 W/mm²	0.016 W/mm²	2.2×10^{-16}	10^{-9}	**pass**
Radial, A_r = 100 000 mm²: q_r = 0.016·2^−0.34	0.01264066 W/mm²	0.01264066 W/mm²	0 %	10^{-9}	**pass**
Axial, A_r ≤ 50 000 mm²: q_r	0.02 W/mm²	0.02 W/mm²	0 %	10^{-9}	**pass**
Axial, A_r = 100 000 mm²: q_r = 0.020·2^−0.16	0.0179005 W/mm²	0.0179005 W/mm²	1.9×10^{-16}	10^{-9}	**pass**
51107 reference surface π/2·(D² − d²)	2323.208 mm²	2323.208 mm²	0 %	10^{-12}	**pass**
Oil heat per l/min and K in the balance (DIN 732, Eq. (12))	28.5 W/K	28.5 W/K	0 %	10^{-9}	**pass**
DIN 732, Eq. (9)–(11): c·ρ/60 = 1.88·0.91·1000/60	28.51333 W/K	28.5 W/K	0.047 %	0.18 %	**pass**
Balance temperature, linear friction	49.34579 °C	49.34579 °C	0 %	10^{-6}	**pass**

Bearing thermal speeds: DIN ISO 15312 and DIN 732

Kind of reference	The standards’ formulae worked by hand
Implements	DIN ISO 15312:2004; DIN 732:2010
Reference source	DIN ISO 15312:2004 and DIN 732:2010 formulae, worked by hand
Cases	16 of 16 within tolerance

Thermal reference speed n_θr from the heat balance of DIN ISO 15312:2004, Eq. (11), under its reference conditions (70 °C outer ring in 20 °C, P1r = 0.05·C_0, νr = 12 mm²/s, f0r and f1r of Table A.1, Φr = qr·Ar), and thermally permissible operating speed nθ of DIN 732:2010, Eq. (1)–(8), (12), with fn from KL·fn^(5/3) + KP·fn = 1 and the approximation of Eq. (3). Neither standard has a worked example: the values are hand solutions of the equations (bisection), the factors read from the tables.

Case	Reference	Axomek	Deviation	Tolerance	
6208: reference heat flow Φr = 0.016·π·18·120	108.5734 W	108.5734 W	0 %	10^{-12}	**pass**

Case	Reference	Axomek	Deviation	Tolerance	
6208: thermal reference speed n_θr (hand: 9 297.2 rpm)	9297.2 rpm	9297.245 rpm	4.8×10^{-6}	0.001 %	pass
6208 run at the reference conditions: nθ (hand: 9 293.5 rpm, f1 = 0.0009·√0.05)	9293.5 rpm	9293.541 rpm	4.5×10^{-6}	0.001 %	pass
NU 2208: f0 of series 22 (Table A.1)	3	3	0 %	10^{-12}	pass
NU 2208: n_θr (hand: 7 035.36 rpm)	7035.36 rpm	7035.363 rpm	4.0×10^{-7}	0.001 %	pass
NU 2208: heat removed Φ = 165 + 570	735 W	735 W	0 %	10^{-12}	pass
NU 2208: KL (Eq. (5), hand: 0.17572)	0.17572	0.175717	0.0017 %	0.01 %	pass
NU 2208: KP (Eq. (6), hand: 0.24057)	0.24057	0.2405686	5.6×10^{-6}	0.01 %	pass
NU 2208: nθ (hand: 13 679.3 rpm)	13679.3 rpm	13679.35 rpm	3.6×10^{-6}	0.001 %	pass
NU 2208: nθ = fn·n_θr (Eq. (4))	13679.35 rpm	13679.35 rpm	2.1×10^{-13}	10^{-9}	pass
NU 2208: fn by the approximation Eq. (3) (hand: 1.93848)	1.93848	1.938476	2.1×10^{-6}	0.001 %	pass
Eq. (2) without the viscous term: fn = 1/KP	2	2	3.6×10^{-15}	10^{-12}	pass
Eq. (2) at KL = KP = 0.5: fn = 1	1	1	3.6×10^{-15}	10^{-12}	pass
Eq. (3) at KL = KP = 0.5 against the exact fn = 1	1	0.9914218	0.86 %	3 %	pass
Spherical roller 23: f0r 4.5 (Table A.1)	4.5	4.5	0 %	10^{-12}	pass
Spherical roller 23: f1r 0.0003 (Table A.1)	0.0003	0.0003	0 %	10^{-12}	pass

Internal geometry worked back from a catalogue's C and C_0

Kind of reference: The standards’ formulae worked by hand

Implements: ISO 281:2007, Eq. (1), (7), (13), (17) with ISO/TR 1281-1's fc; ISO 76:2006, 5.1, 6.1

Reference source: Catalogue-style ratings made from a known geometry (each type's preset with its ball or roller diameter 4 % larger and one element more, a roller 5 % shorter): the geometry fitted to them rates them back

Cases: 26 of 26 within tolerance

C to 0.1 %; C_0 to 3 % for ball bearings (whole ball counts) and 0.5 % for roller bearings.

Case	Reference	Axomek	Deviation	Tolerance	
6207: C	28970.52 N	28970.5 N	8.1×10^{-7}	0.1 %	pass
6207: C_0	18121.09 N	18121.08 N	7.5×10^{-7}	3 %	pass
4207: C	37919.77 N	37919.8 N	9.2×10^{-7}	0.1 %	pass
4207: C_0	33552.52 N	33552.55 N	8.3×10^{-7}	3 %	pass
7207b: C	29765.92 N	29765.9 N	5.0×10^{-7}	0.1 %	pass
7207b: C_0	21171.27 N	20559.92 N	2.9 %	3 %	pass
3207: C	38918.47 N	38918.5 N	8.4×10^{-7}	0.1 %	pass
3207: C_0	34771.39 N	34771.41 N	7.6×10^{-7}	3 %	pass
1207: C	21537.93 N	21537.9 N	1.3×10^{-6}	0.1 %	pass
1207: C_0	6928.944 N	6969.023 N	0.58 %	3 %	pass

Case	Reference	Axomek	Deviation	Tolerance	
qj207: C	44647.17 N	44647.2 N	6.0×10^{-7}	0.1 %	pass
qj207: C_0	34711.65 N	34488.67 N	0.64 %	3 %	pass
51107: C	22645.77 N	22645.8 N	1.5×10^{-6}	0.1 %	pass
51107: C_0	54620.76 N	54279.76 N	0.62 %	3 %	pass
nu207: C	53017.46 N	53017.5 N	7.1×10^{-7}	0.1 %	pass
nu207: C_0	52882.48 N	52882.5 N	3.4×10^{-7}	0.5 %	pass
nnu4920: C	188334 N	188334 N	10^{-7}	0.1 %	pass
nnu4920: C_0	323282.4 N	323282.4 N	2.3×10^{-8}	0.5 %	pass
na4905: C	22360.82 N	22360.8 N	7.5×10^{-7}	0.1 %	pass
na4905: C_0	33922.03 N	33922 N	7.7×10^{-7}	0.5 %	pass
81107: C	32849.4 N	32849.4 N	5.6×10^{-8}	0.1 %	pass
81107: C_0	103703 N	103703 N	3.1×10^{-7}	0.5 %	pass
30207: C	53505.46 N	53505.5 N	7.5×10^{-7}	0.1 %	pass
30207: C_0	62163.77 N	62163.8 N	5.7×10^{-7}	0.5 %	pass
22208: C	92941.2 N	92941.2 N	3.5×10^{-8}	0.1 %	pass
22208: C_0	109385.2 N	109385.2 N	1.9×10^{-9}	0.5 %	pass

Rings or rolling elements softer than 58 HRC: each raceway contact's capacity

Kind of reference	The standards’ formulae worked by hand
Implements	ISO 16281:2025, Formulae (1)–(4), (9); the hardness factor (HRC/58)^3.6, a common convention outside ISO 281; ISO 281:2007's f_c split of Q_ci and Q_ce (ISO/TR 1281-1)
Reference source	Formula (9) worked by hand from the hardened bearing's element loads, each contact's Q_c times the factor of the softer of its ring and the rolling elements; the f_c split worked by hand (audit RR1-S-02's deep groove ball bearing, 3 kN, 45 HRC parts)
Cases	23 of 23 within tolerance

The standard rules; the original rules keep the inner ring's factor alone.

Case	Reference	Axomek	Deviation	Tolerance	
C as given	30000	30000	0 %	10^{-12}	pass
given C, soft outer ring	19818.5	19818.5	3.7×10^{-16}	10^{-9}	pass
soft inner ring: Q_ci	2890.099	2890.099	0 %	10^{-12}	pass
soft inner ring: Q_ce	12716.21	12716.21	0 %	10^{-12}	pass
L10r, soft inner ring	434.8694	434.8694	1.2×10^{-15}	10^{-9}	pass
C, soft inner ring	10560.12	10560.12	1.7×10^{-16}	10^{-9}	pass
C at 60 HRC	25148.05	25148.05	1.7×10^{-11}	10^{-9}	pass
L10r at 60 HRC	5868.113	5868.113	6.0×10^{-11}	10^{-9}	pass
soft outer ring: Q_ci	7205.902	7205.902	0 %	10^{-12}	pass
soft outer ring: Q_ce	5100.138	5100.138	0 %	10^{-12}	pass

Case	Reference	Axomek	Deviation	Tolerance	
L10r, soft outer ring	1685.342	1685.342	4.0×10^{-16}	10^{-9}	pass
C, soft outer ring	16613.23	16613.23	2.2×10^{-16}	10^{-9}	pass
soft balls: Q_ci	2890.099	2890.099	0 %	10^{-12}	pass
soft balls: Q_ce	5100.138	5100.138	1.8×10^{-16}	10^{-12}	pass
soft balls: C	10086.23	10086.23	0 %	10^{-12}	pass
L10r, soft balls	378.5919	378.5919	1.5×10^{-16}	10^{-9}	pass
soft balls: all soft	378.5919	378.5919	0 %	10^{-12}	pass
original rules, HRC 60/45/60: C	25148.05	25148.05	0 %	10^{-12}	pass
original rules, HRC 60/45/60: L10r	5868.113	5868.113	0 %	10^{-9}	pass
original rules, HRC 60/60/45: C	25148.05	25148.05	0 %	10^{-12}	pass
original rules, HRC 60/60/45: L10r	5868.113	5868.113	0 %	10^{-9}	pass
original rules, HRC 45/60/60: C	10086.23	10086.23	0 %	10^{-12}	pass
original rules, HRC 45/60/60: L10r	378.5919	378.5919	1.5×10^{-16}	10^{-9}	pass

Roller edge stress load correction by lamina position

Kind of reference	The standards’ formulae worked by hand
Implements	ISO 16281:2025, 5.3.4, Annex C.2, Formula (C.4); 5.3.6, Formula (29)
Reference source	ISO 16281:2025, Formula (C.4) worked by hand; straight rollers under a radial load, whose laminae carry equal loads, so L_10r goes as the mean of f^(9/2) to the power −8/9
Cases	3 of 3 within tolerance

The life ratio of 41 to 141 laminae follows from the edge factors alone (closed form of Formula (29) for equal lamina loads).

Case	Reference	Axomek	Deviation	Tolerance	
f at |xk|/L_we = 0.25	1.014272	1.014272	5.7058×10^{-8}	1×10^{-6}	pass
f on the end lamina of 41	1.3104	1.310358	4.2235×10^{-5}	0.0001	pass
Straight rollers: L_10r of 41 over 141 laminae	1.063223	1.063229	5.9×10^{-6}	0.001 %	pass

Four-point contact ball bearing rated as double-row angular contact

Kind of reference	The standards’ formulae worked by hand
Implements	ISO 16281:2025, 5.1 (p. 6), 5.2.1.2 Formulae (1), (2) (p. 7), 5.2.3 Formula (10) (p. 9), 5.2.5.3 Formulae (14), (15) (pp. 9–10); ISO 281:2007, 5.1.1 Eq. (1), Table 2
Reference source	ISO 281:2007, Eq. (1) with i = 2 and ISO/TR 1281-1's fc at the actual groove radii (λ 0.95); ISO 16281:2025's Qci, Qce and lives worked from the solver's element loads
Cases	5 of 5 within tolerance

Under predominantly axial load, each ball in two-point contact (the approximation's condition).

Case	Reference	Axomek	Deviation	Tolerance	
Cr (i = 2)	49776.81 N	49776.81 N	0 %	10^{-9}	pass

Case	Reference	Axomek	Deviation	Tolerance	
Qci	7480.903 N	7480.903 N	0 %	10^{-9}	**pass**
Qce	12548.32 N	12548.32 N	0 %	10^{-9}	**pass**
L_10r, 5 kN axial	1348.634 10^6 rev	1348.634 10^6 rev	0 %	10^{-9}	**pass**
L_10mr, 5 kN axial	2689.803 10^6 rev	2689.803 10^6 rev	0 %	10^{-9}	**pass**

Multi-row ball bearing: reference rating lives with each row's a_ISO

Kind of reference: The standards’ formulae worked by hand

Implements: ISO 16281:2025, 5.2.2 to 5.2.5, Formulae (5), (7), (9), (10), (14), (15)

Reference source: ISO 16281:2025, the formulae worked independently from the solver's element loads

Cases: 3 of 3 within tolerance

Case	Reference	Axomek	Deviation	Tolerance	
Double-row angular contact, mostly axial: L_10r	956.4551 10^6 rev	956.4551 10^6 rev	0 %	10^{-9}	**pass**
Double-row angular contact, mostly axial: L_10mr	1009.398 10^6 rev	1009.398 10^6 rev	0 %	10^{-9}	**pass**
Double-row deep groove, rows equally loaded: a_ISO	1.231775	1.231775	0 %	10^{-9}	**pass**

Roller bearing: reference rating lives with each lamina's a_ISO per row

Kind of reference: The standards’ formulae worked by hand

Implements: ISO 16281:2025, 5.3.3 to 5.3.8, Formulae (23)–(25), (27), (30), (31), (34)

Reference source: ISO 16281:2025, the formulae worked independently from the solver's edge-corrected lamina loads

Cases: 4 of 4 within tolerance

Case	Reference	Axomek	Deviation	Tolerance	
Spherical roller, radial and axial: L_10r	395.5641 10^6 rev	395.5641 10^6 rev	0 %	10^{-9}	**pass**
Spherical roller, radial and axial: L_10mr	164.2513 10^6 rev	164.2513 10^6 rev	0 %	10^{-9}	**pass**
Double-row cylindrical roller, radial: L_10r	273.0559 10^6 rev	273.0559 10^6 rev	0 %	10^{-9}	**pass**
Double-row cylindrical roller, radial: L_10mr	312.1268 10^6 rev	312.1268 10^6 rev	0 %	10^{-9}	**pass**

Spherical roller bearing: load distribution of A.3.3

Kind of reference	The standards’ formulae worked by hand
Implements	ISO 16281:2025, A.3.3, Formulae (A.19), (A.20); 6.3.4, Formulae (51)–(54); Annex B.7, Formulae (B.11)–(B.13)
Reference source	ISO 16281:2025, pp. 18, 26–27, 29: the formulae solved independently for a pure axial load (one row loaded, every roller alike): the inner raceway's curvature centre moved axially by δa from A = 2·re − Dwe, the operating contact angle and approach from it, each lamina less the crown gap of Rp = 0.97·re against re, the rollers' axial forces balancing Fa
Cases	2 of 2 within tolerance

Without the edge stress correction, so that the lamina loads are the equilibrium's.

Case	Reference	Axomek	Deviation	Tolerance	
Roller load, 6 kN axial	2042.704 N	2042.704 N	1.3×10^{-12}	0.01 %	**pass**
Operating contact angle αj, 6 kN axial	10.57837 °	10.57837 °	10^{-15}	0.01 %	**pass**

Tapered roller reference profile

Kind of reference	The standards’ formulae worked by hand
Implements	ISO 16281:2025, Annex B.6, Formula (B.10)
Reference source	ISO 16281:2025, Formula (B.10) with the half cone angle β = 0: the logarithmic profile of coefficient 0.000 45
Cases	2 of 2 within tolerance

The reference profile and an entered crowning k = 0.000 45 give the same solve.

Case	Reference	Axomek	Deviation	Tolerance	
L_10r, reference profile against k = 0.000 45	834.697 10^6 rev	834.697 10^6 rev	0 %	exact	**pass**
Largest contact stress, reference profile against k = 0.000 45	1633.287 MPa	1633.287 MPa	0 %	exact	**pass**

Life modification factor a_ISO past e_C·C_u/P = 5

Kind of reference	The standards’ formulae worked by hand
Implements	ISO 281:2007, 9.3.3.4, Eq. (32)
Reference source	Eq. (32) worked by hand at e_C·C_u/P = 5 (audit RR1-O-02's lightly loaded deep groove ball bearing in thin hot oil)
Cases	3 of 3 within tolerance

e_C·C_u/P above 5 is taken as 5, where ISO 281's curves end (9.3.3.4 limits a_ISO to 50, also above 5; holding the ratio is the app's reading); a_ISO is continuous across it.

Case	Reference	Axomek	Deviation	Tolerance	
a_ISO held at 5	36.15161	36.15161	0 %	10^{-9}	**pass**
a_ISO either side of e_C·C_u/P = 5	36.15161	36.15161	1.8×10^{-12}	10^{-6}	**pass**
bounded factor	36.15161	36.15161	0 %	10^{-12}	**pass**

Closed-form solutions

One-way bearings pushed against their facing

Kind of reference	Closed-form solutions
Implements	—
Reference source	Closed form: the check's value is the axial load against the facing, F_a, and its limit 0.05·F_r·tan α (a 7207 B, α = 40°, at F_r 3000 N)
Cases	2 of 2 within tolerance

The 5 % share is Axomek's modelling rule, not a standard's: a single angular contact ball or tapered roller bearing, or a tandem pair, takes no axial load against its facing, which the solver's two-sided raceway would carry.

Case	Reference	Axomek	Deviation	Tolerance	
7207 B, F_r 3000 N, F_a 3000 N towards −x: axial load against the facing	3000 N	3000 N	0 %	10^{-12}	**pass**
7207 B, F_r 3000 N: limit 0.05·F_r·tan 40°	125.8649 N	125.8649 N	1.9×10^{-10}	10^{-8}	**pass**

Spring preload of a floating ball bearing: the bearings' axial loads

Kind of reference	Closed-form solutions
Implements	—
Reference source	Closed form: a spring of force P pressing the floating bearing towards the locating one loads the floating bearing with P and, through the shaft, the locating bearing with P plus the shaft's other axial loads (axial equilibrium ΣF_x = 0)
Cases	8 of 8 within tolerance

User test 4's 30 kW motor rotor (6312 locating, 6212 floating), no other axial load, then a 400 N axial load on the shaft as well.

Case	Reference	Axomek	Deviation	Tolerance	
floating bearing, rigid supports	300 N	300 N	0 %	10^{-9}	**pass**
locating bearing, rigid supports	300 N	300 N	0 %	10^{-9}	**pass**
floating bearing with 400 N, rigid supports	300 N	300 N	0 %	10^{-9}	**pass**
locating bearing with 400 N, rigid supports	100 N	100 N	0 %	10^{-9}	**pass**
floating bearing, coupled	300 N	300 N	0 %	10^{-9}	**pass**
locating bearing, coupled	300 N	300 N	0 %	10^{-9}	**pass**
floating bearing with 400 N, coupled	300 N	300 N	0 %	10^{-9}	**pass**
locating bearing with 400 N, coupled	100 N	100 N	0 %	10^{-9}	**pass**

The same cases calculated by other programs or methods

Bearings: an independent ISO/TS 16281 calculation

Kind of reference	The same cases calculated by other programs or methods
Implements	ISO/TS 16281:2008
Reference source	Results of an independent ISO/TS 16281 calculation program (run 2026-10-10)
Cases	17 of 17 within tolerance

The standard rules against the results of an independent ISO/TS 16281:2008 calculation tool (run 2026-10-10); these are that tool's results, not a standard's. Deep groove ball bearings 6207 (D_pw 53.5, D_w 11.11 mm, Z 9) and 6205 (D_pw 38.5, D_w 7.94 mm, Z 9), grooves 0.52/0.53·D_w, steel E = 207 000 MPa, ν = 0.3, VG 68 (ν_40 = 68, ν100 = 8.6 mm²/s, Walther) at 70 °C, 1500 rpm, Fr = 5 kN with Fa = 0 or 1 kN on the inner ring, diametral clearance Pd 0 or 0.015 mm, and the tool's default bearing (D_pw 35, D_w 8.5 mm, Z 7; Fr 2 kN, Fa 1 kN). With the inner ring held at zero tilt (the standard rules' default for a bearing on its own) Cr, L_10r and the moment holding the ring within 0.1 %. Free to tilt: the tilt within 1.5 % (ours 1.0 % to 1.3 % less; the moment arms are the inner groove centres' radius, not turned with the tilt), p_max within 0.2 % and L_10r within 1 %.

Case	Reference	Axomek	Deviation	Tolerance	
6207: Cr	25.189 kN	25.18859 kN	0.0016 %	0.1 %	**pass**
6207: L_10r, tilt held, Pd 0 mm, Fa 0 kN	126.34 10^6 rev	126.3387 10^6 rev	10.0×10^{-6}	0.1 %	**pass**
6207: L_10r, tilt held, Pd 0.015 mm, Fa 0 kN	117.04 10^6 rev	117.0411 10^6 rev	9.6×10^{-6}	0.1 %	**pass**
6207: L_10r, tilt held, Pd 0 mm, Fa 1 kN	131.9 10^6 rev	131.8995 10^6 rev	3.8×10^{-6}	0.1 %	**pass**
6207: L_10r, tilt held, Pd 0.015 mm, Fa 1 kN	122.35 10^6 rev	122.3521 10^6 rev	0.0017 %	0.1 %	**pass**
6205: Cr	13.76 kN	13.76019 kN	0.0014 %	0.1 %	**pass**
6205: L_10r, tilt held, Pd 0 mm, Fa 0 kN	20.6 10^6 rev	20.59703 10^6 rev	0.014 %	0.1 %	**pass**
6205: L_10r, tilt held, Pd 0.015 mm, Fa 0 kN	19.21 10^6 rev	19.21274 10^6 rev	0.014 %	0.1 %	**pass**
6205: L_10r, tilt held, Pd 0 mm, Fa 1 kN	21 10^6 rev	21.00487 10^6 rev	0.023 %	0.1 %	**pass**
6205: L_10r, tilt held, Pd 0.015 mm, Fa 1 kN	19.56 10^6 rev	19.56447 10^6 rev	0.023 %	0.1 %	**pass**
Default bearing (D_pw 35, D_w 8.5, Z 7): Cr	13.024 kN	13.0237 kN	0.0023 %	0.1 %	**pass**
6207: M_z holding the ring at zero tilt, Pd 0, Fa 1 kN	-20.64 N·m	-20.64483 N·m	0.023 %	0.1 %	**pass**
6207: tilt free, Pd 0 mm, Fa 1 kN, M_z 0 N·m	4.114 mrad	4.061893 mrad	1.3 %	1.5 %	**pass**
6207: tilt free, Pd 0.015 mm, Fa 1 kN, M_z 0 N·m	4.715 mrad	4.657203 mrad	1.2 %	1.5 %	**pass**
6207: tilt free, Pd 0.015 mm, Fa 0 kN, M_z 20 N·m	6.277 mrad	6.215718 mrad	0.98 %	1.5 %	**pass**
6207: p_max, tilt free, Pd 0, Fa 1 kN	2850.05 MPa	2846.271 MPa	0.13 %	0.2 %	**pass**
6207: L_10r, tilt free, Pd 0, Fa 1 kN	89.1 10^6 rev	89.68623 10^6 rev	0.66 %	1 %	**pass**

Bearings: the reference bearing solver

Kind of reference	The same cases calculated by other programs or methods
Implements	ISO 281:2007; ISO 76:2006; ISO/TS 16281:2008
Reference source	The original bearing solver this engine was ported from, on its reference set of cases
Cases	291 of 291 within tolerance

Every case of the reference set run through both solvers (cases the original solver rejects are left out). Shown per case: the largest relative deviation over dynamic and static rating, rolling element loads, maximum contact stress, displacement and rating lives (element loads and displacement relative to their largest value); cases pass within $2 \cdot 10^{-5}$.

Case	Reference	Axomek	Deviation	Tolerance	
type1-base: modified life (worst of 21 values)	167487.2 h	167487.2 h	4.1×10^{-12}	0.002 %	**pass**
type1-loads-0: modified life (worst of 21 values)	4.5994×10^{7} h	4.5994×10^{7} h	3.6×10^{-9}	0.002 %	**pass**
type1-loads-1: modified life (worst of 21 values)	1334.892 h	1334.892 h	4.0×10^{-12}	0.002 %	**pass**
type1-loads-2: modified life (worst of 21 values)	151988.1 h	151988.1 h	10.0×10^{-13}	0.002 %	**pass**
type1-loads-3: modified life (worst of 21 values)	458248.1 h	458248.1 h	6.1×10^{-13}	0.002 %	**pass**
type1-loads-4: modified life (worst of 21 values)	426651.6 h	426651.6 h	8.5×10^{-11}	0.002 %	**pass**
type1-loads-5: modified life (worst of 21 values)	163998 h	163998 h	9.9×10^{-12}	0.002 %	**pass**
type1-loads-6: modified life (worst of 21 values)	111756.9 h	111756.9 h	1.5×10^{-11}	0.002 %	**pass**
type1-loads-7: modified life (worst of 21 values)	32199.83 h	32199.83 h	1.3×10^{-11}	0.002 %	**pass**
type1-loads-8: modified life (worst of 21 values)	122549.4 h	122549.4 h	8.8×10^{-10}	0.002 %	**pass**
type1-operating-0: modified life (worst of 21 values)	437891.9 h	437891.9 h	4.1×10^{-12}	0.002 %	**pass**
type1-operating-1: modified life (worst of 21 values)	14173.15 h	14173.15 h	4.1×10^{-12}	0.002 %	**pass**
type1-operating-2: modified life (worst of 21 values)	3006.469 h	3006.469 h	4.1×10^{-12}	0.002 %	**pass**
type1-operating-3: modified life (worst of 21 values)	8301.402 h	8301.402 h	4.1×10^{-12}	0.002 %	**pass**
type1-clearance-0: modified life (worst of 21 values)	210947.9 h	210947.9 h	4.4×10^{-15}	0.002 %	**pass**
type1-clearance-1: modified life (worst of 21 values)	196117.5 h	196117.5 h	2.6×10^{-11}	0.002 %	**pass**
type1-clearance-2: modified life (worst of 21 values)	119532.1 h	119532.1 h	1.9×10^{-13}	0.002 %	**pass**
type1-offset-0: modified life (worst of 21 values)	156376.6 h	156376.6 h	2.5×10^{-11}	0.002 %	**pass**

Case	Reference	Axomek	Deviation	Tolerance	
type1-offset-1: modified life (worst of 21 values)	153107.6 h	153107.6 h	6.6×10^{-12}	0.002 %	**pass**
type1-material: modified life (worst of 21 values)	33485.45 h	33485.45 h	1.3×10^{-14}	0.002 %	**pass**
type1-random-0: modified life (worst of 20 values)	49055.27 h	49055.27 h	2.0×10^{-12}	0.002 %	**pass**
type1-random-1: modified life (worst of 22 values)	26709.34 h	26709.34 h	1.3×10^{-10}	0.002 %	**pass**
type1-random-2: modified life (worst of 22 values)	18180.55 h	18180.55 h	4.7×10^{-11}	0.002 %	**pass**
type1-random-3: modified life (worst of 22 values)	12470.12 h	12470.12 h	8.6×10^{-12}	0.002 %	**pass**
type1-random-5: modified life (worst of 19 values)	3358.9 h	3358.9 h	2.2×10^{-11}	0.002 %	**pass**
type2-base: modified life (worst of 30 values)	920866.1 h	920866.1 h	4.1×10^{-10}	0.002 %	**pass**
type2-loads-0: modified life (worst of 30 values)	1.1837×10^{8} h	1.1837×10^{8} h	2.1×10^{-12}	0.002 %	**pass**
type2-loads-1: modified life (worst of 30 values)	15028.35 h	15028.35 h	5.1×10^{-11}	0.002 %	**pass**
type2-loads-2: modified life (worst of 30 values)	835919.4 h	835919.4 h	2.8×10^{-10}	0.002 %	**pass**
type2-loads-3: modified life (worst of 30 values)	1.5422×10^{6} h	1.5422×10^{6} h	3.4×10^{-10}	0.002 %	**pass**
type2-loads-4: modified life (worst of 30 values)	1.4546×10^{6} h	1.4546×10^{6} h	3.5×10^{-12}	0.002 %	**pass**
type2-loads-5: modified life (worst of 30 values)	794591.3 h	794591.3 h	2.3×10^{-11}	0.002 %	**pass**
type2-loads-6: modified life (worst of 30 values)	607692.8 h	607692.8 h	2.0×10^{-10}	0.002 %	**pass**
type2-loads-7: modified life (worst of 30 values)	159689.9 h	159689.9 h	3.7×10^{-13}	0.002 %	**pass**
type2-loads-8: modified life (worst of 30 values)	376093.3 h	376093.3 h	1.5×10^{-9}	0.002 %	**pass**
type2-operating-0: modified life (worst of 30 values)	4.6043×10^{6} h	4.6043×10^{6} h	4.1×10^{-10}	0.002 %	**pass**
type2-operating-1: modified life (worst of 30 values)	111764.3 h	111764.3 h	6.1×10^{-10}	0.002 %	**pass**
type2-operating-2: modified life (worst of 30 values)	18976.06 h	18976.06 h	5.7×10^{-10}	0.002 %	**pass**
type2-operating-3: modified life (worst of 30 values)	114256.1 h	114256.1 h	7.5×10^{-10}	0.002 %	**pass**
type2-clearance-0: modified life (worst of 30 values)	1.1108×10^{6} h	1.1108×10^{6} h	5.2×10^{-14}	0.002 %	**pass**
type2-clearance-1: modified life (worst of 30 values)	1.0467×10^{6} h	1.0467×10^{6} h	5.0×10^{-10}	0.002 %	**pass**

Case	Reference	Axomek	Deviation	Tolerance	
type2-clearance-2: modified life (worst of 30 values)	663326.6 h	663326.6 h	3.1×10^{-9}	0.002 %	**pass**
type2-offset-0: modified life (worst of 30 values)	862275.8 h	862275.8 h	7.1×10^{-11}	0.002 %	**pass**
type2-offset-1: modified life (worst of 30 values)	842116.9 h	842116.9 h	8.9×10^{-10}	0.002 %	**pass**
type2-material: modified life (worst of 30 values)	414708 h	414708 h	3.5×10^{-13}	0.002 %	**pass**
type2-random-0: modified life (worst of 32 values)	174228.3 h	174228.3 h	3.1×10^{-12}	0.002 %	**pass**
type2-random-1: modified life (worst of 34 values)	183508.5 h	183508.5 h	8.9×10^{-10}	0.002 %	**pass**
type2-random-2: modified life (worst of 34 values)	186189.4 h	186189.4 h	1.6×10^{-12}	0.002 %	**pass**
type2-random-3: modified life (worst of 32 values)	23074.74 h	23074.74 h	2.5×10^{-10}	0.002 %	**pass**
type2-random-4: modified life (worst of 28 values)	64370.65 h	64370.65 h	1.6×10^{-10}	0.002 %	**pass**
type2-random-5: modified life (worst of 34 values)	169833.7 h	169833.7 h	2.0×10^{-13}	0.002 %	**pass**
type3-base: modified life (worst of 21 values)	28305.12 h	28305.12 h	2.3×10^{-10}	0.002 %	**pass**
type3-loads-2: modified life (worst of 21 values)	54182.39 h	54182.39 h	2.2×10^{-10}	0.002 %	**pass**
type3-loads-3: modified life (worst of 21 values)	64185.05 h	64185.05 h	9.5×10^{-12}	0.002 %	**pass**
type3-loads-4: modified life (worst of 21 values)	74819.76 h	74819.76 h	1.7×10^{-10}	0.002 %	**pass**
type3-loads-5: modified life (worst of 21 values)	175878.1 h	175878.1 h	10^{-9}	0.002 %	**pass**
type3-loads-6: modified life (worst of 21 values)	59007.23 h	59007.23 h	3.4×10^{-10}	0.002 %	**pass**
type3-loads-7: modified life (worst of 21 values)	24052.51 h	24052.51 h	3.5×10^{-9}	0.002 %	**pass**
type3-loads-8: modified life (worst of 21 values)	177788 h	177788 h	3.2×10^{-11}	0.002 %	**pass**
type3-operating-0: modified life (worst of 21 values)	82478.22 h	82478.22 h	2.3×10^{-10}	0.002 %	**pass**
type3-operating-1: modified life (worst of 21 values)	3646.02 h	3646.02 h	2.3×10^{-10}	0.002 %	**pass**
type3-operating-2: modified life (worst of 21 values)	832.9519 h	832.9519 h	2.3×10^{-10}	0.002 %	**pass**
type3-operating-3: modified life (worst of 21 values)	1758.265 h	1758.265 h	2.3×10^{-10}	0.002 %	**pass**
type3-offset-0: modified life (worst of 21 values)	33620.3 h	33620.3 h	7.7×10^{-14}	0.002 %	**pass**

Case	Reference	Axomek	Deviation	Tolerance	
type3-offset-1: modified life (worst of 21 values)	43041.19 h	43041.19 h	3.4×10^{-9}	0.002 %	**pass**
type3-material: modified life (worst of 21 values)	5436.566 h	5436.566 h	1.1×10^{-9}	0.002 %	**pass**
type3-system-rows: modified life (worst of 30 values)	92360 h	92360 h	6.1×10^{-10}	0.002 %	**pass**
type3-random-2: modified life (worst of 20 values)	2610.253 h	2610.253 h	1.1×10^{-9}	0.002 %	**pass**
type4-base: modified life (worst of 30 values)	982667 h	982667 h	1.5×10^{-9}	0.002 %	**pass**
type4-loads-1: modified life (worst of 30 values)	13730.19 h	13730.19 h	2.8×10^{-14}	0.002 %	**pass**
type4-loads-2: modified life (worst of 30 values)	1.0028×10^{6} h	1.0028×10^{6} h	2.4×10^{-9}	0.002 %	**pass**
type4-loads-3: modified life (worst of 30 values)	1.6951×10^{6} h	1.6951×10^{6} h	2.4×10^{-12}	0.002 %	**pass**
type4-loads-4: modified life (worst of 30 values)	1.7114×10^{6} h	1.7114×10^{6} h	1.2×10^{-14}	0.002 %	**pass**
type4-loads-5: modified life (worst of 30 values)	1.1945×10^{6} h	1.1945×10^{6} h	8.3×10^{-11}	0.002 %	**pass**
type4-loads-6: modified life (worst of 30 values)	957318.6 h	957318.6 h	1.1×10^{-10}	0.002 %	**pass**
type4-loads-7: modified life (worst of 30 values)	816062 h	816062 h	10^{-13}	0.002 %	**pass**
type4-loads-8: modified life (worst of 30 values)	1.1765×10^{6} h	1.1765×10^{6} h	3.2×10^{-10}	0.002 %	**pass**
type4-operating-0: modified life (worst of 30 values)	4.9133×10^{6} h	4.9133×10^{6} h	1.5×10^{-9}	0.002 %	**pass**
type4-operating-1: modified life (worst of 30 values)	119550.9 h	119550.9 h	1.5×10^{-9}	0.002 %	**pass**
type4-operating-2: modified life (worst of 30 values)	19682.52 h	19682.52 h	1.5×10^{-9}	0.002 %	**pass**
type4-operating-3: modified life (worst of 30 values)	120251.6 h	120251.6 h	1.5×10^{-9}	0.002 %	**pass**
type4-offset-0: modified life (worst of 30 values)	993268.6 h	993268.6 h	3.8×10^{-12}	0.002 %	**pass**
type4-offset-1: modified life (worst of 30 values)	1.0005×10^{6} h	1.0005×10^{6} h	3.9×10^{-12}	0.002 %	**pass**
type4-material: modified life (worst of 30 values)	452810.4 h	452810.4 h	2.4×10^{-15}	0.002 %	**pass**
type4-system-rows: modified life (worst of 48 values)	1.8199×10^{6} h	1.8199×10^{6} h	9.8×10^{-11}	0.002 %	**pass**
type4-random-0: modified life (worst of 34 values)	938549.5 h	938549.5 h	6.4×10^{-11}	0.002 %	**pass**
type4-random-1: modified life (worst of 32 values)	810881.4 h	810881.4 h	1.4×10^{-14}	0.002 %	**pass**

Case	Reference	Axomek	Deviation	Tolerance	
type4-random-2: modified life (worst of 32 values)	141588.1 h	141588.1 h	4.6×10^{-13}	0.002 %	**pass**
type4-random-3: modified life (worst of 26 values)	45010.45 h	45010.45 h	1.4×10^{-10}	0.002 %	**pass**
type4-random-4: modified life (worst of 34 values)	337817.4 h	337817.4 h	2.1×10^{-12}	0.002 %	**pass**
type5-base: modified life (worst of 30 values)	16265.6 h	16265.6 h	4.1×10^{-13}	0.002 %	**pass**
type5-loads-0: modified life (worst of 30 values)	5.4496×10^{7} h	5.4496×10^{7} h	6.3×10^{-6}	0.002 %	**pass**
type5-loads-1: modified life (worst of 30 values)	283.9226 h	283.9226 h	1.7×10^{-12}	0.002 %	**pass**
type5-loads-2: modified life (worst of 30 values)	16265.6 h	16265.6 h	9.5×10^{-13}	0.002 %	**pass**
type5-loads-3: modified life (worst of 30 values)	36693.97 h	36693.97 h	2.3×10^{-13}	0.002 %	**pass**
type5-loads-4: modified life (worst of 30 values)	36615.91 h	36615.91 h	5.5×10^{-12}	0.002 %	**pass**
type5-loads-5: modified life (worst of 30 values)	1676.992 h	1676.992 h	1.1×10^{-12}	0.002 %	**pass**
type5-loads-6: modified life (worst of 30 values)	2286.324 h	2286.324 h	2.8×10^{-10}	0.002 %	**pass**
type5-operating-0: modified life (worst of 30 values)	57062.08 h	57062.08 h	4.0×10^{-13}	0.002 %	**pass**
type5-operating-1: modified life (worst of 30 values)	5544.718 h	5544.718 h	3.3×10^{-13}	0.002 %	**pass**
type5-operating-2: modified life (worst of 30 values)	1174.175 h	1174.175 h	3.3×10^{-13}	0.002 %	**pass**
type5-operating-3: modified life (worst of 30 values)	1903.374 h	1903.374 h	3.7×10^{-13}	0.002 %	**pass**
type5-clearance-0: modified life (worst of 30 values)	16265.6 h	16265.6 h	4.1×10^{-13}	0.002 %	**pass**
type5-clearance-1: modified life (worst of 30 values)	16265.6 h	16265.6 h	4.1×10^{-13}	0.002 %	**pass**
type5-clearance-2: modified life (worst of 30 values)	16265.6 h	16265.6 h	4.1×10^{-13}	0.002 %	**pass**
type5-offset-0: modified life (worst of 30 values)	16216.02 h	16216.02 h	4.8×10^{-13}	0.002 %	**pass**
type5-offset-1: modified life (worst of 30 values)	16235.69 h	16235.69 h	3.8×10^{-13}	0.002 %	**pass**
type5-material: modified life (worst of 30 values)	7124.227 h	7124.227 h	7.7×10^{-13}	0.002 %	**pass**
type5-system-rows: modified life (worst of 48 values)	51053.22 h	51053.22 h	6.0×10^{-11}	0.002 %	**pass**
type5-random-0: modified life (worst of 32 values)	16526.94 h	16526.94 h	9.6×10^{-13}	0.002 %	**pass**

Case	Reference	Axomek	Deviation	Tolerance	
type5-random-1: modified life (worst of 32 values)	26897.48 h	26897.48 h	3.8×10^{-13}	0.002 %	**pass**
type5-random-2: modified life (worst of 28 values)	24163.17 h	24163.17 h	8.3×10^{-12}	0.002 %	**pass**
type5-random-3: modified life (worst of 26 values)	1993.042 h	1993.042 h	4.9×10^{-13}	0.002 %	**pass**
type5-random-4: modified life (worst of 32 values)	43550.45 h	43550.45 h	2.0×10^{-8}	0.002 %	**pass**
type5-random-5: modified life (worst of 32 values)	6001.144 h	6001.144 h	4.2×10^{-11}	0.002 %	**pass**
type6-base: modified life (worst of 30 values)	38442.95 h	38442.95 h	8.6×10^{-10}	0.002 %	**pass**
type6-loads-0: modified life (worst of 30 values)	5.4113×10^{7} h	5.4113×10^{7} h	1.4×10^{-13}	0.002 %	**pass**
type6-loads-1: modified life (worst of 30 values)	673.9942 h	673.9942 h	1.7×10^{-14}	0.002 %	**pass**
type6-loads-2: modified life (worst of 30 values)	62241.49 h	62241.49 h	1.2×10^{-15}	0.002 %	**pass**
type6-loads-3: modified life (worst of 30 values)	154044.6 h	154044.6 h	4.9×10^{-15}	0.002 %	**pass**
type6-loads-4: modified life (worst of 30 values)	156669.1 h	156669.1 h	5.1×10^{-14}	0.002 %	**pass**
type6-loads-5: modified life (worst of 30 values)	114096.9 h	114096.9 h	2.8×10^{-13}	0.002 %	**pass**
type6-loads-6: modified life (worst of 30 values)	69244.07 h	69244.07 h	3.5×10^{-11}	0.002 %	**pass**
type6-loads-7: modified life (worst of 30 values)	30274.63 h	30274.63 h	1.2×10^{-11}	0.002 %	**pass**
type6-loads-8: modified life (worst of 30 values)	64332.24 h	64332.24 h	2.3×10^{-12}	0.002 %	**pass**
type6-operating-0: modified life (worst of 30 values)	117199.3 h	117199.3 h	8.6×10^{-10}	0.002 %	**pass**
type6-operating-1: modified life (worst of 30 values)	6660.348 h	6660.348 h	8.6×10^{-10}	0.002 %	**pass**
type6-operating-2: modified life (worst of 30 values)	1398.921 h	1398.921 h	8.6×10^{-10}	0.002 %	**pass**
type6-operating-3: modified life (worst of 30 values)	2996.172 h	2996.172 h	8.6×10^{-10}	0.002 %	**pass**
type6-offset-0: modified life (worst of 30 values)	36959.02 h	36959.02 h	2.3×10^{-11}	0.002 %	**pass**
type6-offset-1: modified life (worst of 30 values)	36302.25 h	36302.25 h	1.7×10^{-10}	0.002 %	**pass**
type6-material: modified life (worst of 30 values)	9440.825 h	9440.825 h	4.8×10^{-11}	0.002 %	**pass**
type6-system-rows: modified life (worst of 48 values)	1.524×10^{6} h	1.524×10^{6} h	1.3×10^{-10}	0.002 %	**pass**

Case	Reference	Axomek	Deviation	Tolerance	
type6-random-0: modified life (worst of 32 values)	37517.02 h	37517.02 h	1.2×10^{-9}	0.002 %	**pass**
type6-random-1: modified life (worst of 32 values)	64782.91 h	64782.91 h	9.2×10^{-15}	0.002 %	**pass**
type6-random-2: modified life (worst of 26 values)	26354.27 h	26354.27 h	1.3×10^{-15}	0.002 %	**pass**
type6-random-3: modified life (worst of 26 values)	4115.41 h	4115.41 h	2.1×10^{-9}	0.002 %	**pass**
type6-random-4: modified life (worst of 28 values)	13122.09 h	13122.09 h	2.0×10^{-9}	0.002 %	**pass**
type6-random-5: modified life (worst of 26 values)	1865.392 h	1865.392 h	2.2×10^{-9}	0.002 %	**pass**
type7-base: modified life (worst of 27 values)	23212.97 h	23212.97 h	5.6×10^{-14}	0.002 %	**pass**
type7-loads-0: modified life (worst of 27 values)	5.0243×10^{6} h	5.0243×10^{6} h	1.1×10^{-15}	0.002 %	**pass**
type7-loads-1: modified life (worst of 27 values)	5.0243×10^{6} h	5.0243×10^{6} h	1.1×10^{-15}	0.002 %	**pass**
type7-loads-2: modified life (worst of 27 values)	3088.186 h	3088.186 h	1.1×10^{-14}	0.002 %	**pass**
type7-loads-3: modified life (worst of 27 values)	3088.186 h	3088.186 h	1.1×10^{-14}	0.002 %	**pass**
type7-loads-4: modified life (worst of 27 values)	37625.99 h	37625.99 h	1.3×10^{-12}	0.002 %	**pass**
type7-loads-5: modified life (worst of 27 values)	52286.28 h	52286.28 h	1.3×10^{-10}	0.002 %	**pass**
type7-operating-0: modified life (worst of 27 values)	67001.15 h	67001.15 h	5.4×10^{-14}	0.002 %	**pass**
type7-operating-1: modified life (worst of 27 values)	3476.539 h	3476.539 h	4.4×10^{-14}	0.002 %	**pass**
type7-operating-2: modified life (worst of 27 values)	665.4971 h	665.4971 h	4.0×10^{-14}	0.002 %	**pass**
type7-operating-3: modified life (worst of 27 values)	1753.51 h	1753.51 h	4.8×10^{-14}	0.002 %	**pass**
type7-offset-0: modified life (worst of 27 values)	23212.97 h	23212.97 h	2.5×10^{-15}	0.002 %	**pass**
type7-offset-1: modified life (worst of 27 values)	23212.97 h	23212.97 h	2.5×10^{-15}	0.002 %	**pass**
type7-material: modified life (worst of 27 values)	5369.352 h	5369.352 h	5.3×10^{-15}	0.002 %	**pass**
type7-system-rows: modified life (worst of 42 values)	1.3928×10^{6} h	1.3928×10^{6} h	7.0×10^{-13}	0.002 %	**pass**
type7-random-0: modified life (worst of 25 values)	19754.84 h	19754.84 h	2.0×10^{-14}	0.002 %	**pass**
type7-random-1: modified life (worst of 29 values)	72109.06 h	72109.06 h	6.7×10^{-15}	0.002 %	**pass**

Case	Reference	Axomek	Deviation	Tolerance	
type7-random-2: modified life (worst of 29 values)	69509.44 h	69509.44 h	2.6×10^{-15}	0.002 %	**pass**
type7-random-3: modified life (worst of 26 values)	9047.727 h	9047.727 h	5.0×10^{-15}	0.002 %	**pass**
type7-random-4: modified life (worst of 26 values)	2597.97 h	2597.97 h	4.9×10^{-15}	0.002 %	**pass**
type7-random-5: modified life (worst of 26 values)	6086.636 h	6086.636 h	4.9×10^{-16}	0.002 %	**pass**
type8-base: modified life (worst of 28 values)	35286.91 h	35286.91 h	2.0×10^{-10}	0.002 %	**pass**
type8-loads-0: modified life (worst of 28 values)	5.2072×10^{6} h	5.2072×10^{6} h	5.3×10^{-8}	0.002 %	**pass**
type8-loads-1: modified life (worst of 28 values)	5.2072×10^{6} h	5.2072×10^{6} h	2.2×10^{-8}	0.002 %	**pass**
type8-loads-2: modified life (worst of 28 values)	2959.121 h	2959.121 h	6.0×10^{-15}	0.002 %	**pass**
type8-loads-3: modified life (worst of 28 values)	2959.121 h	2959.121 h	5.9×10^{-9}	0.002 %	**pass**
type8-loads-4: modified life (worst of 28 values)	51663.44 h	51663.44 h	4.5×10^{-14}	0.002 %	**pass**
type8-loads-5: modified life (worst of 28 values)	39242.82 h	39242.82 h	7.8×10^{-15}	0.002 %	**pass**
type8-operating-0: modified life (worst of 28 values)	101166.3 h	101166.3 h	2.0×10^{-10}	0.002 %	**pass**
type8-operating-1: modified life (worst of 28 values)	5508.325 h	5508.325 h	2.0×10^{-10}	0.002 %	**pass**
type8-operating-2: modified life (worst of 28 values)	1040.557 h	1040.557 h	2.0×10^{-10}	0.002 %	**pass**
type8-operating-3: modified life (worst of 28 values)	2849.493 h	2849.493 h	2.0×10^{-10}	0.002 %	**pass**
type8-operating-5: modified life (worst of 28 values)	443.0462 h	443.0462 h	2.0×10^{-10}	0.002 %	**pass**
type8-offset-1: modified life (worst of 28 values)	35286.91 h	35286.91 h	8.9×10^{-15}	0.002 %	**pass**
type8-material: modified life (worst of 28 values)	7250.81 h	7250.81 h	4.9×10^{-13}	0.002 %	**pass**
type8-system-rows: modified life (worst of 44 values)	1.1651×10^{6} h	1.1651×10^{6} h	10^{-10}	0.002 %	**pass**
type8-random-0: modified life (worst of 30 values)	8776.126 h	8776.126 h	2.4×10^{-10}	0.002 %	**pass**
type8-random-1: modified life (worst of 26 values)	2327.001 h	2327.001 h	2.9×10^{-14}	0.002 %	**pass**
type8-random-2: modified life (worst of 30 values)	73619.67 h	73619.67 h	2.7×10^{-9}	0.002 %	**pass**
type8-random-4: modified life (worst of 29 values)	4016.882 h	4016.882 h	2.2×10^{-15}	0.002 %	**pass**

Case	Reference	Axomek	Deviation	Tolerance	
type8-random-5: modified life (worst of 26 values)	18158.41 h	18158.41 h	3.6×10^{-11}	0.002 %	**pass**
type9-base: modified life (worst of 586 values)	86837.5 h	86837.5 h	10.0×10^{-9}	0.002 %	**pass**
type9-loads-0: modified life (worst of 586 values)	3.0646×10^{7} h	3.0646×10^{7} h	6.9×10^{-12}	0.002 %	**pass**
type9-loads-1: modified life (worst of 586 values)	9034.397 h	9034.397 h	4.0×10^{-11}	0.002 %	**pass**
type9-loads-2: modified life (worst of 586 values)	86837.5 h	86837.5 h	6.1×10^{-11}	0.002 %	**pass**
type9-loads-3: modified life (worst of 586 values)	238287.9 h	238287.9 h	1.5×10^{-10}	0.002 %	**pass**
type9-loads-4: modified life (worst of 586 values)	237740.2 h	237740.2 h	2.0×10^{-9}	0.002 %	**pass**
type9-loads-5: modified life (worst of 586 values)	28.53895 h	28.53895 h	2.1×10^{-13}	0.002 %	**pass**
type9-loads-6: modified life (worst of 586 values)	465.1471 h	465.1471 h	3.3×10^{-10}	0.002 %	**pass**
type9-operating-0: modified life (worst of 586 values)	237551.2 h	237551.2 h	10.0×10^{-9}	0.002 %	**pass**
type9-operating-1: modified life (worst of 586 values)	9321.681 h	9321.681 h	10.0×10^{-9}	0.002 %	**pass**
type9-operating-2: modified life (worst of 586 values)	2192.917 h	2192.917 h	10.0×10^{-9}	0.002 %	**pass**
type9-operating-3: modified life (worst of 586 values)	4595.2 h	4595.2 h	10.0×10^{-9}	0.002 %	**pass**
type9-clearance-0: modified life (worst of 586 values)	151969.5 h	151969.5 h	7.5×10^{-9}	0.002 %	**pass**
type9-clearance-1: modified life (worst of 586 values)	128225.1 h	128225.1 h	6.5×10^{-9}	0.002 %	**pass**
type9-clearance-2: modified life (worst of 586 values)	43346.37 h	43346.37 h	2.9×10^{-9}	0.002 %	**pass**
type9-offset-0: modified life (worst of 586 values)	82176.96 h	82176.96 h	4.5×10^{-10}	0.002 %	**pass**
type9-offset-1: modified life (worst of 586 values)	86793.04 h	86793.04 h	9.3×10^{-9}	0.002 %	**pass**
type9-material: modified life (worst of 586 values)	18716.47 h	18716.47 h	10.0×10^{-9}	0.002 %	**pass**
type9-system-rows: modified life (worst of 1160 values)	1.3457×10^{6} h	1.3457×10^{6} h	2.6×10^{-11}	0.002 %	**pass**
type9-lamina-0: modified life (worst of 306 values)	86516.08 h	86516.08 h	1.6×10^{-10}	0.002 %	**pass**
type9-lamina-1: modified life (worst of 586 values)	93795.54 h	93795.54 h	10.0×10^{-9}	0.002 %	**pass**
type9-lamina-2: modified life (worst of 222 values)	97395.82 h	97395.82 h	1.3×10^{-10}	0.002 %	**pass**

Case	Reference	Axomek	Deviation	Tolerance	
type9-random-0: modified life (worst of 627 values)	92.20167 h	92.20167 h	3.4×10^{-10}	0.002 %	**pass**
type9-random-1: modified life (worst of 668 values)	472464 h	472464 h	3.9×10^{-10}	0.002 %	**pass**
type9-random-2: modified life (worst of 668 values)	54.35689 h	54.35689 h	2.9×10^{-13}	0.002 %	**pass**
type9-random-3: modified life (worst of 545 values)	7.35985 h	7.35985 h	6.0×10^{-10}	0.002 %	**pass**
type9-random-4: modified life (worst of 627 values)	167877.1 h	167877.1 h	1.2×10^{-11}	0.002 %	**pass**
type9-random-5: modified life (worst of 504 values)	4713.433 h	4713.433 h	6.8×10^{-13}	0.002 %	**pass**
type10-base: modified life (worst of 1816 values)	1.0281×10^{6} h	1.0281×10^{6} h	5.6×10^{-10}	0.002 %	**pass**
type10-loads-0: modified life (worst of 1816 values)	8.7749×10^{8} h	8.7749×10^{8} h	1.2×10^{-11}	0.002 %	**pass**
type10-loads-1: modified life (worst of 1816 values)	4.9728×10^{6} h	4.9728×10^{6} h	5.6×10^{-13}	0.002 %	**pass**
type10-loads-2: modified life (worst of 1816 values)	1.6866×10^{7} h	1.6866×10^{7} h	3.1×10^{-14}	0.002 %	**pass**
type10-loads-3: modified life (worst of 1816 values)	2.9008×10^{7} h	2.9008×10^{7} h	1.5×10^{-10}	0.002 %	**pass**
type10-loads-4: modified life (worst of 1816 values)	2.8969×10^{7} h	2.8969×10^{7} h	4.7×10^{-11}	0.002 %	**pass**
type10-loads-5: modified life (worst of 1816 values)	1.036×10^{7} h	1.036×10^{7} h	7.4×10^{-9}	0.002 %	**pass**
type10-loads-6: modified life (worst of 1816 values)	2.129×10^{7} h	2.129×10^{7} h	5.2×10^{-11}	0.002 %	**pass**
type10-operating-0: modified life (worst of 1816 values)	4.4463×10^{6} h	4.4463×10^{6} h	5.6×10^{-10}	0.002 %	**pass**
type10-operating-1: modified life (worst of 1816 values)	143103.8 h	143103.8 h	5.6×10^{-10}	0.002 %	**pass**
type10-operating-2: modified life (worst of 1816 values)	18657.1 h	18657.1 h	5.6×10^{-10}	0.002 %	**pass**
type10-operating-3: modified life (worst of 1816 values)	112665.4 h	112665.4 h	5.6×10^{-10}	0.002 %	**pass**
type10-operating-4: modified life (worst of 1816 values)	5378.448 h	5378.448 h	5.6×10^{-10}	0.002 %	**pass**
type10-operating-5: modified life (worst of 1816 values)	7178.486 h	7178.486 h	5.6×10^{-10}	0.002 %	**pass**
type10-clearance-0: modified life (worst of 1816 values)	1.5306×10^{6} h	1.5306×10^{6} h	1.4×10^{-10}	0.002 %	**pass**
type10-clearance-1: modified life (worst of 1816 values)	1.3656×10^{6} h	1.3656×10^{6} h	1.3×10^{-10}	0.002 %	**pass**
type10-clearance-2: modified life (worst of 1816 values)	705883.1 h	705883.1 h	5.6×10^{-9}	0.002 %	**pass**

Case	Reference	Axomek	Deviation	Tolerance	
type10-offset-0: modified life (worst of 1816 values)	1.0225×10^{6} h	1.0225×10^{6} h	2.3×10^{-12}	0.002 %	**pass**
type10-offset-1: modified life (worst of 1816 values)	1.0206×10^{6} h	1.0206×10^{6} h	2.3×10^{-11}	0.002 %	**pass**
type10-material: modified life (worst of 1816 values)	326742.2 h	326742.2 h	5.6×10^{-10}	0.002 %	**pass**
type10-system-rows: modified life (worst of 3620 values)	4.1612×10^{6} h	4.1612×10^{6} h	1.6×10^{-11}	0.002 %	**pass**
type10-lamina-0: modified life (worst of 936 values)	1.0313×10^{6} h	1.0313×10^{6} h	9.4×10^{-11}	0.002 %	**pass**
type10-lamina-1: modified life (worst of 1816 values)	1.0646×10^{6} h	1.0646×10^{6} h	5.6×10^{-10}	0.002 %	**pass**
type10-lamina-2: modified life (worst of 672 values)	1.0503×10^{6} h	1.0503×10^{6} h	5.4×10^{-10}	0.002 %	**pass**
type10-random-0: modified life (worst of 1980 values)	565713.3 h	565713.3 h	1.1×10^{-10}	0.002 %	**pass**
type10-random-1: modified life (worst of 1898 values)	323818.6 h	323818.6 h	3.3×10^{-11}	0.002 %	**pass**
type10-random-2: modified life (worst of 1734 values)	870747.7 h	870747.7 h	1.2×10^{-11}	0.002 %	**pass**
type10-random-3: modified life (worst of 1980 values)	3.3994×10^{6} h	3.3994×10^{6} h	6.0×10^{-12}	0.002 %	**pass**
type10-random-4: modified life (worst of 1898 values)	1.3013×10^{6} h	1.3013×10^{6} h	4.1×10^{-11}	0.002 %	**pass**
type10-random-5: modified life (worst of 1980 values)	817898.1 h	817898.1 h	3.7×10^{-9}	0.002 %	**pass**
type11-base: modified life (worst of 1078 values)	967784.4 h	967784.4 h	2.5×10^{-12}	0.002 %	**pass**
type11-loads-0: modified life (worst of 1078 values)	2.3259×10^{7} h	2.3259×10^{7} h	4.0×10^{-10}	0.002 %	**pass**
type11-loads-1: modified life (worst of 1078 values)	7527.353 h	7527.353 h	2.4×10^{-12}	0.002 %	**pass**
type11-loads-2: modified life (worst of 1078 values)	82828.76 h	82828.76 h	8.1×10^{-10}	0.002 %	**pass**
type11-loads-3: modified life (worst of 1078 values)	257676.6 h	257676.6 h	2.1×10^{-10}	0.002 %	**pass**
type11-loads-4: modified life (worst of 1078 values)	258396.6 h	258396.6 h	1.2×10^{-9}	0.002 %	**pass**
type11-loads-5: modified life (worst of 1078 values)	498.9879 h	498.9879 h	7.9×10^{-11}	0.002 %	**pass**
type11-loads-6: modified life (worst of 1078 values)	25956.05 h	25956.05 h	1.6×10^{-9}	0.002 %	**pass**
type11-operating-0: modified life (worst of 1078 values)	867456.9 h	867456.9 h	3.4×10^{-12}	0.002 %	**pass**
type11-operating-1: modified life (worst of 1078 values)	17752.57 h	17752.57 h	2.5×10^{-12}	0.002 %	**pass**

Case	Reference	Axomek	Deviation	Tolerance	
type11-operating-2: modified life (worst of 1078 values)	4924.288 h	4924.288 h	2.5×10^{-12}	0.002 %	**pass**
type11-operating-3: modified life (worst of 1078 values)	11684.12 h	11684.12 h	2.9×10^{-12}	0.002 %	**pass**
type11-clearance-0: modified life (worst of 1078 values)	1.8964×10^{6} h	1.8964×10^{6} h	7.9×10^{-11}	0.002 %	**pass**
type11-clearance-1: modified life (worst of 1078 values)	1.2835×10^{6} h	1.2835×10^{6} h	2.4×10^{-13}	0.002 %	**pass**
type11-clearance-2: modified life (worst of 1078 values)	157296.9 h	157296.9 h	1.1×10^{-11}	0.002 %	**pass**
type11-offset-0: modified life (worst of 1078 values)	972696.4 h	972696.4 h	1.7×10^{-12}	0.002 %	**pass**
type11-offset-1: modified life (worst of 1078 values)	967597.1 h	967597.1 h	8.5×10^{-12}	0.002 %	**pass**
type11-material: modified life (worst of 1078 values)	190658.4 h	190658.4 h	3.8×10^{-12}	0.002 %	**pass**
type11-system-rows: modified life (worst of 2144 values)	2.4244×10^{6} h	2.4244×10^{6} h	2.8×10^{-11}	0.002 %	**pass**
type11-lamina-0: modified life (worst of 558 values)	960245.2 h	960245.2 h	4.5×10^{-11}	0.002 %	**pass**
type11-lamina-1: modified life (worst of 1078 values)	1.018×10^{6} h	1.018×10^{6} h	2.2×10^{-12}	0.002 %	**pass**
type11-lamina-2: modified life (worst of 402 values)	994015.5 h	994015.5 h	8.6×10^{-11}	0.002 %	**pass**
type11-random-0: modified life (worst of 1119 values)	197007.4 h	197007.4 h	3.3×10^{-12}	0.002 %	**pass**
type11-random-1: modified life (worst of 1037 values)	31963.9 h	31963.9 h	2.7×10^{-10}	0.002 %	**pass**
type11-random-2: modified life (worst of 996 values)	32320.75 h	32320.75 h	1.1×10^{-9}	0.002 %	**pass**
type11-random-3: modified life (worst of 1037 values)	101546.4 h	101546.4 h	1.5×10^{-11}	0.002 %	**pass**
type11-random-4: modified life (worst of 1037 values)	566.137 h	566.137 h	7.6×10^{-8}	0.002 %	**pass**
type11-random-5: modified life (worst of 996 values)	385.4294 h	385.4294 h	1.7×10^{-9}	0.002 %	**pass**
type12-base: modified life (worst of 832 values)	2046.999 h	2046.999 h	3.1×10^{-12}	0.002 %	**pass**
type12-loads-0: modified life (worst of 832 values)	3.6634×10^{6} h	3.6634×10^{6} h	9.9×10^{-16}	0.002 %	**pass**
type12-loads-1: modified life (worst of 832 values)	3.6634×10^{6} h	3.6634×10^{6} h	1.2×10^{-16}	0.002 %	**pass**
type12-loads-2: modified life (worst of 832 values)	56.17192 h	56.17192 h	2.8×10^{-11}	0.002 %	**pass**
type12-loads-3: modified life (worst of 832 values)	56.17192 h	56.17192 h	1.6×10^{-11}	0.002 %	**pass**

Case	Reference	Axomek	Deviation	Tolerance	
type12-loads-4: modified life (worst of 832 values)	1855.137 h	1855.137 h	2.9×10^{-12}	0.002 %	**pass**
type12-loads-5: modified life (worst of 832 values)	1616.112 h	1616.112 h	8.2×10^{-16}	0.002 %	**pass**
type12-operating-0: modified life (worst of 832 values)	6761.548 h	6761.548 h	3.1×10^{-12}	0.002 %	**pass**
type12-operating-1: modified life (worst of 832 values)	632.8951 h	632.8951 h	3.1×10^{-12}	0.002 %	**pass**
type12-operating-2: modified life (worst of 832 values)	131.6262 h	131.6262 h	3.1×10^{-12}	0.002 %	**pass**
type12-operating-3: modified life (worst of 832 values)	236.3059 h	236.3059 h	3.1×10^{-12}	0.002 %	**pass**
type12-operating-5: modified life (worst of 832 values)	119.8669 h	119.8669 h	3.1×10^{-12}	0.002 %	**pass**
type12-offset-0: modified life (worst of 832 values)	2046.999 h	2046.999 h	1.7×10^{-12}	0.002 %	**pass**
type12-offset-1: modified life (worst of 832 values)	2046.999 h	2046.999 h	1.7×10^{-11}	0.002 %	**pass**
type12-material: modified life (worst of 832 values)	514.1936 h	514.1936 h	3.1×10^{-12}	0.002 %	**pass**
type12-system-rows: modified life (worst of 1652 values)	1.3144×10^{6} h	1.3144×10^{6} h	2.4×10^{-12}	0.002 %	**pass**
type12-lamina-0: modified life (worst of 432 values)	2054.212 h	2054.212 h	1.7×10^{-11}	0.002 %	**pass**
type12-lamina-1: modified life (worst of 832 values)	2184.425 h	2184.425 h	3.1×10^{-12}	0.002 %	**pass**
type12-lamina-2: modified life (worst of 312 values)	2260.589 h	2260.589 h	3.0×10^{-13}	0.002 %	**pass**
type12-random-0: modified life (worst of 750 values)	1065.198 h	1065.198 h	1.6×10^{-11}	0.002 %	**pass**
type12-random-1: modified life (worst of 791 values)	5891.94 h	5891.94 h	3.8×10^{-13}	0.002 %	**pass**
type12-random-2: modified life (worst of 914 values)	2593.745 h	2593.745 h	3.8×10^{-12}	0.002 %	**pass**
type12-random-3: modified life (worst of 750 values)	2172.728 h	2172.728 h	2.4×10^{-11}	0.002 %	**pass**
type12-random-4: modified life (worst of 914 values)	7169.682 h	7169.682 h	4.7×10^{-13}	0.002 %	**pass**
type12-random-5: modified life (worst of 750 values)	310.7288 h	310.7288 h	2.0×10^{-11}	0.002 %	**pass**

A bearing with a soft outer ring: its life and life check

Kind of reference	The same cases calculated by other programs or methods
Implements	ISO 16281:2025, Formula (9); the hardness factor (HRC/58)^3.6 on the softer contact's capacity
Reference source	Audit RR1-S-02's independent hand calculation from the model's element loads (Q_ci 7 216.79 N, Q_ce 12 743.99 N, Q_ei 849.10 N, Q_ee 890.17 N): 1 695.6454 h
Cases	1 of 1 within tolerance

A 6207 at 3 kN radial and 1 500 rpm: it passed its 5 000 h before (12 515 h), as if hard.

Case	Reference	Axomek	Deviation	Tolerance	
6207, outer ring 45 HRC: L10r	1695.645 h	1695.645 h	1.5×10^{-8}	0.01 %	**pass**

Consistency checks of a module against its own parts

Bearing thermal speeds over a duty cycle: DIN 732 per load case

Kind of reference	Consistency checks of a module against its own parts
Implements	DIN 732:2010
Reference source	The same bearing solved alone at each load case
Cases	6 of 6 within tolerance

The 6207 bearing template (Fr 3.5 kN) with three load cases (full load, 2.5 × the load at 0.8 of the full-load nθ, unloaded at 0.9 of it): each case's thermally permissible speed nθ (DIN 732:2010, Eq. (1)–(8)) against the bearing solved on its own at that case's loads and speed, which is the single-case value; the full-load case against the bearing's own nθ; each case's speed over its nθ.

Case	Reference	Axomek	Deviation	Tolerance	
Full load: nθ per case = nθ of the bearing alone at the case's loads	6172.592 rpm	6172.592 rpm	0 %	10^{-12}	**pass**
Full load: speed over nθ	0.4698188	0.4698188	0 %	10^{-12}	**pass**
Heavy: nθ per case = nθ of the bearing alone at the case's loads	3424.052 rpm	3424.052 rpm	0 %	10^{-12}	**pass**
Heavy: speed over nθ	1.442173	1.442173	0 %	10^{-12}	**pass**
Idle: speed over nθ	0.6988403	0.6988403	0 %	10^{-12}	**pass**
Full load: nθ per case = the main operating point's nθ	6172.592 rpm	6172.592 rpm	0 %	10^{-12}	**pass**

Not validated against a published reference

- The four-component friction model, and Palmgren’s idle factor f0: DIN 732’s for an oil bath, where the AGMA ISO 14179-1 example compared takes a larger f0 by how deep the bearing dips (the load term M1 and the rest of M0 are compared with that example)
- The elastic outer ring and the high-speed ball model

Stated limits

Standards and scope (help: modules/bearings.md)

- Ratings follow ISO 281:2007, with fc from ISO/TR 1281-1:2008 at the actual groove radii, and ISO 76:2006 for C_0. Life from the element loads follows ISO 16281:2025.
- ISO 16281:2025 defines a_ISO for steel bearings only: a hybrid bearing whose life passes only through a_ISO (its basic life is below **Minimum life**) is "not determined" rather than passed.
- Above n·d_m = 1 000 000 mm/min (speed times pitch diameter) the rolling elements' centrifugal forces change the load distribution, and a note says so. Ball bearings can count them with the high-speed model; for roller bearings they are left out.
- ISO 281 rates a bearing with a contact angle near 45° differently as a radial or as a thrust bearing, so its ratings and lives jump there (ISO 16281:2025, Annex F).
- Hybrid bearings (steel rings, silicon nitride elements of E 300 000 MPa or more) follow ISO 20056-1/-2:2017: they keep their own moduli, C and C_u agree with ISO 20056-1, and C_0 and s_0 take ISO 20056-2's contact stress limits, 4 600 MPa for ball and 4 200 MPa for roller bearings (see Static load rating and static safety). ISO 20056-1 gives no b_m for hybrid spherical and tapered roller bearings: they are rated as steel ones, with a warning.
- Below κ = 0.1, where ISO 281 gives no a_ISO, the value at κ = 0.1 is used, with a warning.
- Fits change the clearance of deep groove ball and radial roller bearings only.
- The rings are rigid unless "Elastic outer ring" is ticked; the subsurface stress assumes frictionless contact.
- The four-component model leaves out churning losses. Minimum load and grease interval are bearing makers' rules of thumb.
- Thermal speeds are not given for thrust ball, double-row deep groove and double-row caged cylindrical roller bearings.
- Loads are taken as entered: bearings have no shock factor (f_d). With shocks, multiply the loads by it, or add a load case at the peak so that s_0 sees it.
- The presets and the bearing finder hold metric bearings only: no inch-bore bearings, adapter sleeves, pillow blocks or insert bearings. A bearing of another size can be imported from its maker's data (**Import CSV**, dimensions in mm); a pillow block's housing and a sleeve are not rated.
- Slewing rings are out of scope (bolted, flexible rings, rated by the maker's own curves); the four-point contact type is for the standard series (QJ). A four-point contact bearing whose pitch diameter is more than 15 times its ball diameter has a slewing ring's proportions: its checks are not determined, with a warning at the pitch diameter.
- A designation the bearing is not (another type, number of rows or size than the one modelled, such as "6207" on a double-row bearing) leaves every check of the bearing not determined until you use the designation's catalogue bearing or clear the designation. Its life and static safety then read "not determined", with no number, wherever they are shown (the shaft's results say "can't be determined"): the numbers would be those of the bearing as modelled. The same holds for any life or static safety whose check is not determined.

What is not covered (help: theory/iso-281.md)

- Failure modes other than subsurface fatigue: wear, micropitting, smearing, corrosion, electrical erosion, water in the oil.
- Very heavy loads (P above C_0 or 0.5·C): no warning.

- Truncation of the contact ellipse at the groove edge (ISO 281, Introduction): deep groove, angular contact and four-point ball bearings (single and double row) warn with the **Contact ellipse at shoulders** check; self-aligning and thrust ball bearings are not checked (see Bearings); the life is not changed by it.
- Hoop stress from tight fits and temperature effects on the steel (ISO/TR 1281-2, Clause 10).
- Hybrid bearings: rated by ISO 20056-1 and -2 with their own moduli (Annex D's effect of the moduli, by Hertz) and ISO 20056-2's static stress limits for silicon nitride elements.

Limits of the method **(help: theory/iso-76.md)**

- s_0 judges the contacts only. Ring, cage and housing strength and the fit are not part of it.
- ISO 76 excludes designs or conditions in which the contact area is considerably truncated, for example a contact ellipse running over the groove edge under high axial load (Clause 1). For deep groove, angular contact and four-point ball bearings (single and double row) the **Contact ellipse at shoulders** check warns when the ellipse runs over the shoulder (see Bearings); s_0 itself is not changed by it.
- The stress limits assume hardened bearing steel of current quality (Clause 1). For rings or elements softer than 58 HRC the app lowers σ_lim, and with it C_0 and s_0, by HV/HV(58 HRC), the Vickers hardness of ISO 18265's conversion table (the softest of the inner ring, outer ring and elements counts). For balls, whose capacity goes with σlim³, that is ISO 3408-5's static hardness factor fH0 = (HV/654)³ for ball screws: at 54 HRC (577 HV) σ_lim falls to 4 200 · 577/653 = 3 711 MPa and C_0 and s_0 to (577/653)³ = 0.69 of the hardened bearing's; for rollers s_0 goes with the square. The dynamic rating C takes (HRC/58)^3.6 (see ISO 281).
- Elements running directly on a shaft or in a housing are covered only when that raceway is equivalent to a bearing ring (Clause 1).

Known limits

- Units (R-387, R-931): the app and `axomek` read the same units typed, with the same factors, and show the same US customary units; a test holds them to it (`units-parity.spec.ts`), also for a field shown in another unit than its value's. It lists what still differs, on purpose or refused (never read wrongly): a design search's bound on a deflection is typed in µm (mil) in the app, as its check shows it, and in mm (in) by `axomek optimize --require`, which refuses nm there; a bolt's expansion coefficient α_S is typed in 10^{-6}/K in the app, in 1/K in `axomek`. A bare number in a bearing clearance is in µm in the app (the unit it shows) and in mm in `axomek` without `--units imperial` (base units, as every bare argument; mil with it, as in the app, but mm again with `--json`, which always reads base units). The bolt's expansion field refuses its own unit typed out (`12 10⁻⁶/K` reads as two numbers): type `12e-6 1/K`, or a bare 12 with metric units shown (with US units shown a bare number is in 10^{-6}/°F). The text output and the report keep a few units metric that the app shows in US units (m, J, GPa, √MPa, mm³, a steering's feel: `SCREEN_ONLY` in `crates/axm-model/src/units.rs`). Not compared: fields the app shows in % of a plain fraction (a torque amplitude, a modulus share), and the change lists of the Assistant and `axomek diff`, which give a clearance in mm.

- A motor bearing's minimum load and gyroscopic slip over its rotor's static eccentricity (user test rr1-motor; see "From 0.3.0"). While any rotor in the project pulls, their pass (or a warning for a bearing its axial load holds) counts only in the case rated in full, where all of these hold: - an uncoupled shaft alone: Couple linked bearings off, in no group (a rigid coupling, a bearing between shafts) and in no housing whose seats move each other; - no load cases; - the minimum load checked on the radial load F_r alone (not the equivalent load P of angular contact, tapered roller and thrust bearings), one bearing on its support; - no axial load on the shaft (an axial load shares out anew and can hold or let go of a bearing between the states); - the least loaded state (the least radial load, linear in the pulls without coupling) solved exactly, and the bearing loaded at the eccentricity. Every other such pass or held warning is not decided, saying why: "enter the eccentricity 0 to see it" (the Electric motor example, coupled, with an eccentricity entered, is one; it ships centred). Every bearing's gyroscopic slip margin is not decided in every case while any rotor pulls: the least radial load does not bound its balls' gyroscopic margin (a later release bounds the margin itself). A later release: a bound on the coupled shaft's least bearing load between centred and the eccentricity (the bearings' load–deflection laws bound how far load moves between them), solved with the groups' and housings' own equilibrium in each state and per load case; P, the axial holding and the balls' loads minimised themselves.
- While any rotor in the project pulls (a static eccentricity), the life, static safety, thermally permissible speed (DIN 732) and case depth of every bearing in the project are not decided (a pass or a held warning): a pull against another load leaves a bearing less loaded at the eccentricity than centred, and no bound from the supports' loads holds for a bearing's rating. Enter the eccentricity 0 to compare. A later release rates them at both states.
- With a rotor's pull, a one-way bearing's axial direction check can still pass in a narrow window between the states, as its 5 % threshold scales with the radial load; and a shaft's own checks (static and fatigue strength, deflection) are rated at the eccentricity as given, where a pull against the weight bends the span less than centred. A later release rates them at both states.
- A bearing's thermally permissible speed nθ is hidden beside any thermal speed check that is not determined, one whose speed is not its shaft's too, while the bearing's speed card still draws it (on the safe side: a figure left out); a later release hides it only where it is not decided.
- The rotor-pull refusals above apply to every bearing in the project once any rotor in it pulls, a gearbox's output shaft's too, though no pull reaches it; a later release takes only the shafts a pull reaches.
- On the steering's ball nut shaft, the friction of the nut's linked bearings shows in the power flow as power leaving that shaft (19 W in the Ball-nut EPS example): the nut's stage passes on what reaches its thread. The totals and efficiencies hold it once, as the report's power loss does.
- A bearing life that is not determined (see "From 0.3.0") is said so in place of its number, with these limits in 0.3.0: its basic life is hidden with it, also where only the modified life is undetermined (κ below 0.1, a hybrid's a_ISO), which is on the safe side; without a life limit only a contradicting designation or a slewing ring's proportions mark it, so κ below 0.1 then shows a plain life (the verdict lists that life as not checked; a hybrid's modified life reads "not determined" with or without a limit); and the variant comparison's CSV writes "not determined" where a number stood.

- A result not determined is hidden display by display, not by one shared rule: in 0.3.0 the displays reviewed (the report, Show the calculation, the CLI's text, the comparisons, the Overview, the cards, the modules' tiles and the Assistant's digest) say "not determined". Those found last are refused instead: the Changes dialog's and Compare projects' check rows and the Overview's key results of keys and bolts say "not determined (see the check)"; the assembly view gives no share of the limit for a check not determined or not computed; and in the report an element's results, and a key's or a fit's results its checks go by (safeties, capacities, permissible values, pressures, shear, the fit's Turning row; with `axomek solve`'s key and fit lines), say "not determined (see the checks)" as soon as one of its checks is not determined, determined ones with them; a key's page refuses its flank pressures and tooth shear stress the same way (its permissible values and capacity, which do not depend on its rating, stay). A bolted joint with a check not determined (beyond VDI 2230's limit G, a duty not computed) has its whole Results block refused the same way in the report's section and in `axomek solve`'s line (its tightening torque, permissible assembly preload, resiliences and embedding with it, which the bolt page still shows); its page refuses its preload needed, load factor, clamp loads, stresses, opening and the joint diagram, but the notes under them, on its page and under the report's section, still quote the eccentric model's figures (M_Sb/W_b, F_Sm/F_0.2min, F_K against F_Kab). Make it pass and `axomek fix` give such a check no value either. A module's page, once one of an item's checks is not determined, shows no tile's caption (it may quote another check's figure) and refuses its tiles and Details tied to no check (a worm pair's load value C / Clim), and the Overview refuses the item's tiles tied to no check the same way, including ones that do not depend on the rating: a worm pair's method, centre distance and ratio, a motor's power and magnetic stiffness read "not determined" too; a later release shows those again. A design search meets no requirement or objective on a result of a design with any check not determined (a requirement's own check may be cleared, and the system life and power loss take every item's), its dimensions apart, nor on a fit's yield safety resting on an estimated yield strength, even where that figure itself would stand: the safe side; a later release decides each figure on its own. A bearing whose life the engine puts outside its rating method (κ below 0.1 where it turns, balls in four-point contact, a hybrid, whose a_ISO is defined for steel only) has all its lives, basic and modified, meet no search requirement or objective, as a minimum, a maximum or an objective, its minimum life cleared or not, and so has the system life with such a member; it says "not determined" without a life check too. The safe side: a hybrid's basic life, or one at a low κ, would stand; in a later release a search requirement on a figure whose own check is cleared is evaluated as that check, each life on its own. A design with a bearing the engine marks outside what it rates (a designation that contradicts it, a slewing ring's proportions) takes no calculated result in a search, its dimensions apart but that bearing's own (inferred from its geometry as modelled), as a design with a check not determined does, whatever its own checks say. A bearing whose designation contradicts it shows only its results' warnings it can vouch for: those at its own path, of the bearing as modelled, are left out (a validation warning at that same path too, such as that its geometry is estimated), its fields' warnings stay. A check not determined is said in a short form wherever its message would be rewritten: what it checks, named from its id, "not determined" and its kind's reason in fixed words, else "(see the check)", never a part of its message, which can hold a name the user wrote and the figures after it; a name that holds an engine mark's words can still pick another kind's reason (no figures). The full engine message, with the reason in context, comes back in a later release, when every display shares one rule. The bearing module's speed chart is not drawn while the bearing's operating nθ is not determined (a rotor's pull, a designation its geometry contradicts), saying why; `axomek speed-limits` then lists the curves' samples without an

operating point. A later release draws the curves apart from that load. A display not yet reviewed may still give such a number beside its check, which says "not determined". A later release routes every display of a check's value through one rule, with a test over every template. Known in 0.3.0, closed in a later release by that one shared display rule: - the Assistant's solve data still carries a check's value and utilisation where the check is not determined (a bearing's thermal speed whose nθ is withheld: its speed and share of the limit, from which the withheld nθ can be worked out); the check is flagged not determined there, the solve tool tells the Assistant never to quote a figure so flagged, and the check's row and every display say "not determined", never a pass; - a rack pinion shaft not rated at its mesh's peak because its only drive is solved for one power direction (the Dual-pinion EPS example's SH5, driven by worm pair WG1) is said, in the verdict's and the report's short reason, to be refused because its torque sources cannot be shared; its check's own note gives the right cause; - a design search takes no result from a design with any check not determined, so a project whose steering duty leaves a warning over its states may find no design meeting a shaft or bearing requirement (the safe side; a later release decides each figure on its own); - `Life0utside::against_bounds` keeps a maximum branch no caller uses (the search reads the hybrid flag directly); a later release removes it; - the Overview's governing check of a module, between two checks not determined, keeps the first in the list (the bearing listed first) where the engine picks by id, so the module tile and the verdict can name different checks, both said not determined; - a variant comparison can mark as best the life, and its check's margin, of a bearing the engine marks unrated (its designation contradicts its geometry), though the bearing's own results say its figures are of the bearing as modelled: compare such a design only after its designation is cleared or corrected; - a comparison of three or more designs can leave out a result row whose neighbouring columns each agree within the tolerance while the first and last do not (10 000, 10 000.8 and 10 001.6 h); `--all` shows it; - a comparison of designs whose life checks all pass on a hybrid's basic life (its modified life outside its rating method, so no best) leaves that check row out of the default CLI comparison and the report, though the minimum lives and the checks' shares differ (a hybrid 6309 at a 1 million h minimum against 4 million h); `--all` shows it. - a hybrid whose modified life fails its minimum keeps that failing check's number in its check row (the fail stands, on the safe side; the number rests on a_ISO, defined for steel bearings only), and so does any life check failing outside its rating method (a steel bearing turning below κ = 0.1, or with its balls in four-point contact): the report's Summary of checks and its Summary page's Governing checks show its share and draw its utilisation bar like any other (a 30207 on 2 / 1 cSt oil against a 20 000 h minimum), though that share cannot be compared with another check's; the governing pick and the orders leave it unranked, and the typed results' report follow-up shows no bar for it. Show the calculation of a hybrid's life explains the modified life, said not determined, not the basic life its passing check goes by; - the Assistant's shortened record of an earlier solve lists a failing life check with its value (a hybrid QJ 207 in four-point contact: 94 476 h against a 1 000 000 h minimum) but without the check's outside-method flag or reason, though the same record says the bearing's life is not determined; the fail stands, and the full solve the Assistant reads carries the flag; - over a duty cycle whose cases stay within the rating method, the Assistant's solve data still gives the nominal summary's modified life where the loads as entered put κ below 0.1 (the Overhung pulley's BR2 on 10 / 2.5 cSt oil at 150 °C nominal, 35 °C in its one case: 107 191 h), with no note that it is outside the method; the duty's life, which the check and every display go by, is right (a nominal summary in four-point contact is marked); - Show the calculation of a check that fails with an unbounded share says it fails "whatever the value of the checked quantity"; for an axial preload that lifts off (a cross-located pair whose housing runs hotter than its shaft) the share is of the lifted-off

state only, and the same check passes once the temperatures no longer lift it; the check's message gives the cause; - "Make it pass" on a check not determined still steers its search by the check's share as modelled (a proposal is made only where the check is determined and passes after the change), and its "tried" list may give that share as the closest reached; - the load-spectrum card's chart still plots a bearing's damage shares where its table withholds them (a case's life outside the rating method with the life check failing); read the table; - the report's summary page labels a passing hybrid's life "modified reference rating life" before the parenthesis that says it is the basic life its check goes by; the bearing section is right; - the agent guide says a hybrid's check `value` is its basic life; where the check fails (`outsideMethod`) it is the modified life it fails on; - a variant comparison of a steel and a hybrid bearing that both pass their life check leaves that row out of the default CLI comparison and the report, though the two columns go by different lives (the row's `labels` say which); `--all` shows it; - the Overview's utilisation chart plots the share of a check failing outside its rating method beside the others, ordered first among the failures but drawn as if comparable; the fail itself is shown; - a hybrid roller bearing on estimated geometry passes on the lower of its two methods' basic lives (correct), but the report's summary page names that life with the other method's label, the Assistant's basis text points at `basicLifeHours` (the picked method's, possibly the higher) instead of the check's value, and a check made not determined by the other method's lower basic life does not quote it; the bearing finder's `basicLifeHours` and the bearing page's basic life rows show the picked method's too; the check's value and status are right; A later release types every result figure as determined or not determined, and every display, ranking and short message reads that type: this closes the ranking, display and short-message limits above for good.

- `axomek check --strict` and `axomek solve --strict` exit 3 for every file with something its method does not assess, which no input clears yet (4 where a check fails and the file was solved completely; a file that was not, an unknown field under `--strict` among it, exits 3 first): a rated key or spline (reversals and load peaks), a rated fit (its shaft seat), a belt or chain drive (its capacity), a fatigue-rated weld (its toes), a yoke on its stop, a support with a designation only and a planetary set without a planet bearing. Check those by hand and use `axomek check` without `--strict` for such files (it still exits 4 on a failing check and 3 on one not determined); a way to mark an item as checked by hand is for a later release.

- A test over every example and five more cases (a hybrid, a four-point contact bearing, a designation its geometry contradicts, the electric motor pulled off centre, a life below κ 0.1 failing beside a determined fail) replaces every figure the engine withholds and checks that no display shows it and no verdict or ranking moves with it (the "poison" harness). What it still finds is listed, each with its reason, in `crates/axm-cli/tests/poison/expected-fail.json`. Beside the limits above it found: the report's appendix of calculations gives a figure not determined as a number (a contradicted bearing's life, the pulled motor's critical speed); `axomek diff --json` and Compare projects give the check and result rows of such a life their numbers without an `outsideMethod` flag; the design search's `--json` gives a candidate's `objective` with no flag of its own where it is such a figure or one not determined (its `undetermined` names the metric; the search does not choose by it); `axomek find-bearing --json` lists a candidate's checks in an order its outside-method life's share moves (the candidates' order does not); and `axomek solve --json` gives `shaft.governingCriticalSpeed` with no flag where the check `minCriticalSpeed` is not determined (read that check's `undetermined`). A bearing's contact stress shows as a number where the static safety that goes by it is not determined: in the Overview's key results, `axomek solve`'s `p max` and the report's Max contact stress (read the static safety's "not determined" beside it). The app's checks panel picks µm or mm for the limit of a length check below 1 mm by the value as modelled, also where the check says "not determined" (no fixture has such a check yet). The harness itself does not tell a leak that moves to the same words on another line, a further difference inside a line whose wording already differs, or a leak moving to another item at the same place of a list from the one its table lists; nor, in the report's decisions, items whose ids are numbers alone ("1", "2").

- An electric motor's critical speed and magnetic stability with a static eccentricity on coupled bearings (review of ut4-motor-critical-speed; see "From 0.3.0"). In 0.3.0: - A pull not along a bearing's other loads (more than 1° apart: a belt or a gear's force across the weight) leaves both not rated: the bearing's load turns as the eccentricity grows, and its stiffness along y or z can be least in between, which neither the states taken nor a bound with each bearing equally stiff every way at its least load bounded (a reviewer's case read 9 988 rpm where a state inside the bound gives 9 604 rpm). This blocks a belt- or gear-driven motor with an eccentricity from the coupled critical speed, and 0.3.0 names no way around it: no stiffness entered on the supports (with coupling off) is known to be safe, as the least one lies somewhere between centred and pulled (keep the eccentricity: its pull counts in the bearing loads and the air gap; 0 would drop it from them). - A state the coupling cannot settle exactly (bearings loose in their clearance, e.g. the Electric motor example without its wave spring, centred) leaves a passing result not decided (a warning). A motor whose bearings carry no load with the rotor centred (the weight off, a vertical motor without preload) has both not rated, with a hint (a horizontal motor: turn the weight on; a vertical one, or one held only by a preload, 0.3.0 cannot rate). The hints of the not decided and not determined results name no stiffness to enter: none is known to be safe there (review: a recipe of the bearings' stiffness without the pull could pass a motor that fails). - A coupled bearing under an axial load (a spring preload, as in the Electric motor example, or the shaft's own axial loads) or preloaded from within (an X or O pair's preload, a negative operating clearance) leaves a passing result not decided (a warning): its radial stiffness need not grow with its radial load, so the states taken bound nothing between them (a reviewer's case: e = 0.2 passed 15 000 rpm at 15 908 rpm where e = 0.15 gives 13 878 rpm). - A spring preload is refused on any shaft joined to another by a bearing between them (shafts solved together do not carry it), also where the spring would act on its own housing. - The unbalance response is the given eccentricity's, with a warning: at a smaller one it can be larger. - Up to four pulling rotors on a shaft; more have both not rated. The axial modes of a state keep the locating bearings' axial stiffness and the housings' seat motions of the pulled state. A critical speed or magnetic stability that is not rated fails, shown as "not rated" in place of its placeholder value (0 rpm, a share of 100 %), and What matters most leaves it out, saying why. Planned for a later release: the rotor pulled within its bearings' clearance to where they hold it (it never sits centred there), in place of the centred state; and a sweep of the eccentricity with a proven bound between its steps, for a pull across the bearings' loads, for bearings under an axial load (a spring preload) and for the unbalance response.
- Speed inputs and variants, for a later release (user test 4 reviews): a speed typed in the base that moves the train's input applies to every variant that does not override it, so a variant with another ratio and no case speeds turns its other shafts by its own ratio from the speed typed (the fan's 900 rpm typed: its motor at 1440 rpm, not the base's 1800), and the message names only the base's speeds, not the variant's. A variant that unmounts the drive or puts it on another shaft is not checked, so after such a move its old input shaft can be left with no speed, its items and bearings at their own. With a variant active, the note under a following shaft's speed still says a speed typed there makes it the input, which the app then refuses with the reason. In the base, adding an item with a speed (a rotor at 900 rpm) is refused whenever a variant exists, as the addition is replayed with each variant active, also one with nothing that conflicts, and the refusal says that variant is active: add it without a speed, then type the speed.

- Steering duty, for a later release (user test ut4): - A worm pair or a gear pair over a steering duty keeps its own required life (a worm pair rated at its worst state as if that lasted its life, a gear pair over the cases at its own life's cycles); the duty's length checks the ball nut's, the ball screws' and the bearings' lives. - Editing a mesh's Peak rack force for static rating in the duty's refusal card makes the duty not computed again, so the card (and the field in it) goes until Compute duty runs; with two refused meshes, the second is edited after recomputing, or on its mesh's own panel. - The drivetrain's natural frequencies (system modes) are dropped as out of date when another shaft is shown, though they do not depend on it; Compute them again. - While the duty computes, a check it decides reads "computing…" in every list of checks the app shows, also another report's (a version compared) of the same kind; the project's verdict line keeps the engine's count (a failure at the state counts) and says the duty is being computed. - A design search's work budget counts a remembered result of the steering duty at every request within one solve, so a design that runs on and on is stopped later than it could be (its time limit and Cancel still stop it); the engine's other remembered results (a gear tooth root's or a housing's finite elements) are not counted, which the budget's floor of 5000 steps covers.
- Make it pass and the gear warnings, for a later release (user test ut4 review): the per-gear face width warning (`gears.<id>.details.faceWidths`) and the tooth contact warning (`gears.<id>.toothContactFaceLoad`) exist only while they warn, so Make it pass and `axomek fix` cannot find a change that clears them (a new face width b clears the per-gear widths, but no trial reads that as passing). Change the field the warning names by hand. The same holds for a bearing's axial direction check (`bearings.<id>.axialDirection`): it exists only while it fails, so Make it pass and `axomek fix` cannot find a change that clears it, and no trial turns a bearing round (Opus review of land-t). Turn the bearing round, or choose one that takes axial load both ways, by hand.
- Bearing loads, for a later release (user test ut4): bearings have no application or shock factor (f_d) of their own; their loads are taken as entered, in the life and the static safety, and the report does not print one. For a machine with shocks, multiply the loads yourself or add a load case at the peak (the bearing help says so).
- Rotating loads, for a later release (user test ut4, Codex review of ut4-small): a bearing's loads are taken as fixed in direction. A load that turns with the shaft (an unbalance, an exciter) has no setting of its own: **Outer ring rotates** gives its load zone and life, but also the cage and element speeds, the friction and the high-speed model's centrifugal and gyroscopic forces of a turning outer ring, which the bearing does not have. The app and the help say to check the life both ways and keep the shorter. A load of unknown direction, or both senses of rotation, is not a study of its own: give each direction worth checking as a load case (a negative factor reverses a load).
- Slewing rings are out of scope (user test ut4): the four-point contact type rates the standard series (QJ) as a rolling bearing with rigid rings; a slewing ring's rings are bolted and flexible, and it is rated by its maker's curves. The app and the help say so, and a four-point contact bearing of a slewing ring's proportions (D_pw/D_w above 15) has its checks not determined (from 0.3.0). A thin-section four-point bearing of such proportions is not determined either (the safe side).
- A belt's shaft load is the running one, its strands less their centrifugal tension F_c; at standstill a belt (friction or timing) pulls with its whole pretension, about 2·F_c more (a fast V-belt in user test RR1: 1.11 kN running, 1.40 kN at standstill). The shafts' and bearings' static checks take the running load times its peak factor, so the standstill pull is not checked; the drives help says so. Static checks at the larger of the two are for a later release (RR1 U-A21).

- Steering misuse events (user test ut4): the misuse events rate the pinions' tooth roots (their contact stress is reported, not checked) and the pinions' shafts statically; the ball nut and its bearing are rated in the steering state and over the duty's strokes, not at a misuse event's peak, and the elements keep their own peak factor for a static peak. A worm pair that drives an assist pinion (a plastic wheel too), the other shafts and the bearings are not rated at a misuse event either, and only the report's method notes say so: the results do not list them as not checked for it. A mesh's peak rack force for static rating is a case of its own, the whole force through that pinion with the assist failed, rated for root and contact, so the same kerb strike can pass as a misuse event and fail there; the checks do not name it as the assist-failed case. Both err on the safe side or are disclosed; listing them on screen is for a later release (RR1 U-A25).
- Keyboard and screen readers, for a later release (user tests ut3 and RR1): the app's screen reader checks read Windows' UI Automation from a real Edge window (`npm run smoke:screen-reader`, on demand, not in CI) and the browser's accessibility tree; no run with NVDA or JAWS themselves has been made, and a check by a person with one is still advised. A check's rule written as a formula in its name (a bearing's minimum load, "k_r 0.025·(ν·n/1000)^(2/3)·(d_m/100)²") is read out symbol by symbol. A press on a fix button while a list shows an option not yet applied applies the list first, and the click is lost: press again once the suggestions are solved. A chart's "Show the values as a table" button shows only with the keyboard focus on it or the pointer over the chart (always shown only where the main pointer is touch: on a touchscreen laptop whose main pointer is a trackpad it starts hidden), so voice control that numbers what is shown may skip it on a computer with a mouse. The edges between the columns have no single-pointer way to resize them other than dragging (the keyboard has its own; WCAG 2.2 2.5.7 asks for one), and their 24 px targets reach from the border into the drawing's and the results' left side, so a control within 24 px of that border is pressed by its edge instead. On a window too narrow for both chosen widths, a column's edge does not go wider than it is shown (narrow the other column first). With a part shown in the Shaft view's results column, the inputs' edge reckons with the results width chosen rather than the part's, so it may stop short of the room there is, and a drag on a window too narrow for both may change the results width kept. A refusal already there as the Shaft page opens (a shaft without supports solved on the Assembly page first) is in the page's status region from the start, so a screen reader need not say it on its own: it is read with the results.
- While an edit solves, the app shows the results of the state before it (another project's never). They are dimmed and marked "Updating…" in the results columns, the drawings and their cards, the Steering and Assembly pages, and a balancing load's torque. Some places are not dimmed: the status marks in a module's list of parts, the figures an editor shows (a linked bearing's solved loads, the shaft's strength panel), the dialogs and the report. The header's "Updating results…" covers those. A solve that lands within 0.15 s is not marked at all, so that the page does not flicker (R-943). The system modes and the steering duty, computed on request, follow the inputs rather than the project: after another project opens they say "Out of date" until computed for it (never counted as computed), and a steering duty computed for a project of the same inputs counts for it. After a licence change while the Overview is open, its power flow can show the plan before's figures unmarked for as long as its own job runs.

- Shaft drawing: a bearing against a shoulder takes its seat class only where the bearing's bore chamfer r_s min is known: ISO 15's diameter series 0, 2 and 3 of single deep groove, self-aligning, four-point and double row angular contact ball bearings and cylindrical and spherical roller bearings. A single angular contact or tapered roller bearing (whose front face's chamfer is smaller and whose facing the model does not tie to the shoulder), needle and thrust bearings and other plans leave that seat plain, with a hint to check the chamfer.
- The 2D shaft view's bearing half-sections, all for a later release: on a rack shaft, the room kept for a bearing's texts is worked out at its support's undisplaced position, so at another rack position a larger seat can bring them closer to the view's edge; below about 132 px of view height there is no room left for those texts and the radial scale jumps to the shaft's own; on a thin shaft a bearing's texts can overlap the load labels next to it; the ground hatch and the texts under a support are centred on the support, not on a wide or off-centre pair; and the view's tests check that a half-section is reused while nothing it is drawn from changes, less that each change draws it anew.
- Searches in a browser: Make it pass, What matters most and the design search run over the pool of engine workers; the bearing finder, sweeps, the preload study and the comparison of variants still run in one worker, one design after another (the desktop app and the command line run them on the pool). A design that runs on and on in a browser has no work budget there (WebAssembly cannot unwind): the search's time budget ends it, its helper stopped, and Cancel does at once.
- A refusal's fix buttons, all for a later release: for an item whose id does not start with a capital letter, a suggestion is found only when it changes that item itself; in a locked project the buttons are disabled without saying why; the item solves a suggestion takes while something is refused are not counted in the solve-work table; the ball screw's lead table follows ISO 3408-1's nominal leads, not yet checked against the standard. A planet bearing whose outside diameter reaches the planet's root diameter is refused and its field marked, with no button (no number to search).
- Spherical roller bearings: the rollers' end and shoulder contact (a guide ring, ribs) and their skew are not modelled; ISO 16281:2025 does not rate them either.
- Four-point contact ball bearings with balls in four-point contact (a radial load) have no rating in ISO 16281:2025: their life is shown and its check is not determined.
- ISO 281 rates a bearing at a 45° contact angle differently as a radial and as a thrust bearing, so lives jump there (ISO 16281:2025, Annex F).
- One-way bearings, for a later release (Opus review of ut4-bearings): a single angular contact ball bearing free to tilt (on a shaft) whose axial load along its facing is below $F_r \cdot \tan \alpha$ (as a floating bearing, or locating with next to no axial load) is rated with its ring tilted to balance it, as the solver finds it, and passes; a real one needs a second bearing or a preload to keep its rings together. Held square, or a tapered roller bearing, it is refused. A thrust ball or roller bearing whose axial load reverses between load cases is rated each way (no field says whether it is single- or double-direction). The axial direction check fails only a load against the facing. On a floating support the bearing finder keeps a one-way candidate's preset facing and does not turn it to face the support's spring preload (`springPreload`): facing against the spring by more than the check's allowance (5 % of $F_r \cdot \tan \alpha$), the candidate fails its axial direction check and is not offered as passing, though turned round it could pass, on the safe side (Opus review of the ut4-motor-critical-speed merge). Turn such a bearing round by hand (`facing`).

- A roller bearing counts as having its geometry entered only when its roller diameter or count and its roller length both differ from the estimate it came with: a maker's value equal to the estimate, or a roller bearing imported before 0.3.0 (whose estimate does not record its roller length) until it is imported again, keeps the lower of the two lives, on the safe side. A catalogue file with the elements but no roller length records the estimated length (`estimate.elementsEntered`), and a bearing whose designation leaves its preset keeps the preset's estimate (each one of a chain of renames, `earlierEstimates`), so neither counts as entered on an estimated roller length. A roller bearing with no estimate known (imported or renamed before 0.3.0 under a name of its own) counts as estimated.
- A ball screw on its shaft (`screws[].mounting`) loads it with the nut at one place (`mounting.position`), not along its stroke; its end fixity is checked against the bearings' places and axial roles, not their tilt stiffness, and its unsupported length is not compared with the bearing span. On the safe side, as the screw's states rate the shaft: a load step is taken as driving (the larger torque), and the screw shaft's fatigue (and that of every shaft joined to it) takes its whole torque and axial force as fully reversed over two or more cases even where they never turn round. Refused rather than combined: a mounted screw with a motion profile (its ramps' bearing damage needs the life integrated over the speed), one with steps of its own in a project with load cases or steering strokes, and a second screw with steps of its own beside one whose steps are the cases (a later release).
- A bearing against a shaft shoulder, or one whose retaining ring groove adjoins its face, gets no seat tolerance on the shaft drawing: the model holds no bearing chamfer r_s to clear the fillet against, and counts the groove as cutting the seat. Every shaft example shows it since their bearings abut their shoulders (a later release).
- Find a bearing and the seat's list of fitting presets keep a support's centre: a wider bearing than one against a shoulder crosses it and is listed as not fitting until the support is moved (a later release).

- A steering's ball nut (`steering.system.assist.ballNut`) is no radial support of the rack itself (0.3.0): the rack needs a support there (a bushing with the nut bearing's stiffness, as the "Belt-driven ball-nut EPS" template has). The torque the nut passes into the rack's thread is not put on the rack (it is held against turning by the pinion's teeth and the yoke, whose ratings leave it out), and the nut's travel is not checked against the threaded length: keep the thread long enough for the nut over the rack's travel. On-centre, the feel's friction torque takes the preload's and the nut bearing's drags but not the motor's, which only the return force counts; it takes the nut's drag with the motor idle (the bearing's friction under that drag at the creep speed), so where the assist curve gives torque within the friction band the band comes out a little wide (a 1000 N preload on the template's nut, before its bearing's drag was counted: 0.601 N·m against 0.591 N·m solved at the band's edge). The steering's balance counts the friction of the bearings linked on the nut's shaft only (as a drag at the nut, see "From 0.3.0"): a bearing linked on another shaft of the steering's drive (the column, the pinion, the motor) is in the power loss but its friction is not in the driver's and the motor's torques, which come out low by it (no example links one). The nut bearing's friction in the balance is its rating's as the solve rates it (its housing's seats moved as the solve moves them); should the two differ, the solve warns at `steering.system.assist.ballNut`. A bearing linked on the nut's shaft whose outer ring sits in another shaft (a bearing between shafts) refuses the steering's state: its friction turns with the two shafts' relative speed and may drive the nut, which the balance does not hold; seat it in the housing, or unlink it to take `bearingDrag`. For a later release. Its thread is taken right-handed (a left-hand one only turns the nut the other way). Without a duty its screw is rated in the steering state alone, so its life is that state's; a steering duty with strokes rates it over the duty's states (0.3.0). The rack's misuse events leave the nut's force out: the meshes hold the event alone, on the safe side for their teeth and the bar between them and the joints, while the nut and its bearing are rated in the state only.
- Modules left out of the verdict (`report.contents.exclude`), until a later release, all on the side of saying more: a verdict whose kept checks pass close to their limits reads `partial`, not `close`; a torsion bar's "not checked" line and the first shaft's issues at `settings.*` or `rack.*` stay in the verdict when steering or the shaft is left out; a module switch's issue count is that module's alone, which can differ from the verdict's while others are off; `axomek variant compare` and the design search (`axomek optimize`) count the checks left out; a matched pair's warning (`pairs.*`) stays in the verdict when its bearing is left out; `axomek tree`'s root has no partial state (the nodes left out say `leftOut`).
- A tandem pair takes ISO 76's double-row static equivalent load (X0 1 and twice the single row's Y0) in Palmgren's load term and its thermally permissible speed, as an O or X pair does. ISO 76 takes a tandem set's two bearings each with the single-row factors at its share of the load, so the pair's P0/C0, friction and heat are overstated and its nθ understated: on the safe side.
- With gravity on, a shaft placed by a bearing in another shaft (seated in it) is refused when the two are solved together and its frame is turned by what places that shaft (a universal joint, a worm pair, a rack mesh): the model of shafts solved together leaves the bearings between them out, and with them that placement (0.3.0). Place the shaft by another connection or at coordinates, or switch gravity off.
- Bearings on their own (none linked to a shaft) have no system life while one of them is "not set up yet": the set's life is not shown rather than shown without it. On the safe side.
- A bearing "not set up yet" makes no check, also when it has limits set (`minLifeHours`, `minStaticSafety`), and the verdict does not name it: it has nothing to rate until it has loads and a speed or a shaft support.

- A new bearing (the app's + Bearing, `addBearing` without loads in its patch) starts with a 3 kN radial starter load (Fy) of its preset, shown and editable in its Loads section and its summary. Without a speed it is then not rated (not "not set up yet") until a speed is entered or the load cleared; on a shaft support the shaft's loads replace it.
- A bearing whose eC·Cu/P stays within about 0.01 of 5 (where aISO leaves its cap of 50) along a whole stroke keeps the duty's search for that regime change refining: one such test case needs some 168 s natively per derivation, so it stops at the work limit and the bearing's checks over the strokes are undecided rather than split at the change. The app computes the duty on request, in a worker of its own with progress and Cancel (0.3.0). The work limits stay as they were: they are counted, not timed, so the app's duty equals `axomek solve`'s, and in the browser's engine (some 1.2 to 1.5 times the native time measured) the tooth rating's limit already takes about a minute and the derivation's some 3 to 4 min, past the minute the on-request duty aims for. Keying the FE root assessment by a rounded lead is not done: it would change the results of duties that finish today, and no bound puts the rounded lead's FE/ISO factor on the safe side. Such a bearing is not decided by a direct bracket either: along a part its eC·Cu/P follows the force one way and the speed (through the viscosity ratio) the other, so it can dip below 5 and back between two samples, which only a search for its least value finds, and that value is not provably single-troughed. Both stay for a later release.
- Over steering-derived load cases a shaft's FKM damage check (with a number of cycles) is undecided where it passes when the shaft's loads follow the speed (0.3.0): every shaft of a mounted stage (a gear or worm pair, a rack mesh, a drive), whose mesh efficiency is rated at each case's speed and enters a member's torque, and whose belt's pull rises with the square of the speed, and every shaft joined to one by a coupling or a bearing between shafts. The cases' chord split bounds that damage only where the stresses follow the force alone. In belt-drive with SH1's FKM over 800000 cycles and one-way strokes over a 126–130 mm/rev lead, the cases passed 6.681 at 6.681179 where the strokes give 6.680947.
- A steering duty whose strokes reverse the rack force (a return pass against a resisting load, a signed load below zero) is refused at `steering.duty` beside anything the load cases reach that a case cannot reverse (0.3.0): a mounted gear or worm pair or rack mesh, a drive, a coupling, a rotor, a bearing with loads of its own, a gear pair rated over the cases. The derived cases reverse only the entered shaft loads and ball screw forces, so the others would be rated at the forward loads on the reversed passes (a rotor's torque, unreversed, cancelled the reversed input in one review's case: 18.444302 where 9.222151 is right). In gearbox-single-stage with a rack mesh and resisting strokes with return passes, BR1 passed 5.5 at 5.778100 where the reversed pass gives 5.260948. One-way strokes are rated; reversing stages over the duty come in a later release.

- A steering duty over a steering system (0.3.0) rates the rack meshes' teeth, and with strokes the system's shafts, the bearings linked on them that take the load cases, its worm pairs and its ball nut over its states (see Steering system). Not rated over the duty (a later release): a mounted gear pair (rated at the system's `state`, its checks not rated over the duty, never a pass: its rating takes its solved shafts' KHβ, which the states do not give it yet), and likewise fits, keys and housings. A rotor's air gap is not rated over the states (its band between them is not drawn; never a pass: rate it at the duty's heaviest state). A bearing's warning over the states (a minimum load undecided at its valley) is never satisfied, and its gyroscopic slip (the high-speed model's) is the system's state's, a pass not rated over the duty. Beside such a duty a design search's metrics are refused but the shaft's mass (their values over the states have no bound between them, and some items' results are the system's state's): search with requireChecks, whose checks take the bands. The ball nut's life is the states' Miner sum (as a bearing's), its static safety the worse of the states' and the system's own state's. Left at the system's `state`: the couplings, a bearing with `useLoadCases` false, and everything at a misuse event (the events rate the pinions only; the elements keep their own `peakFactor` for a static peak, which over the duty multiplies every state's loads, as over load cases). A worm pair and a drive take the worst of their states, each rated as if it lasted their life: on the safe side, as DIN 3996 holds no load spectrum here. A shaft's states are each rated with their stresses fully reversed (as a mounted ball screw's states are), on the safe side where a stroke does not turn round. The band of a bearing's checks is drawn from its life, static safety, contact stress and thermal speed in the states, relative to each check's utilisation. Two supports in seats of one housing (whose seats push each other with the bearings' loads) leave the elements unrated over the duty (a warning at `steering.duty`, their checks never a pass): the states do not couple the seats again (a later release). The project's own load cases beside such a duty rate the shafts and bearings instead (a warning at `steering.duty`), and their checks are then not rated over the duty (never a pass): remove the load cases to rate them over it. The elements' states are some four times the duty points (284 for the Dual-pinion EPS's duty), each a state solve and a solve of every shaft: at most 1200 (`system_cases::CASE_LIMIT`) and at most 12,000 solves (`WORK_LIMIT`: each state one, and one for each shaft and each bearing on a shaft; some 25 s natively and 40 s in the browser), past which the elements are not rated over the duty (a warning at `steering.duty`; their checks never pass) while the teeth are. Measured under load on the build machine (one process, natively): the Dual-pinion EPS's duty 9.3 s for the teeth and 19 s in all; the Belt-driven ball-nut and Column EPS examples given its duty 2.4 s and 2.8 s for the teeth, 7.0 s and 5.2 s in all (168 and 88 states); the browser's engine took about the same under that load (17.9 s, 7.6 s and 6.3 s). It refuses a stroke's `torque` table (the system splits the rack force, not a pinion's torque); a rack mesh off the system's rack is not rated over the duty (a warning; it keeps its rating at its own load). Each duty point is a state solve (some 15 to 30 ms natively), at most 4000 per duty. Rating the teeth of two pinions is the slow part: the Dual-pinion EPS template's duty (a parking, a manoeuvring and an on-centre stroke, both ways) takes some 4.7 s natively and 7.3 s in the browser's engine on the build machine (some 20 s natively on a laptop, 4 s of it the state solves); the app computes it on request, with progress and Cancel (0.3.0), and solves each edit without it. Positions past the rack's travel are refused (each point is a state of the system, within its stops). The lift is rated within 1e-4 mm of the states' between points (the contact ratios within some 4e-5), not raised; the forces are raised to bound theirs. Misuse events are rated at standstill (vehicle speed 0, forward efficiencies, no friction).

- The steering duty rates only the bearings linked to the system's shafts' supports. A support that names a bearing by its designation only links none: nothing rates it, at the system's `state` or over the duty; the duty lists it (`elements.unlinked`) and the summary names it as not assessed. The steering examples draw their bearings so: the Dual-pinion EPS's 6001 and HK2816 on the steering pinion and 6001 and 6203 on the assist pinion's shaft, the Column EPS's and the Steering rack's likewise, every other support of theirs (the column, the intermediate and input shafts, the motors) plain. Only the Belt-driven ball-nut EPS links a bearing, its nut bearing BR1 (see "From 0.3.0"). So the Dual-pinion EPS's duty rates no bearing. The designations are drawn, not sized: linked as drawn (6001 preset, default grease), the Dual-pinion EPS's assist-pinion 6001 at S51 would not carry its parking strokes (static safety some 0.6). Sizing and linking the examples' bearings is for a later release; until then link bearings at those supports to rate them over the duty (a needle bearing such as the HK2816 as a bearing of your own).
- Over a steering system's duty states (0.3.0), all on the side of saying more or of reporting less, for a later release: a rack buckling span that exists in some of a span's states and not in others (the bar moves under the supports) draws no marginal band over that change; a shaft standing still at the system's `state` takes FKM's checks over a number of cycles at the worst state for all its cycles instead of their Miner sum; a state that gives the ball nut no thrust can leave its rating over the duty refused; the system's own state, kept among the worst-case checks, is rated with its mean and amplitude stresses where the duty's states take theirs fully reversed (a state heavier than every duty point is rated more leniently than they are); a refused duty marks every drive's checks, also those of a drive off the steering's train; a bearing's band for its checks other than life and static safety is drawn from those quantities' bands rather than its own; `atState` does not list bearings off the shafts or planet bearings (rated at their own loads), nor the couplings.
- Over steering-derived load cases a linked bearing's minimum load check does not evaluate the axial load's holding (0.3.0): a radial load below the minimum fails there even where the axial load keeps the elements loaded; a preload still holds, and the bearing's own loads keep the exception. Precise handling comes in a later release.
- Over steering-derived load cases the checks that fail at a quantity's low point (valley checks: a linked bearing's minimum load; its gyroscopic slip, which only the high-speed model gives, is undecided with it) are undecided where they pass (0.3.0): the cases bound loads and stresses from above, not their lowest values between the cases. A preload that holds in every case keeps the check, and a module at its own loads is rated as entered. The checks at a peak (stresses, lives, safeties, temperatures, speeds) are bounded. Precise valley bounds come in a later release.
- DIN 732 Table A.1's friction factors are applied to angular contact bearings below 22°, a range the table does not list.
- A shaft–bearing coupling (Couple linked bearings) that does not settle is not rated. A bearing under radial load alone (no axial load) has almost no tilt stiffness inside its clearance, so its tilt has no settled value: the paint mixer of the release-candidate retest (a11y) settles with its paddle hub at 600 N but not at 500 N, where the upper 6206 still carries some 270 N. The refusal names the bearing and says so (from 0.3.2); a coupling that settles such a bearing (its tilt held where its stiffness vanishes) is for 0.3.8. Meanwhile turn Couple linked bearings off, or give that bearing less clearance or a preload.
- Compare variants shows no housing results (a housing wall's stiffness, stresses or modes) for a later release (user test ut4): a variant that changes a housing is compared by its shafts', bearings' and checks' results only.

- A deflection, tilt or critical speed on rigid supports (bearings linked, "Couple linked bearings" off) leaves out the bearings' give; the results, checks and "Show the calculation" say so, and the results offer "Check with the bearings' stiffness". Coupling linked bearings by default is for a later release (user test ut4).
- Messages and refusals, for a later release (RR1 branch 22, what it left): - Validation errors from an element's own engine are placed at the field their message starts with, but about 75 of the project's validation errors still give no hint beyond their message ("must be positive"), and a JSON value of the wrong type in `axomek apply` is refused without its field's name (RR1 TA-F53, agent F5). - A power typed into a torque field is converted at the speed field's value at that moment, said only until it is applied; the torque then stays when the speed changes (RR1 U-A28). A key placed on a shaft keeps its own diameter and torque, and a keyed section can't be swept, optimised or proposed by `fix` (U-A27). "What matters most" can rate a screw's root diameter as not moving its speed check (U-A24, not traced). The planetary refusals and fixes U-C3 lists (a nearby set the search finds but the card does not, a ring shift offered before a planet change, the default planet count beside a typed ring), the tilt note shown above a failing minimum load, a warning's reason shown only on the Overview, "Driven by" offering stages that close a loop, and a single tapered roller bearing's raw "could not reach equilibrium" in the bearing finder. - The Assistant's results give a life in whole hours, as the app shows it, so a life just under its minimum (19 999.6 h against 20 000 h) reads as the minimum there; its check, failing, keeps the exact value. Where one case of a duty cycle puts a bearing outside its rating method (κ below 0.1, balls in four-point contact), the Assistant is told to quote none of its lives, a hybrid's basic life included, not even the modified life at the loads as entered where that one is within its method (the safe side). The room the app makes to keep a field in place is kept through a mouse click on a button in the panel, not yet through a tap on a touch screen (Opus review of RR1 branch 22). - A support's own Designation is shown as its linked bearing's and can't be typed over while a bearing is linked; the value saved in the file stays and comes back when the bearing is unlinked.
- Shoulder heights and the assembly order are not checked: a shaft whose seats have no shoulder for a bearing or a hub to sit against, or with a part that cannot slide on from either end, can pass every check, and Make it pass, the design search and the Assistant's designs can lead to one (user test RR1: the Assistant's lightest shafts, "All 16 checks pass"). The shaft help says so, and the Assistant is told to keep shoulders and a mountable layout. A shoulder (abutment) check and a mountable-from-an-end check are for a later release (RR1 U-A34, decision 4).
- US customary units change the display and the entry only: the bearings are metric (no inch-bore bearings, adapter sleeves, pillow blocks or insert bearings in the presets or the bearing finder), the steels go by their EN names, the keys by DIN 6885-1, and the shaft drawing is metric on an ISO sheet, which its dialog says with US units shown (its fits table gives inches beside the mm). The help (Units, Bearings) says so. Inch drawings, ANSI sheets and US parts are not in 0.3.0 (RR1 decision 5).
- The Contact module's materials have no hardness: the default permissible pressure (ISO 76's 4 200 MPa for a point contact, 4 000 MPa for a line contact) takes both bodies as hardened to bearing hardness, and a passing pressure check does not say so; the help does. A softer body needs **Custom E and ν** with its own permissible pressure. Stating the assumption on the check is for a later release (RR1 U-A37).

- The number format chosen in Settings › General is kept apart from the other settings: Reset to defaults leaves it as it is (choose Automatic to go back), and a Settings window already open in another tab shows the choice it had when it opened (the app uses the newer one the next time it is opened). A number typed in the desktop app while Settings is open and the computer's format is still being read is dropped with a message that can sit behind Settings; close Settings to see it. On a screen narrower than about 330 px the end of "1.234,5 (comma)" is cut off until Settings is scrolled sideways (landing review of numfmt-032, for a later release). With the comma format, an English "1,500" pasted into a field is 1.5, shown as "= 1.5" under it before it applies, not refused. The bearing duty cycle table's cells (speed, torque, temperature, share, factors) put their value back without saying why when a number reads two ways there, as for any text they do not read; the number fields say what to write (numfmt-032, for a later release).
- A QJ 207 linked to no support whose loads are entered by hand as a starter's (Fa 1500 N, Fy 2000 N, the earlier one's, or Fa 3000 N, Fy 1000 N, today's; the rest 0) is warned about as rated on loads nobody entered; the warning is advice only. The test that every example and Add choice solves determined (R-956) solves Add Planetary on a blank project, not as a stage driven by another; does not fail on a bearing row left not set up (no example has one); and checks that each Add choice the app sends is covered, not that a covered choice's patch matches the app's.

Hertz contact and subsurface stress

Point and line contacts by Hertz’s theory, and the stresses below the surface.

Methods and standards

Hertzian contacts of two bodies on their own: the principal radii of each body (sphere, cylinder, plane, groove with its conformity, torus, crowned roller or general radii, positive convex and negative concave) with their principal planes at any angle give the relative curvature as the sum of the bodies' curvature tensors, its sum Σρ, difference F(ρ) = cos τ (the auxiliary angle) and the orientation of the contact ellipse.

Point contacts after Hertz: the ellipticity k = a/b solved from F(ρ) = ((k² + 1)·E − 2K)/((k² − 1)·E) with the complete elliptic integrals K and E computed exactly by the arithmetic–geometric mean, the semi-axes a = (2k²E/π)^(1/3)·(3F·η/(2Σρ))^(1/3) and b = (2E/(πk))^(1/3)·(3F·η/(2Σρ))^(1/3), the peak pressure p0 = 3F/(2πab), the approach δ = (2K/π)·(π/(2k²E))^(1/3)·(3F·η/(2Σρ))^(2/3)·Σρ/2 and the stiffness dF/dδ = 1.5·F/δ, with η = (1 − ν_1²)/E_1 + (1 − ν_2²)/E_2.

A cylinder or crowned roller against a body straight along its axis is a line contact, sliced into laminae as ISO 16281:2025 slices a roller: each lamina carries cs·(δ − gap)^(10/9) with the standard's constant 35 948·L^(8/9)/n (Formulae (43), (45), (46)) taken for one contact (2^(10/9) times the roller's two contacts in series) and scaled to the materials by the elastic constants, the gap from the crown (a circular arc of the given drop, the logarithmic profile k·D·ln(1/(1 − (2y/L)²)) or the reference profile of ISO 16281:2025, B.5, Formulae (B.7)–(B.9)) and the tilt, the optional edge stress concentration of C.2, Formula (C.4), and each lamina's pressure and half-width by Hertz's line contact, b = √(4q·η/(πL·Σρ)) and p0 = 2q/(πbL).

The stresses below the surface: in a line contact McEwen's plane strain solution and, for a sliding traction μ·p along x, the Smith–Liu solution; below an ellipse Boussinesq's and Cerruti's potentials of the ellipsoidal pressure, their derivatives in Carlson's integral R_D of the ellipsoidal coordinate and one integral over it (Gauss–Legendre), exact anywhere below the surface and checked against the point-force solutions summed over the ellipse; the pressure taken as unaffected by the traction (exact for bodies of one material).

The peaks of the maximum shear stress (Tresca), the von Mises stress and the orthogonal shear stresses τxz (Lundberg–Palmgren's closed form without traction) and τyz with their depths, the stresses down the contact axis, stress maps across (xz) and along (yz) the contact, the curves of pressure, semi-axes, approach and stiffness against load; checked against a permissible peak pressure of 4200 MPa for point and 4000 MPa for line contacts as ISO 76 bases its static load ratings on them (or as given) and an optional shear limit; suggested hardened case depth 3 × the depth of the maximum shear stress (rule of thumb); Hertz theory assumes elastic, smooth bodies and a contact small against their radii, which the results note when the semi-axis exceeds a tenth of the smallest radius.

Subsurface stress: Hertz contact without friction, stress field below the contact centre by numerical integration (ellipse) or closed form (line contact); suggested case depth 3 × depth of the maximum shear stress (rule of thumb).

Validation

44 of 44 cases within their tolerance, in 3 groups.

Closed-form solutions

Hertz contact module

Kind of reference	Closed-form solutions
Implements	Hertz contact; ISO 16281:2025, Formula (43)
Reference source	K. L. Johnson, Contact Mechanics (1985), §3.4, §4.2, §4.3, Eq. (3.63)
Cases	26 of 26 within tolerance

Point and line contacts of the contact module against Hertz's closed forms (K. L. Johnson, Contact Mechanics, 1985, §4.2 and §4.3), the ellipse relation B/A = ((a/b)²·E(e) − K(e))/(K(e) − E(e)) of §4.2 with K and E by independent quadrature, the bearing solver's own contact (its Brent root and Carlson integrals) for a ball in a groove, the classical stress peaks below circular and line contacts (Johnson Eq. 3.63, §4.2), the surface stress of a sliding line contact (Smith–Liu: 2μp0 at the trailing edge) and the ISO 16281:2025 load–deflection of a line contact (Formula (43), one of a roller's two contacts). Steel E 210 000 MPa, ν 0.3 unless stated.

Case	Reference	Axomek	Deviation	Tolerance	
Sphere R 6 on a plane, 2000 N: a	0.4272659 mm	0.4272659 mm	0 %	10^{-9}	**pass**
Sphere on a plane: approach δ = a²/R	0.03042602 mm	0.03042602 mm	1.1×10^{-16}	10^{-9}	**pass**
Sphere on a plane: p0 = 3F/(2πa²)	5230.883 MPa	5230.883 MPa	0 %	10^{-9}	**pass**
Sphere on a plane: stiffness 1.5·F/δ	98599.82 N/mm	98599.82 N/mm	1.5×10^{-16}	10^{-9}	**pass**
Steel sphere R 5 on ceramic sphere R 15, 1000 N: a	0.2741237 mm	0.2741237 mm	0 %	10^{-9}	**pass**
Steel on ceramic spheres: p0	6354.016 MPa	6354.016 MPa	0 %	10^{-9}	**pass**
Crossed cylinders R 10 at 90°, 1500 N: a = b	0.4602582 mm	0.4602582 mm	0 %	10^{-9}	**pass**
Crossed cylinders R 10 at 90°: ellipticity 1	1	1	0 %	10^{-9}	**pass**
Crossed cylinders R 10 and R 40: B/A from the ellipse = R2/R1	4	4	8.9×10^{-16}	10^{-8}	**pass**
Crossed cylinders R 10 and R 40: K(e) against quadrature	2.359524	2.359524	9.4×10^{-16}	10^{-10}	**pass**
Crossed cylinders R 10 and R 40: approach δ = 3F·K(e)/(2π·a·E*)	0.01638169 mm	0.01638169 mm	1.1×10^{-15}	10^{-9}	**pass**
Cylinders R 8 and R 12, L 20, 15 kN: half-width b	0.1993115 mm	0.1993115 mm	2.8×10^{-16}	10^{-9}	**pass**
Parallel cylinders: p0 = √(F·Σρ/(π·L·η))	2395.571 MPa	2395.571 MPa	3.8×10^{-16}	10^{-9}	**pass**
Line contact: τ_max/p0 = 0.300	0.3	0.3002831	0.094 %	0.5 %	**pass**
Line contact: depth of τ_max z/b = 0.786	0.786	0.7861514	0.019 %	0.5 %	**pass**
Line contact: τxz/p0 = 0.250 at z/b = 0.5	0.25	0.25	0 %	10^{-6}	**pass**
Line contact approach: half the roller's ISO 16281:2025 deflection (Q/(35 948·L^(8/9)))^0.9	0.02072626 mm	0.02072626 mm	3.3×10^{-16}	10^{-9}	**pass**
Line contact, μ 0.25: σx at the trailing edge = 2μ·p0	1197.786 MPa	1197.786 MPa	0 %	10^{-9}	**pass**

Case	Reference	Axomek	Deviation	Tolerance	
Circular contact, ν 0.3: τ_max/p0 (Eq. 3.63; ≈ 0.31 at ν 0.3)	0.3100205	0.3100204	2.6×10^{-7}	10^{-6}	**pass**
Circular contact, ν 0.3: depth of τ_max z/a (≈ 0.48 at ν 0.3)	0.48086	0.48125	0.081 %	0.5 %	**pass**
Circular contact, ν 0.25: τ_max/p0 (Eq. 3.63; ≈ 0.31 at ν 0.3)	0.3216784	0.3216784	1.1×10^{-7}	10^{-6}	**pass**
Circular contact, ν 0.25: depth of τ_max z/a (≈ 0.48 at ν 0.3)	0.46431	0.4640625	0.053 %	0.5 %	**pass**
Ellipse of k > 1000: b against the line contact of its middle	0.04611211 mm	0.04611201 mm	2.3×10^{-6}	0.01 %	**pass**
Ball Ø10 in a 0.52 groove of a deep groove bearing: p0 as the bearing solver	3527.782 MPa	3527.782 MPa	1.6×10^{-14}	10^{-6}	**pass**
Ball in the groove: semi-major axis a as the bearing solver	2.231883 mm	2.231883 mm	3.4×10^{-14}	10^{-6}	**pass**
Ball in the groove: semi-minor axis b as the bearing solver	0.2363774 mm	0.2363774 mm	1.8×10^{-14}	10^{-6}	**pass**

Hertz contact: subsurface stress

Kind of reference	Closed-form solutions
Implements	Hertz contact, subsurface stress field
Reference source	K. L. Johnson, Contact Mechanics (1985), §3.4 and §4.2
Cases	11 of 11 within tolerance

Stresses below frictionless Hertz contacts against K. L. Johnson, Contact Mechanics (1985), §3.4 and §4.2; ν = 0.3, p0 = 1.

Case	Reference	Axomek	Deviation	Tolerance	
Line contact: max shear τ1/p0 = 0.300	0.3	0.3002831	0.094 %	0.5 %	**pass**
Line contact: depth of max shear z/b = 0.786	0.786	0.7861514	0.019 %	0.5 %	**pass**
Line contact: orthogonal shear τxz/p0 = 0.250	0.25	0.25	0 %	0.5 %	**pass**
Line contact: depth of τxz max z/b = 0.500	0.5	0.5	0 %	1 %	**pass**
Circular contact at z = a: σz/p0 = −1/2 (Eq. 3.63)	-0.5	-0.5	0 %	0.1 %	**pass**
Circular contact at z = a: σr/p0 (Eq. 3.63)	-0.02898239	-0.02898239	1.3×10^{-15}	0.5 %	**pass**
Circular contact: max shear τ1/p0 ≈ 0.31	0.31	0.3100205	0.0066 %	2 %	**pass**
Circular contact: depth of max shear z/a ≈ 0.48	0.48	0.4808645	0.18 %	2 %	**pass**
Circular contact, surface centre: σr/p0 = −(1 + 2ν)/2 (Eq. 3.41)	-0.8	-0.8	2.6×10^{-10}	0.1 %	**pass**
Circular contact, surface at r = a: σr/p0 = (1 − 2ν)/3 (Eq. 3.42)	0.1333333	0.1333222	0.0083 %	0.1 %	**pass**
Line contact: von Mises σ_v/p0 ≈ 0.557 on the axis (Eq. 4.49)	0.557	0.5575162	0.093 %	0.2 %	**pass**

Subsurface stress map below a long contact ellipse

Kind of reference	Closed-form solutions
Implements	Boussinesq's potentials for the ellipsoidal Hertz pressure (Johnson, Contact Mechanics, §3.2, §3.6)
Reference source	McEwen's line contact (Johnson, Contact Mechanics, Eq. 4.47, plane strain), which a long ellipse's rolling plane approaches, over the app's map grid (81 × 49 points, ±2.5·b, 3·b deep) at a/b 400; the exact ellipse field at audit RR1-S-06's points at a/b 150
Cases	7 of 7 within tolerance

p0 2000 MPa, b 0.01 mm, ν 0.3; within 1 % of p0 of the line over the whole map.

Case	Reference	Axomek	Deviation	Tolerance	
a/b 150, x = 0, z = 0.1875 b: von Mises	846.2317 MPa	846.2317 MPa	4.67×10^{-7} MPa	0.001 MPa	**pass**
a/b 150, x = 0, z = 0.375 b: von Mises	1008.527 MPa	1008.527 MPa	9.618×10^{-8} MPa	0.001 MPa	**pass**
a/b 400 against the line contact, largest difference of von Mises: x 0 b, z 0.0625 b	778.4943 MPa	776.0681 MPa	2.426212 MPa	20 MPa	**pass**
a/b 400 against the line contact, largest difference of τ_max: x 0.8750000000000001 b, z 0.0625 b	282.7471 MPa	281.4071 MPa	1.339932 MPa	20 MPa	**pass**
a/b 400 against the line contact, largest difference of τxz: x 2.5 b, z 3 b	-189.6331 MPa	-189.6243 MPa	0.008802997 MPa	20 MPa	**pass**
a/b 400 against the line contact, largest difference of σx: x 1.4375000000000004 b, z 0.0625 b	-96.82753 MPa	-94.83182 MPa	1.99571 MPa	20 MPa	**pass**
a/b 400 against the line contact, largest difference of σz: x 0 b, z 3 b	-632.4555 MPa	-632.4377 MPa	0.01778706 MPa	20 MPa	**pass**

Stated limits

Standards and scope **(help: modules/contact.md)**

Point contacts follow Hertz's theory with exact elliptic integrals; line contacts are sliced like an ISO 16281 roller, so crowned and tilted rollers carry an uneven line load. The stresses below the surface are exact for frictionless contact and include the surface traction when a traction coefficient is given: below an ellipse from Cerruti's solution in either direction, below a line from the Smith–Liu solution across it and the anti-plane shear along it.

Traction on unlike materials. The traction is laid on the Hertz pressure without changing it. That is exact when both bodies have the same elastic constants. For unlike ones a tangential traction also shifts the pressure and the contact (the coupling Dundurs' second parameter β measures, from 0 for alike bodies to about 0.29 for a rigid body on steel), which this model leaves out. When |β| is above 0.1 with a traction, a note says the traction's stresses are

approximate: steel on aluminium (β about −0.12), grey cast iron (about −0.12) or tungsten carbide (about 0.11). Steel on Si_3N_4 (about 0.04), bronze (about −0.07) or stainless steel (about −0.02) stays below it.

Not covered: the coupling of traction and pressure on unlike materials (only noted), plastic deformation and residual stresses from hardening. The edge loading of a point contact is only warned of (see Results).

The materials carry no hardness: the default permissible pressure takes both bodies as hardened through to bearing hardness, and a passing pressure check does not repeat that. For a softer surface (a pin of unhardened steel, or a thin case), choose **Custom E and ν** and enter the permissible pressure of its material and hardening.

Limits **(help: theory/hertz-contact.md)**

- Frictionless contact: surface traction, which moves the maximum shear towards the surface at high friction, is not included.
- No residual stresses from hardening or from the fit, no hoop stress.
- Smooth surfaces: roughness peaks and dents from particles (the subject of e_C in ISO 281) are not resolved.
- Elastic material: above the yield strength, plastic flow changes the field; the static safety keeps the contact below the ISO 76 limit.

Known limits

- A Hertz line contact loaded to the ends of its shorter cylinder without `edgeConcentration` passes its pressure and shear checks on the laminae's pressure, with a warning: the end faces raise the pressure there, which only the ISO 16281 edge factor (C.2, or a crown) takes. A point contact cut off by a roller's ends fails its checks (0.3.0); the line contact's edge rating comes in a later release. Turn on `edgeConcentration` or crown the roller.

Lubrication and friction

Oil viscosity over temperature, the viscosity ratio κ and the friction and power loss of bearings.

Methods and standards

Lubrication: kinematic viscosity at the operating temperature from ν_40 and ν100 by the Walther equation (ASTM D341); minimum and central film thickness after Hamrock–Dowson (balls) and Pan–Hamrock (rollers, mean load per unit length), reduced for inlet shear heating by φT = 3.94/(3.94 + L^0.62), L = η0·β·u²/k (Murch & Wilson, β from the Walther curve) and, when an oil supply layer is given, for starvation by φS = R/(1 + R^1.7)^(1/1.7), R = supply layer / flooded central film (Chevalier et al., point contacts); film parameter λ against the composite RMS roughness; friction torque after Palmgren, M = M0 + M1 with f0 by bearing type and lubrication method and f1·P1 from the loads (f0, f1 and P1 after DIN 732:2010 / DIN ISO 15312:2004, Table A.1, for each type's common series; the νn < 2000 branch M0 = 160·10⁻⁷·f0·dm³ after Palmgren), or (Friction model: Four-component model) after the four-component model M = φish·φrs·Mrr + Msl + Mseal + Mdrag: Mrr = Grr·(ν·n)^0.6 and Msl = Gsl·μsl with Grr and Gsl from each bearing's loads (the resultant of its element forces), dm and representative series constants R1–R4, S1–S4 (62, 72 B, 33 A, 12, QJ 2, 511, NU 2, 811, 302, 222; double-row deep groove and cylindrical bearings as two rows sharing the load), inlet shear heating φish = 1/(1 + 1.84·10⁻⁹·(n·dm)^1.28·ν^0.64), kinematic replenishment φrs = exp(−Krs·ν·n·(d + D)·√(Kz/(2(D − d)))) with Krs 3·10⁻⁸ oil and 6·10⁻⁸ grease, μsl = φbl·0.12 + (1 − φbl)·μEHL with φbl = exp(−2.6·10⁻⁸·(n·ν)^1.4·dm) and μEHL 0.05 (0.02 cylindrical, 0.002 tapered), the given seal torque and, in an oil bath to the centre of the lowest element, the viscous drag 1.093·10⁻⁷·n²·dm³·(n·dm²·ft/ν)^−1.379·Rs (the churning term VM is not included).

The ISO 281 viscosity ratio κ takes the operating viscosity from the Walther curve, as catalogue viscosity charts do, unless a bearing selects the original solver's straight line between 40 and 100 °C (Viscosity at operating temperature: Linear between 40 and 100 °C; the default under rules "original").

With a heat balance the operating temperature is where the friction power equals the heat removed through housing and shaft, K·(T − T_ambient) with K from the ISO 15312 reference heat flow (q_r = 0.016 W/mm² on A_r = π·B·(D + d) at 50 K for radial bearings, reduced above 50 000 mm²) unless given, plus 28.5 W per l/min (DIN 732, Eq. (12)) and K of circulating oil.

Thermal speeds: the speed at which π·n/(30·10³)·[10⁻⁷·f0·(ν·n)^(2/3)·dm³ + f1·P1·dm] equals the heat given off, dm = (D + d)/2; the thermal reference speed n_θr by DIN ISO 15312:2004 under its reference conditions (outer ring 70 °C in 20 °C, P1r = 0.05·C_0 radial or 0.02·C_0 thrust, νr = 12 or 24 mm²/s, f0r and f1r of its Table A.1 by type and dimension series from the designation, else the type's common series, Φr = qr·Ar), the thermally permissible operating speed nθ by DIN 732:2010 at the permissible outer ring temperature (the bearing's Permissible temperature, else 70 °C) with the operating viscosity there, f0 (times the lubrication method's factor) and f1·P1 of its Table A.1 (f1 with P0/C_0), the seal torque, and the heat removed through the seats (the heat balance's heat transfer, else the ISO 15312 reference) and by circulating oil, solved exactly for fn = nθ/n_θr in KL·fn^(5/3) + KP·fn = 1 (the approximation of its Eq. (3) reported alongside); a pair of angular contact ball bearings takes the paired rows; thrust ball, double-row deep groove and double-row caged cylindrical roller bearings are not covered.

The operating speed is checked against nθ, a warning unless a permissible temperature is set.

Over a duty cycle each load case's speed is checked against the nθ under that case's loads (an unloaded case that turns with the friction of no load), the worst case making the check.

Power loss: bearing friction after Palmgren (M0 from f0·(ν·n)^(2/3)·dm³, M1 = f1·P1·dm, by bearing type and lubrication method) plus a given seal torque; gear mesh losses after ISO/TR 14179-2 (Niemann and Schlenk), P = P_in·μmz·HV with μmz = 0.048·(Fbt/b/(vΣC·ρredC))^0.2·η^−0.05·Ra^0.25 (mineral oil, η = 0.88·ν at the pair's oil temperature, ν on the Walther line (ASTM D341) of the oil's viscosity index (95 unless given), at a

load case's temperature in that case for a mounted pair that takes the cases', at 40 °C when none is given, Ra = R_z/6; at the working pitch circle: Fbt = Ftw/cos αwt = Ft/cos αt, vΣC = 2·vtw·sin αwt) and HV from the addendum contact ratios; a mounted wheel that drives loses P1·(1/η − 1), in the pair's own mesh loss as in the power loss.

The overall efficiency is 1 − the total loss over the power into the gear train (the driving members of the pairs and drives whose shaft no other pair or drive drives).

No-load gear losses (churning, windage) and seal losses not entered are not included.

Validation

4 of 4 cases within their tolerance, in 1 groups.

Worked examples of standards and books

Viscosity index of an oil

Kind of reference	Worked examples of standards and books
Implements	ISO 2909:2002, methods A and B, Table 1
Reference source	ISO 2909:2002, the worked examples of methods A (ν_40 73.30, ν100 8.86 mm²/s) and B (ν_40 22.83, ν100 5.05 mm²/s)
Cases	4 of 4 within tolerance

The index is printed as a whole number: Axomek's is compared within half a unit.

Case	Reference	Axomek	Deviation	Tolerance	
Method A: L at ν100 8.86 mm²/s (Table 1)	119.94 mm²/s	119.94 mm²/s	1.4211×10^{-14} mm²/s	0.005 mm²/s	**pass**
Method A: H at ν100 8.86 mm²/s (Table 1)	69.48 mm²/s	69.48 mm²/s	1.4211×10^{-14} mm²/s	0.005 mm²/s	**pass**
Method A: viscosity index	92	92.42965	0.4296473	0.5	**pass**
Method B: viscosity index	156	156.4235	0.4234829	0.5	**pass**

Not validated against a published reference

- Film thickness and the viscosity ratio κ
- The four-component bearing friction and oil drag

Stated limits

What is not covered **(help: theory/lubrication.md)**

- Non-Newtonian behaviour and the limiting shear stress of the oil, which affect traction rather than film thickness.
- Roughness effects on the film itself (micro-EHL); λ treats the surfaces statistically.
- Grease ageing and bleeding apart from the relubrication rule and the supply layer you enter.
- The EP-additive allowance of ISO 281, 9.3.3.3.4.
- Water in the oil.

Known limits

- A gear pair's mesh loss (ISO/TR 14179-2) takes the oil's viscosity at the pair's `oilTemperature`; a pair without one (every pair saved before it was an input) takes the oil at 40 °C, so in a hot sump its mesh loss comes out about 11 % low (ISO VG 220 at 95 °C): the efficiency slightly high, the heat slightly low. Give the oil's working temperature. The formula's uncertainty from the flank roughness is larger.
- A gear oil under about 6.5 mm²/s at 40 °C has no viscosity–temperature line at viscosity index 95 (ISO 2909): its scuffing check is refused at `viscosityIndex`, and with it the pair's curves, tooth contact and root FE, until the check is removed or the oil's own higher index is given. No gear oil is that thin; a later release keeps the other analyses.
- A bearing whose eC·Cu/P stays within about 0.01 of 5 (where aISO leaves its cap of 50) along a whole stroke keeps the duty's search for that regime change refining: one such test case needs some 168 s natively per derivation, so it stops at the work limit and the bearing's checks over the strokes are undecided rather than split at the change. The app computes the duty on request, in a worker of its own with progress and Cancel (0.3.0). The work limits stay as they were: they are counted, not timed, so the app's duty equals `axomek solve`'s, and in the browser's engine (some 1.2 to 1.5 times the native time measured) the tooth rating's limit already takes about a minute and the derivation's some 3 to 4 min, past the minute the on-request duty aims for. Keying the FE root assessment by a rounded lead is not done: it would change the results of duties that finish today, and no bound puts the rounded lead's FE/ISO factor on the safe side. Such a bearing is not decided by a direct bracket either: along a part its eC·Cu/P follows the force one way and the speed (through the viscosity ratio) the other, so it can dip below 5 and back between two samples, which only a search for its least value finds, and that value is not provably single-troughed. Both stay for a later release.

Limits, fits and press fits

ISO 286 limit deviations, ISO 492 bearing tolerances, the fits of bearing seats and press fits by DIN 7190.

Methods and standards

Fits and press fits: ISO 286 limit deviations; elastic press fit after DIN 7190-1 (plane stress, thick-walled cylinders, different materials), interference lost to smoothing 0.8·(Rz_A + Rz_I), capacity from the smallest and stresses from the largest interference, shrink temperature with an assembly clearance of 0.001·d; at operating temperatures of shaft and hub the interference change d·(α_shaft·ΔT_shaft − α_hub·ΔT_hub) from the assembly temperature, the capacity from the smaller and the stresses from the larger interference of the assembly and operating state; turning, the interference lost to the hub growing more than the shaft (rotating thick-walled cylinders, or a linked housing seat's FE centrifugal growth) and the hub bore's centrifugal hoop stress E·u/a added to the hub stress at the pressure left turning.

The hub is checked with DIN 7190's von Mises stress $\sqrt{3}\cdot p/(1 - Q^2)$ at the bore; the exact plane-stress value $p\cdot\sqrt{(3 + Q^4)}/(1 - Q^2)$ (up to 2 % higher at Q = 0.5) is reported alongside.

Validation

177 of 177 cases within their tolerance, in 7 groups.

Worked examples of standards and books

Press fits: interference, smoothing, joint pressure and the shrinking temperature

Kind of reference	Worked examples of standards and books
Implements	DIN 7190-1:2017 (elastic cylindrical press fits: smoothing 0.8·(Rz_Ai + Rz_Ia), joint pressure p = Z·E_A/(D_F·K), joining temperature); ISO 286-2 for the fits' deviations
Reference source	Roloff/Matek Maschinenelemente, 21st ed. (2013), example 12.1 (pp. 413 to 414); Roloff/Matek Maschinenelemente Aufgabensammlung, 15th ed. (2010), problem 12.8 (p. 82, solution pp. 297 to 298)
Cases	7 of 7 within tolerance

Example 12.1 picks H6/r6 so that its least interference is exactly Z_k + G = 14 + 10 µm, the interference its least joint pressure p_Fk ≈ 9.7 MPa needs: the engine's least pressure of that fit is compared with p_Fk. The book rounds the smoothing G to whole µm (10.08 to 10 µm in 12.1, 5.2 to 5 µm in 12.8); in 12.8 that is worth 0.35 % of the pressure (0.05 MPa), which that case's tolerance adds to the printed precision. Example 12.1's joining temperature is not compared: the book takes a joining clearance of half the interference, the engine 0.001·D_F.

Case	Reference	Axomek	Deviation	Tolerance	
Example 12.1: least interference Ü_u	24 µm	24 µm	3.5527×10^{-15} µm	0.5 µm	**pass**
Example 12.1: largest interference Ü′_o	62 µm	62 µm	0 µm	0.5 µm	**pass**
Example 12.1: smoothing G	10 µm	10.08 µm	0.08 µm	0.5 µm	**pass**
Example 12.1: least joint pressure (p_Fk)	9.7 MPa	9.693313 MPa	0.006687327 MPa	0.05 MPa	**pass**
12.8: largest interference Ü′_o	62 µm	62 µm	0 µm	0.5 µm	**pass**
12.8: smoothing G	5 µm	5.2 µm	0.2 µm	0.5 µm	**pass**

Case	Reference	Axomek	Deviation	Tolerance	
12.8: largest joint pressure p′_Fg	13.5 MPa	13.41381 MPa	0.08618971 MPa	0.1 MPa	pass

Standards’ tables and printed values

Limits and fits: ISO 286 and ISO 492

Kind of reference	Standards’ tables and printed values
Implements	ISO 286-1/-2:2010; ISO 492:2014
Reference source	ISO 286-2:2010 tables of limit deviations; ISO 492:2014 normal tolerance class
Cases	28 of 28 within tolerance

Limit deviations in µm against the ISO 286-2 tables, and ISO 492 normal-class bearing tolerances.

Case	Reference	Axomek	Deviation	Tolerance	
k5 at Ø35: upper	13 µm	13 µm	0 %	exact	pass
k5 at Ø35: lower	2 µm	2 µm	0 %	exact	pass
k6 at Ø35: upper	18 µm	18 µm	0 %	exact	pass
k6 at Ø35: lower	2 µm	2 µm	0 %	exact	pass
m6 at Ø60: upper	30 µm	30 µm	0 %	exact	pass
m6 at Ø60: lower	11 µm	11 µm	0 %	exact	pass
n6 at Ø100: upper	45 µm	45 µm	0 %	exact	pass
n6 at Ø100: lower	23 µm	23 µm	0 %	exact	pass
p6 at Ø40: upper	42 µm	42 µm	0 %	exact	pass
p6 at Ø40: lower	26 µm	26 µm	0 %	exact	pass
g6 at Ø25: upper	-7 µm	-7 µm	0 %	exact	pass
g6 at Ø25: lower	-20 µm	-20 µm	0 %	exact	pass
f6 at Ø50: upper	-25 µm	-25 µm	0 %	exact	pass
f6 at Ø50: lower	-41 µm	-41 µm	0 %	exact	pass
H7 at Ø72: upper	30 µm	30 µm	0 %	exact	pass
H7 at Ø72: lower	0 µm	0 µm	0 µm	exact	pass
K7 at Ø40: upper	7 µm	7 µm	0 %	exact	pass
K7 at Ø40: lower	-18 µm	-18 µm	0 %	exact	pass
M7 at Ø40: upper	0 µm	0 µm	0 µm	exact	pass
M7 at Ø40: lower	-25 µm	-25 µm	0 %	exact	pass
N7 at Ø40: upper	-8 µm	-8 µm	0 %	exact	pass
N7 at Ø40: lower	-33 µm	-33 µm	0 %	exact	pass
P7 at Ø40: upper	-17 µm	-17 µm	0 %	exact	pass
P7 at Ø40: lower	-42 µm	-42 µm	0 %	exact	pass
M6 at Ø280: upper	-9 µm	-9 µm	0 %	exact	pass
M6 at Ø280: lower	-41 µm	-41 µm	0 %	exact	pass
Bearing bore Ø35, normal class: lower	-12 µm	-12 µm	0 %	exact	pass
Bearing outside Ø72, normal class: lower	-13 µm	-13 µm	0 %	exact	pass

Press fits: a steel's yield strength at size, at most its size class's minimum

Kind of reference	Standards’ tables and printed values
Implements	DIN 743-2:2012, 6.2, eqs. (11) and (14); the steels' guaranteed minimum yield strengths by size
Reference source	EN 10083-2:2006, Table 9 (p. 20): C45 +QT; EN 10083-3:2006 as printed in Decker, Maschinenelemente, 21st ed. (2023), Table 2.5: 34CrMo4, 42CrMo4, 34CrNiMo6 and 30CrNiMo8 +QT up to 160 mm; EN 10025-2:2019, Table 6 (p. 27): S235 and S355 up to 250 mm
Cases	105 of 105 within tolerance

Each class sampled just above its lower end, at its middle and at its upper end. The reference is the lower of K1 times the list's 16 mm value and the class's minimum, which the yield strength taken must equal; it is never above the class's minimum.

Case	Reference	Axomek	Deviation	Tolerance	
C45 +QT at Ø0.5	490 MPa	490 MPa	0 %	10^{-12}	pass
C45 +QT at Ø8	490 MPa	490 MPa	0 %	10^{-12}	pass
C45 +QT at Ø16	490 MPa	490 MPa	0 %	10^{-12}	pass
C45 +QT at Ø16.5	430 MPa	430 MPa	0 %	10^{-12}	pass
C45 +QT at Ø28	430 MPa	430 MPa	0 %	10^{-12}	pass
C45 +QT at Ø40	423.7032 MPa	423.7032 MPa	0 %	10^{-12}	pass
C45 +QT at Ø40.5	370 MPa	370 MPa	0 %	10^{-12}	pass
C45 +QT at Ø70	370 MPa	370 MPa	0 %	10^{-12}	pass
C45 +QT at Ø100	357.4064 MPa	357.4064 MPa	0 %	10^{-12}	pass
34CrMo4 +QT at Ø0.5	800 MPa	800 MPa	0 %	10^{-12}	pass
34CrMo4 +QT at Ø8	800 MPa	800 MPa	0 %	10^{-12}	pass
34CrMo4 +QT at Ø16	800 MPa	800 MPa	0 %	10^{-12}	pass
34CrMo4 +QT at Ø16.5	650 MPa	650 MPa	0 %	10^{-12}	pass
34CrMo4 +QT at Ø28	650 MPa	650 MPa	0 %	10^{-12}	pass
34CrMo4 +QT at Ø40	650 MPa	650 MPa	0 %	10^{-12}	pass
34CrMo4 +QT at Ø40.5	550 MPa	550 MPa	0 %	10^{-12}	pass
34CrMo4 +QT at Ø70	550 MPa	550 MPa	0 %	10^{-12}	pass
34CrMo4 +QT at Ø100	550 MPa	550 MPa	0 %	10^{-12}	pass
34CrMo4 +QT at Ø100.5	500 MPa	500 MPa	0 %	10^{-12}	pass
34CrMo4 +QT at Ø130	500 MPa	500 MPa	0 %	10^{-12}	pass
34CrMo4 +QT at Ø160	500 MPa	500 MPa	0 %	10^{-12}	pass
42CrMo4 +QT at Ø0.5	900 MPa	900 MPa	0 %	10^{-12}	pass
42CrMo4 +QT at Ø8	900 MPa	900 MPa	0 %	10^{-12}	pass
42CrMo4 +QT at Ø16	900 MPa	900 MPa	0 %	10^{-12}	pass
42CrMo4 +QT at Ø16.5	750 MPa	750 MPa	0 %	10^{-12}	pass
42CrMo4 +QT at Ø28	750 MPa	750 MPa	0 %	10^{-12}	pass

Case	Reference	Axomek	Deviation	Tolerance	
42CrMo4 +QT at Ø40	750 MPa	750 MPa	0 %	10^{-12}	**pass**
42CrMo4 +QT at Ø40.5	650 MPa	650 MPa	0 %	10^{-12}	**pass**
42CrMo4 +QT at Ø70	650 MPa	650 MPa	0 %	10^{-12}	**pass**
42CrMo4 +QT at Ø100	650 MPa	650 MPa	0 %	10^{-12}	**pass**
42CrMo4 +QT at Ø100.5	550 MPa	550 MPa	0 %	10^{-12}	**pass**
42CrMo4 +QT at Ø130	550 MPa	550 MPa	0 %	10^{-12}	**pass**
42CrMo4 +QT at Ø160	550 MPa	550 MPa	0 %	10^{-12}	**pass**
34CrNiMo6 +QT at Ø0.5	1000 MPa	1000 MPa	0 %	10^{-12}	**pass**
34CrNiMo6 +QT at Ø8	1000 MPa	1000 MPa	0 %	10^{-12}	**pass**
34CrNiMo6 +QT at Ø16	1000 MPa	1000 MPa	0 %	10^{-12}	**pass**
34CrNiMo6 +QT at Ø16.5	900 MPa	900 MPa	0 %	10^{-12}	**pass**
34CrNiMo6 +QT at Ø28	900 MPa	900 MPa	0 %	10^{-12}	**pass**
34CrNiMo6 +QT at Ø40	864.7004 MPa	864.7004 MPa	0 %	10^{-12}	**pass**
34CrNiMo6 +QT at Ø40.5	800 MPa	800 MPa	0 %	10^{-12}	**pass**
34CrNiMo6 +QT at Ø70	782.0675 MPa	782.0675 MPa	0 %	10^{-12}	**pass**
34CrNiMo6 +QT at Ø100	729.4008 MPa	729.4008 MPa	0 %	10^{-12}	**pass**
34CrNiMo6 +QT at Ø100.5	700 MPa	700 MPa	0 %	10^{-12}	**pass**
34CrNiMo6 +QT at Ø130	690.6601 MPa	690.6601 MPa	0 %	10^{-12}	**pass**
34CrNiMo6 +QT at Ø160	660 MPa	660 MPa	0 %	10^{-12}	**pass**
30CrNiMo8 +QT at Ø0.5	850 MPa	850 MPa	0 %	10^{-12}	**pass**
30CrNiMo8 +QT at Ø8	850 MPa	850 MPa	0 %	10^{-12}	**pass**
30CrNiMo8 +QT at Ø16	850 MPa	850 MPa	0 %	10^{-12}	**pass**
30CrNiMo8 +QT at Ø16.5	846.1378 MPa	846.1378 MPa	0 %	10^{-12}	**pass**
30CrNiMo8 +QT at Ø28	779.762 MPa	779.762 MPa	0 %	10^{-12}	**pass**
30CrNiMo8 +QT at Ø40	734.9953 MPa	734.9953 MPa	0 %	10^{-12}	**pass**
30CrNiMo8 +QT at Ø40.5	733.4362 MPa	733.4362 MPa	0 %	10^{-12}	**pass**
30CrNiMo8 +QT at Ø70	664.7573 MPa	664.7573 MPa	0 %	10^{-12}	**pass**
30CrNiMo8 +QT at Ø100	619.9907 MPa	619.9907 MPa	0 %	10^{-12}	**pass**
30CrNiMo8 +QT at Ø100.5	619.3647 MPa	619.3647 MPa	0 %	10^{-12}	**pass**
30CrNiMo8 +QT at Ø130	587.0611 MPa	587.0611 MPa	0 %	10^{-12}	**pass**
30CrNiMo8 +QT at Ø160	561 MPa	561 MPa	0 %	10^{-12}	**pass**
S355 at Ø0.5	355 MPa	355 MPa	0 %	10^{-12}	**pass**
S355 at Ø8	355 MPa	355 MPa	0 %	10^{-12}	**pass**
S355 at Ø16	355 MPa	355 MPa	0 %	10^{-12}	**pass**
S355 at Ø16.5	345 MPa	345 MPa	0 %	10^{-12}	**pass**
S355 at Ø28	345 MPa	345 MPa	0 %	10^{-12}	**pass**

Case	Reference	Axomek	Deviation	Tolerance	
S355 at Ø40	345 MPa	345 MPa	0 %	10^{-12}	pass
S355 at Ø40.5	335 MPa	335 MPa	0 %	10^{-12}	pass
S355 at Ø51.5	335 MPa	335 MPa	0 %	10^{-12}	pass
S355 at Ø63	327.8462 MPa	327.8462 MPa	0 %	10^{-12}	pass
S355 at Ø63.5	325 MPa	325 MPa	0 %	10^{-12}	pass
S355 at Ø71.5	322.7729 MPa	322.7729 MPa	0 %	10^{-12}	pass
S355 at Ø80	318.2701 MPa	318.2701 MPa	0 %	10^{-12}	pass
S355 at Ø80.5	315 MPa	315 MPa	0 %	10^{-12}	pass
S355 at Ø90	313.5488 MPa	313.5488 MPa	0 %	10^{-12}	pass
S355 at Ø100	309.3253 MPa	309.3253 MPa	0 %	10^{-12}	pass
S355 at Ø100.5	295 MPa	295 MPa	0 %	10^{-12}	pass
S355 at Ø125	295 MPa	295 MPa	0 %	10^{-12}	pass
S355 at Ø150	293.0721 MPa	293.0721 MPa	0 %	10^{-12}	pass
S355 at Ø150.5	285 MPa	285 MPa	0 %	10^{-12}	pass
S355 at Ø175	285 MPa	285 MPa	0 %	10^{-12}	pass
S355 at Ø200	281.5403 MPa	281.5403 MPa	0 %	10^{-12}	pass
S355 at Ø200.5	275 MPa	275 MPa	0 %	10^{-12}	pass
S355 at Ø225	275 MPa	275 MPa	0 %	10^{-12}	pass
S355 at Ø250	272.5955 MPa	272.5955 MPa	0 %	10^{-12}	pass
S235 at Ø0.5	235 MPa	235 MPa	0 %	10^{-12}	pass
S235 at Ø8	235 MPa	235 MPa	0 %	10^{-12}	pass
S235 at Ø16	235 MPa	235 MPa	0 %	10^{-12}	pass
S235 at Ø16.5	225 MPa	225 MPa	0 %	10^{-12}	pass
S235 at Ø28	225 MPa	225 MPa	0 %	10^{-12}	pass
S235 at Ø40	225 MPa	225 MPa	0 %	10^{-12}	pass
S235 at Ø40.5	215 MPa	215 MPa	0 %	10^{-12}	pass
S235 at Ø51.5	215 MPa	215 MPa	0 %	10^{-12}	pass
S235 at Ø63	215 MPa	215 MPa	0 %	10^{-12}	pass
S235 at Ø63.5	215 MPa	215 MPa	0 %	10^{-12}	pass
S235 at Ø71.5	213.6666 MPa	213.6666 MPa	0 %	10^{-12}	pass
S235 at Ø80	210.6859 MPa	210.6859 MPa	0 %	10^{-12}	pass
S235 at Ø80.5	210.5205 MPa	210.5205 MPa	0 %	10^{-12}	pass
S235 at Ø90	207.5605 MPa	207.5605 MPa	0 %	10^{-12}	pass
S235 at Ø100	204.7647 MPa	204.7647 MPa	0 %	10^{-12}	pass
S235 at Ø100.5	195 MPa	195 MPa	0 %	10^{-12}	pass
S235 at Ø125	195 MPa	195 MPa	0 %	10^{-12}	pass

Case	Reference	Axomek	Deviation	Tolerance	
S235 at Ø150	194.0055 MPa	194.0055 MPa	0 %	10^{-12}	pass
S235 at Ø150.5	185 MPa	185 MPa	0 %	10^{-12}	pass
S235 at Ø175	185 MPa	185 MPa	0 %	10^{-12}	pass
S235 at Ø200	185 MPa	185 MPa	0 %	10^{-12}	pass
S235 at Ø200.5	175 MPa	175 MPa	0 %	10^{-12}	pass
S235 at Ø225	175 MPa	175 MPa	0 %	10^{-12}	pass
S235 at Ø250	175 MPa	175 MPa	0 %	10^{-12}	pass

Press fits: S235 and S355's guaranteed yield strength by size against EN 10025-2

Kind of reference	Standards’ tables and printed values
Implements	EN 10025-2:2019, Table 6 (minimum yield strength R_eH by nominal thickness, taken as the part's diameter)
Reference source	EN 10025-2:2019, Table 6 (p. 27): S235 and S355, every quality, up to 250 mm; Clause 1 (p. 6): thicknesses over 250 up to 400 mm for flat products only
Cases	19 of 19 within tolerance

The class minimum the app takes at each diameter, on both sides of class boundaries (a class runs over its lower end up to and including its upper end), against the table's value for that thickness. Beyond 250 mm the app holds none (Table 6's class over 250 up to 400 mm is for flat products, and a fit's round parts are not) and a passing yield check is not determined.

Case	Reference	Axomek	Deviation	Tolerance	
S235 at Ø16: guaranteed minimum	235 MPa	235 MPa	0 %	10^{-12}	pass
S235 at Ø16.5: guaranteed minimum	225 MPa	225 MPa	0 %	10^{-12}	pass
S235 at Ø40: guaranteed minimum	225 MPa	225 MPa	0 %	10^{-12}	pass
S235 at Ø63: guaranteed minimum	215 MPa	215 MPa	0 %	10^{-12}	pass
S235 at Ø100: guaranteed minimum	215 MPa	215 MPa	0 %	10^{-12}	pass
S235 at Ø100.5: guaranteed minimum	195 MPa	195 MPa	0 %	10^{-12}	pass
S235 at Ø175: guaranteed minimum	185 MPa	185 MPa	0 %	10^{-12}	pass
S235 at Ø200.5: guaranteed minimum	175 MPa	175 MPa	0 %	10^{-12}	pass
S235 at Ø250: guaranteed minimum	175 MPa	175 MPa	0 %	10^{-12}	pass
S355 at Ø16: guaranteed minimum	355 MPa	355 MPa	0 %	10^{-12}	pass
S355 at Ø16.5: guaranteed minimum	345 MPa	345 MPa	0 %	10^{-12}	pass
S355 at Ø50: guaranteed minimum	335 MPa	335 MPa	0 %	10^{-12}	pass
S355 at Ø80: guaranteed minimum	325 MPa	325 MPa	0 %	10^{-12}	pass
S355 at Ø80.5: guaranteed minimum	315 MPa	315 MPa	0 %	10^{-12}	pass
S355 at Ø120: guaranteed minimum	295 MPa	295 MPa	0 %	10^{-12}	pass
S355 at Ø200: guaranteed minimum	285 MPa	285 MPa	0 %	10^{-12}	pass
S355 at Ø150.5: guaranteed minimum	285 MPa	285 MPa	0 %	10^{-12}	pass
S355 at Ø225: guaranteed minimum	275 MPa	275 MPa	0 %	10^{-12}	pass

Case	Reference	Axomek	Deviation	Tolerance	
S355 at Ø250: guaranteed minimum	275 MPa	275 MPa	0 %	10^{-12}	**pass**

Press fits: C45 +QT's yield strength at size against EN 10083-2

Kind of reference	Standards’ tables and printed values
Implements	DIN 743-2:2012, 6.2, eq. (14), from C45 +QT's 490 MPa at 16 mm, at most the class's minimum
Reference source	EN 10083-2:2006, Table 9 (p. 20): C45 +QT R_e min. 490 MPa up to 16 mm, 430 MPa over 16 up to 40 mm, 370 MPa over 40 up to 100 mm
Cases	7 of 7 within tolerance

Inside a class K1 lies above the class's minimum near its lower end (at Ø50 K1 gives 407.6 MPa against 370 MPa), so the minimum is taken there; at a class's upper end K1 lies below it (423.7 against 430 MPa at Ø40, 357.4 against 370 MPa at Ø100). The value taken is never above the table and at most 4 % below it.

Case	Reference	Axomek	Deviation	Tolerance	
C45 +QT at Ø16	490 MPa	490 MPa	0 %	4 %	**pass**
C45 +QT at Ø17	430 MPa	430 MPa	0 %	4 %	**pass**
C45 +QT at Ø40	430 MPa	423.7032 MPa	1.5 %	4 %	**pass**
C45 +QT at Ø41	370 MPa	370 MPa	0 %	4 %	**pass**
C45 +QT at Ø50	370 MPa	370 MPa	0 %	4 %	**pass**
C45 +QT at Ø95	370 MPa	361.1176 MPa	2.4 %	4 %	**pass**
C45 +QT at Ø100	370 MPa	357.4064 MPa	3.4 %	4 %	**pass**

The standards’ formulae worked by hand

Press fits: yield strength at the part's size

Kind of reference	The standards’ formulae worked by hand
Implements	DIN 743-2:2012, 6.2, eqs. (11) and (14) (technological size factor K1 of the yield strength, d_B = 16 mm); DIN 7190-1:2017 (yield safety R_e/σ_v)
Reference source	DIN 743-2:2012, 6.2, eqs. (11) and (14), worked by hand at the starter fit's sizes
Cases	6 of 6 within tolerance

The hub takes K1 at its outside diameter, the shaft at its diameter at the seat (d_eff, the part's largest diameter, as DIN 743-2 6.2 takes it for a shaft). The starter fit: an S355 hub of Ø100 (eq. (11), 1 − 0.26·lg(100/32): 309.3 MPa, below EN 10025-2:2019 Table 6's 315 MPa over 80 up to 100 mm) on a 42CrMo4 +QT shaft of Ø50 (eq. (14), 1 − 0.34·lg(50/16): 748.6 MPa, held to EN 10083-3's 650 MPa over 40 up to 100 mm); its yield safeties are these strengths over the DIN 7190 stresses.

Case	Reference	Axomek	Deviation	Tolerance	
S355 hub at Ø100: K1	0.871339	0.871339	0 %	10^{-12}	**pass**
S355 hub at Ø100: R_e	309.3253 MPa	309.3253 MPa	0 %	10^{-12}	**pass**
42CrMo4 +QT shaft at Ø50: K1	0.831751	0.831751	0 %	10^{-12}	**pass**
42CrMo4 +QT shaft at Ø50: R_e, the class's minimum	650 MPa	650 MPa	0 %	10^{-12}	**pass**
hub yield safety R_e/σ_vA	1.616776	1.616776	0 %	10^{-12}	**pass**

Case	Reference	Axomek	Deviation	Tolerance	
shaft yield safety R_e/σ_vI	7.845977	7.845977	0 %	10^{-12}	**pass**

Closed-form solutions

Press fit: joint pressure, stresses and the growth of a turning hub

Kind of reference	Closed-form solutions
Implements	DIN 7190-1 (elastic interference fit of thick cylinders); Lamé's rotating disc for the growth at speed
Reference source	Lamé's thick-cylinder and rotating-disc solutions: H7/s6 at Ø50 mm, hub Ø100 mm, equal steels, solid shaft
Cases	5 of 5 within tolerance

Roughness smoothing left out for the pressure (R_z 0), as the closed form has none.

Case	Reference	Axomek	Deviation	Tolerance	
Rotating disc Ø50/Ø100 at 1000 rad/s: bore growth	0.002029669 mm	0.002029669 mm	2.1×10^{-16}	10^{-12}	**pass**
Solid rotating disc Ø100 at 1000 rad/s: rim growth $\rho\omega^2 b^3(1-\nu)/(4E)$	0.0008177083 mm	0.0008177083 mm	1.3×10^{-16}	10^{-12}	**pass**
Largest joint pressure $p = E\cdot\delta/(2d)\cdot(1-Q^2)$	92.925 MPa	92.925 MPa	1.5×10^{-16}	10^{-6}	**pass**
Hub bore equivalent stress $\sqrt{3}\cdot p/(1-Q^2)$	214.6011 MPa	214.6011 MPa	2.6×10^{-16}	4.3×10^{-7}	**pass**
Solid shaft equivalent stress p	92.925 MPa	92.925 MPa	1.5×10^{-16}	10^{-6}	**pass**

Not validated against a published reference

- DIN 7190 transmission capacity and slip, the stresses in hub and shaft against their permissible joint pressure, the press-in force, the joining temperature and turning joints: no worked example is compared (the cases are Lamé’s closed forms)

Stated limits

Standards and scope **(help: modules/fits.md)**

- Limit deviations after ISO 286-1 and ISO 286-2. The deviations are verified against the ISO 286-2 tables in `axomek verify` ("Limits and fits: ISO 286 and ISO 492", 28 checks), with ISO 286-1's special case for M6 above 250 up to 315 mm (ES −9 µm).
- Press fit after DIN 7190-1, elastic range only: plane stress, thick-walled cylinders, solid or hollow shaft, different materials.- Rotating hub after the rotating thick-walled cylinder, or from a linked housing seat's FE growth.
- Yield strengths of the steels at the part's size by DIN 743-2's K1 (6.2), taking the hub's outside diameter and the shaft's diameter at the seat as d_eff, at most the guaranteed minimum of the steel's size class (EN 10083-2, EN 10083-3; EN 10025-2:2019, Table 6 (p. 27) for S235 and S355); a shaft larger elsewhere is weaker there, which the shaft strength check takes.

- Not covered: elastic-plastic press fits, stepped or non-cylindrical hubs (except through a linked housing), conical press fits, fretting and fatigue of the joint, and the stress concentration at the hub edge. The press fit as a notch in the shaft is part of the shaft strength check.

Known limits

- A result not determined is hidden display by display, not by one shared rule: in 0.3.0 the displays reviewed (the report, Show the calculation, the CLI's text, the comparisons, the Overview, the cards, the modules' tiles and the Assistant's digest) say "not determined". Those found last are refused instead: the Changes dialog's and Compare projects' check rows and the Overview's key results of keys and bolts say "not determined (see the check)"; the assembly view gives no share of the limit for a check not determined or not computed; and in the report an element's results, and a key's or a fit's results its checks go by (safeties, capacities, permissible values, pressures, shear, the fit's Turning row; with `axomek solve`'s key and fit lines), say "not determined (see the checks)" as soon as one of its checks is not determined, determined ones with them; a key's page refuses its flank pressures and tooth shear stress the same way (its permissible values and capacity, which do not depend on its rating, stay). A bolted joint with a check not determined (beyond VDI 2230's limit G, a duty not computed) has its whole Results block refused the same way in the report's section and in `axomek solve`'s line (its tightening torque, permissible assembly preload, resiliences and embedding with it, which the bolt page still shows); its page refuses its preload needed, load factor, clamp loads, stresses, opening and the joint diagram, but the notes under them, on its page and under the report's section, still quote the eccentric model's figures (M_Sb/W_b, F_Sm/F_0.2min, F_K against F_Kab). Make it pass and `axomek fix` give such a check no value either. A module's page, once one of an item's checks is not determined, shows no tile's caption (it may quote another check's figure) and refuses its tiles and Details tied to no check (a worm pair's load value C / Clim), and the Overview refuses the item's tiles tied to no check the same way, including ones that do not depend on the rating: a worm pair's method, centre distance and ratio, a motor's power and magnetic stiffness read "not determined" too; a later release shows those again. A design search meets no requirement or objective on a result of a design with any check not determined (a requirement's own check may be cleared, and the system life and power loss take every item's), its dimensions apart, nor on a fit's yield safety resting on an estimated yield strength, even where that figure itself would stand: the safe side; a later release decides each figure on its own. A bearing whose life the engine puts outside its rating method (κ below 0.1 where it turns, balls in four-point contact, a hybrid, whose a_ISO is defined for steel only) has all its lives, basic and modified, meet no search requirement or objective, as a minimum, a maximum or an objective, its minimum life cleared or not, and so has the system life with such a member; it says "not determined" without a life check too. The safe side: a hybrid's basic life, or one at a low κ, would stand; in a later release a search requirement on a figure whose own check is cleared is evaluated as that check, each life on its own. A design with a bearing the engine marks outside what it rates (a designation that contradicts it, a slewing ring's proportions) takes no calculated result in a search, its dimensions apart but that bearing's own (inferred from its geometry as modelled), as a design with a check not determined does, whatever its own checks say. A bearing whose designation contradicts it shows only its results' warnings it can vouch for: those at its own path, of the bearing as modelled, are left out (a validation warning at that same path too, such as that its geometry is estimated), its fields' warnings stay. A check not determined is said in a short form wherever its message would be rewritten: what it checks, named from its id, "not determined" and its kind's reason in fixed words, else "(see the check)", never a part of its message, which can hold a name the user wrote and the figures after it; a name that holds an engine mark's words can still pick another kind's reason (no figures). The full engine message, with the reason in context, comes back in a later release, when every display shares one rule. The bearing module's speed chart is not drawn while the bearing's operating nθ is not determined (a rotor's pull, a designation its geometry

contradicts), saying why; `axomek speed-limits` then lists the curves' samples without an operating point. A later release draws the curves apart from that load. A display not yet reviewed may still give such a number beside its check, which says "not determined". A later release routes every display of a check's value through one rule, with a test over every template. Known in 0.3.0, closed in a later release by that one shared display rule: - the Assistant's solve data still carries a check's value and utilisation where the check is not determined (a bearing's thermal speed whose nθ is withheld: its speed and share of the limit, from which the withheld nθ can be worked out); the check is flagged not determined there, the solve tool tells the Assistant never to quote a figure so flagged, and the check's row and every display say "not determined", never a pass; - a rack pinion shaft not rated at its mesh's peak because its only drive is solved for one power direction (the Dual-pinion EPS example's SH5, driven by worm pair WG1) is said, in the verdict's and the report's short reason, to be refused because its torque sources cannot be shared; its check's own note gives the right cause; - a design search takes no result from a design with any check not determined, so a project whose steering duty leaves a warning over its states may find no design meeting a shaft or bearing requirement (the safe side; a later release decides each figure on its own); - `Life0utside::against_bounds` keeps a maximum branch no caller uses (the search reads the hybrid flag directly); a later release removes it; - the Overview's governing check of a module, between two checks not determined, keeps the first in the list (the bearing listed first) where the engine picks by id, so the module tile and the verdict can name different checks, both said not determined; - a variant comparison can mark as best the life, and its check's margin, of a bearing the engine marks unrated (its designation contradicts its geometry), though the bearing's own results say its figures are of the bearing as modelled: compare such a design only after its designation is cleared or corrected; - a comparison of three or more designs can leave out a result row whose neighbouring columns each agree within the tolerance while the first and last do not (10 000, 10 000.8 and 10 001.6 h); `--all` shows it; - a comparison of designs whose life checks all pass on a hybrid's basic life (its modified life outside its rating method, so no best) leaves that check row out of the default CLI comparison and the report, though the minimum lives and the checks' shares differ (a hybrid 6309 at a 1 million h minimum against 4 million h); `--all` shows it. - a hybrid whose modified life fails its minimum keeps that failing check's number in its check row (the fail stands, on the safe side; the number rests on a_ISO, defined for steel bearings only), and so does any life check failing outside its rating method (a steel bearing turning below κ = 0.1, or with its balls in four-point contact): the report's Summary of checks and its Summary page's Governing checks show its share and draw its utilisation bar like any other (a 30207 on 2 / 1 cSt oil against a 20 000 h minimum), though that share cannot be compared with another check's; the governing pick and the orders leave it unranked, and the typed results' report follow-up shows no bar for it. Show the calculation of a hybrid's life explains the modified life, said not determined, not the basic life its passing check goes by; - the Assistant's shortened record of an earlier solve lists a failing life check with its value (a hybrid QJ 207 in four-point contact: 94 476 h against a 1 000 000 h minimum) but without the check's outside-method flag or reason, though the same record says the bearing's life is not determined; the fail stands, and the full solve the Assistant reads carries the flag; - over a duty cycle whose cases stay within the rating method, the Assistant's solve data still gives the nominal summary's modified life where the loads as entered put κ below 0.1 (the Overhung pulley's BR2 on 10 / 2.5 cSt oil at 150 °C nominal, 35 °C in its one case: 107 191 h), with no note that it is outside the method; the duty's life, which the check and every display go by, is right (a nominal summary in four-point contact is marked); - Show the calculation of a check that fails with an unbounded share says it fails "whatever the value of the checked quantity"; for an axial preload that lifts off

(a cross-located pair whose housing runs hotter than its shaft) the share is of the lifted-off state only, and the same check passes once the temperatures no longer lift it; the check's message gives the cause; - "Make it pass" on a check not determined still steers its search by the check's share as modelled (a proposal is made only where the check is determined and passes after the change), and its "tried" list may give that share as the closest reached; - the load-spectrum card's chart still plots a bearing's damage shares where its table withholds them (a case's life outside the rating method with the life check failing); read the table; - the report's summary page labels a passing hybrid's life "modified reference rating life" before the parenthesis that says it is the basic life its check goes by; the bearing section is right; - the agent guide says a hybrid's check `value` is its basic life; where the check fails (`outsideMethod`) it is the modified life it fails on; - a variant comparison of a steel and a hybrid bearing that both pass their life check leaves that row out of the default CLI comparison and the report, though the two columns go by different lives (the row's `labels` say which); `--all` shows it; - the Overview's utilisation chart plots the share of a check failing outside its rating method beside the others, ordered first among the failures but drawn as if comparable; the fail itself is shown; - a hybrid roller bearing on estimated geometry passes on the lower of its two methods' basic lives (correct), but the report's summary page names that life with the other method's label, the Assistant's basis text points at `basicLifeHours` (the picked method's, possibly the higher) instead of the check's value, and a check made not determined by the other method's lower basic life does not quote it; the bearing finder's `basicLifeHours` and the bearing page's basic life rows show the picked method's too; the check's value and status are right; A later release types every result figure as determined or not determined, and every display, ranking and short message reads that type: this closes the ranking, display and short-message limits above for good.

- `axomek check --strict` and `axomek solve --strict` exit 3 for every file with something its method does not assess, which no input clears yet (4 where a check fails and the file was solved completely; a file that was not, an unknown field under `--strict` among it, exits 3 first): a rated key or spline (reversals and load peaks), a rated fit (its shaft seat), a belt or chain drive (its capacity), a fatigue-rated weld (its toes), a yoke on its stop, a support with a designation only and a planetary set without a planet bearing. Check those by hand and use `axomek check` without `--strict` for such files (it still exits 4 on a failing check and 3 on one not determined); a way to mark an item as checked by hand is for a later release.

- A steering duty over a steering system (0.3.0) rates the rack meshes' teeth, and with strokes the system's shafts, the bearings linked on them that take the load cases, its worm pairs and its ball nut over its states (see Steering system). Not rated over the duty (a later release): a mounted gear pair (rated at the system's `state`, its checks not rated over the duty, never a pass: its rating takes its solved shafts' KHβ, which the states do not give it yet), and likewise fits, keys and housings. A rotor's air gap is not rated over the states (its band between them is not drawn; never a pass: rate it at the duty's heaviest state). A bearing's warning over the states (a minimum load undecided at its valley) is never satisfied, and its gyroscopic slip (the high-speed model's) is the system's state's, a pass not rated over the duty. Beside such a duty a design search's metrics are refused but the shaft's mass (their values over the states have no bound between them, and some items' results are the system's state's): search with requireChecks, whose checks take the bands. The ball nut's life is the states' Miner sum (as a bearing's), its static safety the worse of the states' and the system's own state's. Left at the system's `state`: the couplings, a bearing with `useLoadCases` false, and everything at a misuse event (the events rate the pinions only; the elements keep their own `peakFactor` for a static peak, which over the duty multiplies every state's loads, as over load cases). A worm pair and a drive take the worst of their states, each rated as if it lasted their life: on the safe side, as DIN 3996 holds no load spectrum here. A shaft's states are each rated with their stresses fully reversed (as a mounted ball screw's states are), on the safe side where a stroke does not turn round. The band of a bearing's checks is drawn from its life, static safety, contact stress and thermal speed in the states, relative to each check's utilisation. Two supports in seats of one housing (whose seats push each other with the bearings' loads) leave the elements unrated over the duty (a warning at `steering.duty`, their checks never a pass): the states do not couple the seats again (a later release). The project's own load cases beside such a duty rate the shafts and bearings instead (a warning at `steering.duty`), and their checks are then not rated over the duty (never a pass): remove the load cases to rate them over it. The elements' states are some four times the duty points (284 for the Dual-pinion EPS's duty), each a state solve and a solve of every shaft: at most 1200 (`system_cases::CASE_LIMIT`) and at most 12,000 solves (`WORK_LIMIT`: each state one, and one for each shaft and each bearing on a shaft; some 25 s natively and 40 s in the browser), past which the elements are not rated over the duty (a warning at `steering.duty`; their checks never pass) while the teeth are. Measured under load on the build machine (one process, natively): the Dual-pinion EPS's duty 9.3 s for the teeth and 19 s in all; the Belt-driven ball-nut and Column EPS examples given its duty 2.4 s and 2.8 s for the teeth, 7.0 s and 5.2 s in all (168 and 88 states); the browser's engine took about the same under that load (17.9 s, 7.6 s and 6.3 s). It refuses a stroke's `torque` table (the system splits the rack force, not a pinion's torque); a rack mesh off the system's rack is not rated over the duty (a warning; it keeps its rating at its own load). Each duty point is a state solve (some 15 to 30 ms natively), at most 4000 per duty. Rating the teeth of two pinions is the slow part: the Dual-pinion EPS template's duty (a parking, a manoeuvring and an on-centre stroke, both ways) takes some 4.7 s natively and 7.3 s in the browser's engine on the build machine (some 20 s natively on a laptop, 4 s of it the state solves); the app computes it on request, with progress and Cancel (0.3.0), and solves each edit without it. Positions past the rack's travel are refused (each point is a state of the system, within its stops). The lift is rated within 1e-4 mm of the states' between points (the contact ratios within some 4e-5), not raised; the forces are raised to bound theirs. Misuse events are rated at standstill (vehicle speed 0, forward efficiencies, no friction).

- A solid wall's hub whose rim lies past 80 % of the way from its bore to the plate's nearest edge (or to half way to the next bore's hub) is refused, with the largest diameter that fits: the mesh's ring between hub and wall lies there at most. On the safe side; a hub nearly as wide as the wall round it comes in a later release.
- A parallel key's K_λ is read from DIN 6892's Figures 4 to 6 at 14 values of Q_A and six of l_tr/d (within about 0.005 of the figures), not by the more accurate method the standard cites; its Annex D.1 and D.3 print 1.07 where the figures give 1.087 and 1.097, so the app's pressures there come out 1.6 % and 2.5 % above the examples'. A thin hub (d/D from 0.6), whose own strength DIN 6892 6.2.2 then asks to be proven, is noted, not assessed. The friction factor K_R of an interference fit is not in the app (the key is rated as if the fit carried nothing, on the safe side). Nor are the load direction change factor f_W and the peak load factor f_L (6.1.3), which lower the permissible pressure for a drive that reverses or peaks often: not on the safe side, so every key's and spline's result says that reversals and the number of starts or peaks are not assessed (from 0.3.0; user test RR1). Inputs for them are planned for a later release.
- A press fit's yield strengths are sized for steels only (DIN 743-2's K1 at the hub's outside diameter and the shaft's seat diameter): the cast irons and aluminium of the list keep their listed values, and a yield strength entered by hand is taken as the part's. A shaft larger elsewhere than at its seat is weaker than its seat diameter gives (DIN 743-2 takes its largest diameter), which only the shaft strength check takes. The size classes the app holds end at 100 mm for C45 +QT, 160 mm for the alloy steels and 250 mm for S235 and S355 (EN 10025-2:2019, Table 6, p. 27; its class over 250 up to 400 mm is for flat products only, Clause 1, and is not taken for a fit's round parts), and it holds none for E335 and the case-hardening steels' cores: beyond them a passing yield check is not determined until the guaranteed yield strength at that size is entered. The class is taken at the part's finished diameter, not at the size of the bar or forging it is turned from, whose class the steel's guarantee follows and which can be lower (an S355 hub of Ø40 from Ø45 bar: 335 MPa, where Ø40 takes 345; review of 0.3.2): for a part near its yield limit, enter the yield strength of the stock's size. E335's classes (Table 7) wait for an input of the stock's size for the same reason. A fit is not linked to a shaft, so its torque is not compared with the torque its seat passes (a placed key's is): a later release.
- The shaft strength check (DIN 743) takes DIN 743-2's K1 alone, as the standard prescribes where the strength at the part's size is not known (6.2): for an alloy steel inside a size class it can lie above the class's guaranteed minimum (42CrMo4 +QT at Ø50: 748.6 MPa against EN 10083-3's 650 MPa), as in any calculation after DIN 743. Enter the guaranteed strengths ("strengths as given for the size") for a shaft near its limit. Press fits hold it to the class (see "From 0.3.0"). Keys of form AB, C to J and lift-out bevels are not in the app, nor a keyway shorter than its key: the key's length stands for both, where DIN 6892 takes the shorter for l_tr and K_λ. A spline's K_λ is still the estimate of 1 up to a length of one diameter, + 0.4 per further diameter: DIN 6892 is for parallel keys only, and DIN 5466-1's factors are not taken.

- A key's and a spline's yield strength (shaft, hub and key) is the material's value at 16 mm, whatever the part's size, as in 0.2.7. A larger part's can be lower (the release-candidate test's keyhub: an S355 hub of Ø90, 355 → about 314 MPa by DIN 743-2's K1), and the safeties shown are then optimistic. For a part over 16 mm, enter the supplier's guaranteed yield strength at that size. The app's Materials lists hold the steels' 16 mm values and have no field for it, so set it in the file or with echo `'{"op":"updateKey","id":"K1","patch":{"hubMaterial":{"name":"S355 at 90 mm","yieldStrength":314}}}' | axomek apply <file> -` (piped, as PowerShell 5.1 drops the quotes of an argument; `shaftMaterial`, `keyMaterial` likewise), under a name of its own: a material that keeps a listed name is shown in the app with the list's Re, though the file's value is the one rated. Taking it at size, as press fits do, is for a later release.
- The press-fit method of the Fits module (DIN 7190-1) is written from the app's method notes: the text of DIN 7190-1 itself was not at hand to check it against.
- The shaft drawing does not draw a hub seat with its tolerance class, and has no general tolerance line (ISO 2768), for a later release: note 5 names each untoleranced seat by its diameter and where it runs, with the class of a fit case calculated for that diameter, to be specified before manufacture (user test ut4).
- Housings: a seat a support, fit or rib sits in can't be moved by an edit, and putting a seat found on a bore onto that bore's own patch (**Seat it here**) counts as a move, so it is refused while something sits there; unlink it first, or seat it on the patch before linking (housing-032 review, O3).
- US customary units change the display and the entry only: the bearings are metric (no inch-bore bearings, adapter sleeves, pillow blocks or insert bearings in the presets or the bearing finder), the steels go by their EN names, the keys by DIN 6885-1, and the shaft drawing is metric on an ISO sheet, which its dialog says with US units shown (its fits table gives inches beside the mm). The help (Units, Bearings) says so. Inch drawings, ANSI sheets and US parts are not in 0.3.0 (RR1 decision 5).

Cylindrical gears: ISO 6336

Load capacity of spur and helical gears by ISO 6336 (pitting, tooth root, scuffing, life), the tooth geometry, accuracy grades by ISO 1328, loaded tooth contact and the mesh loss by ISO/TR 14179-2.

Methods and standards

Gears: ISO 6336-1, -2 and -3 in the edition chosen per pair (2019 by default, 2006 as ISO/TR 6336-30 rates its examples), for external and internal spur and helical gears generated by a basic rack: mesh stiffness c′ and cγα (6336-1, 9.3); dynamic factor Kv by method B with the sub-critical, main resonance and supercritical ranges (tip relief counted for grade 5 or finer); face load factor K_Hβ by method C from the pinion shaft layout (f_sh, f_ma = √(fHβ1² + fHβ2²), running-in yβ) or as given, K_Fβ = K_Hβ^NF; K_Hα and K_Fα by method B from fpb or ffα less yα; the deviations f_pt and ffα of the larger gear and fHβ of each from the ISO 1328-1:1995 grade unless entered; σ_H = ZB/ZD·σ_H0·√(K_A·Kv·K_Hβ·K_Hα) with ZH, ZE, Z_ε, Z_β = 1/√cos β and the single contact factors ZB/ZD; the limit stress σ_HG = σ_Hlim·Z_NT·ZL·ZV·ZR·ZW·ZX with the work hardening factor ZW (above 1 for a through-hardened gear meshing with a surface-hardened one) and ZX = 1; σ_F = Ft/(b·m_n)·YF·YS·Y_β·YB·YDT·K_A·Kv·K_Fβ·K_Fα with YF and YS from the 30° tangent at the outer point of single tooth contact, the rim factor YB and the deep tooth factor YDT (grade 4 or finer with tip relief and εαn above 2.05); σ_FG = σ_Flim·Y_ST·Y_NT·YδrelT·YRrelT·YX with Y_ST = 2; Z_NT and Y_NT with an optional decline to 10^{10} cycles, ZL·ZV·ZR, ZW and YδrelT·YRrelT·YX running to their static values in the limited life range.

The 2019 edition of ISO 6336-2 and -3 (the default) gives helical gears ZB = ZD = √f_ZCa (1.0, 1.07 or 1.2 by how the flanks are modified, ISO 6336-2:2019 Table 3; 1.07 with any tip relief or flank modification unless given), takes ZV at the working pitch circle and widens ZW; in YF it includes the load distribution factor fε (1/√εαn for ε_β ≥ 1) and Y_β is divided by cos³β. The 2006 edition takes ZB = ZD = 1 for helical gears with ε_β ≥ 1, fε = 1 and the 2006 Y_β.

Safeties are the limit stresses σ_HG and σ_FG (the permissible contact and root stresses in the results) over the stresses; the permissible stresses σ_HP = σ_HG/S_Hmin and σ_FP = σ_FG/S_Fmin are those limits over the required minimum safeties.

Checked against ISO/TR 6336-30 example 1.

Scuffing (with a scuffing input) by the flash temperature method of ISO/TS 6336-20:2022: the flash temperature at each point of the path of contact by the closed form of its Annex A (A.5) with the mean coefficient of friction by method C (23) (lubricant factor XL with the oil's dynamic viscosity, roughness factor from the mean Ra), the thermo-elastic factor XM (A.13), the approach factor XJ (clause 8), the geometry factor XG (A.6/A.7) with the angle factor Xαβ in the form that reproduces the TS's Table A.1, and the load sharing factor XΓ of clause 9 for unmodified profiles (tolerance class, buttressing of helical gears) or adequate tip relief (judged against the optimal tip relief Ceff and the equivalent tip reliefs of Annex B), interpolated over the overlap ratio; the transverse unit load wBt with the multiple path factor K_mp (11); the bulk temperature θM = θoil + 0.47·XS·Xmp·θflm from the mean flash temperature (20, 22); the maximum contact temperature θBmax = θM + θfl,max; the scuffing temperature from the oil's FZG A/8.3/90 load stage (77), raised at short contact exposure times for anti-scuff oils (75), and SB = (θS − θoil)/(θBmax − θoil) (78).

The integral temperature criterion after ISO/TR 13989-2:2000 (the predecessor of ISO/TS 6336-21): μmC (1) with KBγ, XR and η_oil, the flash temperature at the pinion tip θflaE (19) with XM, XBE, Xαβ, the run-in factor XE, the approach factor XQ and the tip relief factor XCa, the mean flash temperature θfla,int = θflaE·Xε (18), θM = θoil + 0.7·Xmp·θfla,int·XS (20), θint = θM + 1.5·θfla,int (17), the scuffing integral temperature θintS = θMT + XWrelT·1.5·θfla,int,T from the FZG A/8.3/90 stage (94)–(97) and SintS = θintS/θint (14).

A load spectrum by ISO 6336-6: each bin solved on its own with K_A = 1 (the spectrum is the load variation), the damage summed (Palmgren–Miner) on S–N curves that are the rating's permissible stresses at the life curve's knee points at the bin's speed (Z_NT or Y_NT with ZL·ZV·ZR, ZW and YδrelT·YRrelT·YX running to their static values in the limited life range, log–log linear in between, the permissible stress never rising with the cycles: the running minimum over the knee points; the limited life slope going on above the static knee and, with a life factor below 1 at 10^{10} cycles, beyond 10^{10}, Annex C), the safety being the lower of the fatigue safety, the factor on the stresses that makes the damage sum 1 (5.4), and the static safety σstatic/σ at the spectrum's highest stress (5.1, C.3), a bin held at 0 rpm included (no cycles, no damage, but a load of the life), so no spectrum is safer than its worst bin statically; lives the life curves cannot rate are named in a note; the safeties over the life are the rating at each life.

Each gear's contact must stay on its involute: the start of active profile, where the mate's tip meets it, at or above its form circle (ISO 21771: the end of the basic rack's straight flank, or where the trochoid of the rack's tip rounding crosses the involute of an undercut tooth; a ring's towards its root from its shaper cutter); else the pair is refused (involute interference inside the base circle, the mate's tip in the root fillet above it) with the tip diameter that clears it (a bevel pair's virtual gears with the profile shift).

The tooth forces are F_t = 2000·T_1/d_1 at the reference circle, F_r = F_t·tan α_wt and F_a = F_t·tan β, no net axial force for a double helical pair.

The mesh loss (ISO/TR 14179-2, after Niemann and Schlenk) takes the oil's viscosity at the pair's oil temperature (oilTemperature, 0 to 150 °C), along the Walther line (ASTM D341) through ν_40 and the ν100 of the oil's viscosity index (ISO 2909; 95, a mineral oil, unless given), as ISO/TS 6336-20:2022, 6.4, Formula (25), takes the same lubricant term at the oil temperature (a load case's temperature in that case, for a mounted pair that takes the cases'); without an oil temperature it takes the oil at 40 °C, and in a hot sump its coefficient of friction, and so the mesh loss, comes out about 11 % low (an ISO VG 220 oil at 95 °C), the efficiency slightly high and the heat slightly low. The scuffing check takes its η_oil (unless given) and the FZG test's XL at 90 °C on the same line.

Loaded tooth contact of cylindrical pairs (external and internal, spur and helical) with flank modifications (tip and root relief, linear or parabolic, by roll length or start diameter; profile crowning C_α; lead crowning C_β; helix slope modification C_Hβ; end relief): the plane of action in slices across the face width, at mesh positions over one base pitch; at each contact point a spring of ISO 6336-1's single stiffness c′, falling parabolically from the middle of the path of contact to c′·(0.25 + 0.75/ε_α) at its ends so that its mean is cγα; independent slices (the spring model of method C); the initial separation from both flanks' modifications and the mesh misalignment (the shaft layout's f_sh and the given misalignment, a linear gap across the face).

A double helical pair's two helices are mirrored about the middle of the face.

The normal force Fbn at the nominal torque (no load factors) is carried by the springs in compression (active set iterated to the exact solution), giving the static transmission error along the transverse line of action, the mesh stiffness, the face load factor K_Hβ as the largest over the mean line load averaged over the mesh, and the Hertz line contact pressure from the local load per unit length of contact line and the normal reduced radius of curvature.

The coupling of neighbouring slices through tooth bending, edge effects at the face ends and dynamic effects are not modelled.

Tooth root by finite elements: the virtual spur gear's teeth generated by the basic rack the rating takes for each gear (its dedendum less a finishing allowance, root radius and generating profile shift; a ring's by its shaper cutter), compared with the rating's YF·YS without the 2019 edition's load distribution factor fε (a protuberance's residual undercut is in ISO's factors only).

A gear with a thin rim: the whole virtual gear in the plane (eight-node quadrilaterals), the loaded tooth and its neighbours finely meshed, the rim of its thickness on the web as a plane-stress region of thickness bs/b, held at half the rim's inside radius; compared with the rating's YF·YS·YB.

Gearboxes: parallel shafts with common y and z axes, each solved on its own.

A mounted gear pair loads both shafts at its seats with the tooth forces at the working pitch circle: tangential Ftw = 2000·T_1/dw1 (which carries the pinion torque there; ISO 6336's Ft at the reference circle stays the rating's), radial Ftw·tan αwt and axial Ft·tan β (the transverse and axial parts of the normal force, Ftw/cos αwt·tan βb), none for a double helical pair; they act at the working pitch radii, a/(u + 1) and a·u/(u + 1) between the axes of an external pair, a/(u − 1) and a·u/(u − 1) on the far side of the pinion from the ring's axis for an internal one (the ring turning with the pinion), equal and opposite on pinion and wheel; the axial force adds a bending moment from its lever arm, the tangential force the torque.

The driven member gets the driving torque through the ratio less the mesh loss: T·u·η on a driven wheel, T·u/η from a driving wheel (η the pair's ISO/TR 14179-2 mesh efficiency).

A load in balance mode takes the torque that balances its shaft's other loads (in each load case, the case's).

Housing deflection, mesh misalignment from shaft deflection and the mesh stiffness coupling the shafts are not included.

Validation

593 of 593 cases within their tolerance, in 28 groups.

Worked examples of standards and books

Gears: ISO/TR 6336-30 example 1 (Annex A)

Kind of reference	Worked examples of standards and books
Implements	ISO 6336-1/-2/-3:2006 and ISO 6336-5:2016, as the examples apply them
Reference source	ISO/TR 6336-30:2017, the worked example of the group’s title
Cases	51 of 51 within tolerance

Single helical case-carburized pair z 17/103, m_n 8 mm, β 15.8°, b 100 mm, x 0.145/0, a 500 mm, d_a 159.66/872.35 mm, basic rack 1.4/0.39, C_a 70 µm, grade 5, E_h 1500/500 MPa, T_1 9 kN·m at 360 rpm, 50 000 h, K_Hβ by method C (l 125 mm, s 0, d_sh 100 mm), K_v by method B, Z_NT/Y_NT 0.85 at 10^{10}: the detailed calculation of ISO/TR 6336-30:2017, Annex A (ISO 6336-1/-2/-3:2006, ISO 6336-5:2016), to 0.1 %. Annex A takes the density as 7.83 kg/dm³ (the engine 7.85), which moves the resonance ratio N by 0.1 %; K_v, the K factors and Z_NT/Y_NT are printed to 3–4 digits. K_Hβ, printed 1.16 (Formula 41), is held to 1.15956, what Annex A's own values give: 1 + F_βy·c_γβ/(2·F_m/b) = 1 + 27.46409·14.84512/(2·1277.59) = 1.159561, and F_tH/F_m = 148 144/127 759 N = 1.159558 (A.8.3), to 1e-5 (F_m and F_tH printed to the newton, 4e-6 each). K_Hα by method B before its limits, K_Hα-B 0.95952 (Formulae 71 and 72; then 1 by 8.3.3), to 1e-5 (five decimals and FtH's newton).

Case	Reference	Axomek	Deviation	Tolerance	
ε_α	1.54909	1.54909	6.3×10^{-8}	0.01 %	**pass**
ε_β	1.08337	1.083369	1.2×10^{-6}	0.01 %	**pass**
ε_γ	2.63246	2.632459	4.6×10^{-7}	0.01 %	**pass**
F_t	127352 N	127352.4 N	3.0×10^{-6}	0.001 %	**pass**
c'	12.37047 N/(mm·µm)	12.37047 N/(mm·µm)	1.5×10^{-7}	0.01 %	**pass**
c_γα	17.46485 N/(mm·µm)	17.46485 N/(mm·µm)	2.1×10^{-8}	0.01 %	**pass**
c_γβ	14.84512 N/(mm·µm)	14.84512 N/(mm·µm)	1.5×10^{-7}	0.01 %	**pass**

Case	Reference	Axomek	Deviation	Tolerance	
N	0.0396	0.03964653	0.12 %	0.3 %	**pass**
K_v	1.003	1.003193	0.019 %	0.05 %	**pass**
f_sh	14.70913 µm	14.70919 µm	3.9×10^{-6}	0.03 %	**pass**
f_ma	12.74755 µm	12.74755 µm	9.5×10^{-8}	0.001 %	**pass**
F_βy	27.46409 µm	27.46415 µm	2.3×10^{-6}	0.03 %	**pass**
K_Hβ	1.15956	1.159562	1.3×10^{-6}	0.001 %	**pass**
K_Fβ	1.12803	1.128028	2.1×10^{-6}	0.01 %	**pass**
K_Hα-B	0.95952	0.9595199	1.5×10^{-7}	0.001 %	**pass**
K_Hα	1	1	0 %	10^{-9}	**pass**
K_Fα	1	1	0 %	10^{-9}	**pass**
Z_H	2.39533	2.395334	1.8×10^{-6}	0.001 %	**pass**
Z_E	189.8117 √MPa	189.8117 √MPa	2.3×10^{-9}	10^{-6}	**pass**
Z_ε	0.803	0.8034552	0.057 %	0.06 %	**pass**
Z_β	1.01944	1.019444	3.7×10^{-6}	0.001 %	**pass**
σ_H0	1206.582 MPa	1206.582 MPa	3.0×10^{-9}	0.01 %	**pass**
σ_H1	1301.353 MPa	1301.356 MPa	1.9×10^{-6}	0.02 %	**pass**
Z_L	1.04739	1.047386	3.7×10^{-6}	0.001 %	**pass**
Z_v	0.96911	0.9691142	4.3×10^{-6}	0.001 %	**pass**
Z_R	0.96599	0.9659878	2.3×10^{-6}	0.001 %	**pass**
Z_NT1	0.91	0.9100545	0.006 %	0.6 %	**pass**
Z_NT2	0.962	0.9617587	0.025 %	0.06 %	**pass**
σ_HP1	1338.481 MPa	1338.481 MPa	3.0×10^{-10}	0.01 %	**pass**
σ_HP2	1414.526 MPa	1414.526 MPa	3.3×10^{-9}	0.01 %	**pass**
S_H1	1.02853	1.028528	2.3×10^{-6}	0.03 %	**pass**
S_H2	1.08696	1.086963	2.6×10^{-6}	0.03 %	**pass**
Y_F1	1.55687	1.556822	0.0031 %	0.1 %	**pass**
Y_F2	1.33678	1.336751	0.0021 %	0.1 %	**pass**
Y_S1	1.82438	1.824397	9.5×10^{-6}	0.1 %	**pass**
Y_S2	2.07491	2.074924	6.6×10^{-6}	0.1 %	**pass**
Y_β	0.86833	0.8683333	3.8×10^{-6}	0.001 %	**pass**
σ_F01	392.6177 MPa	392.6104 MPa	0.0018 %	0.1 %	**pass**
σ_F02	383.4087 MPa	383.4037 MPa	0.0013 %	0.1 %	**pass**
σ_F1	444.2961 MPa	444.2897 MPa	0.0015 %	0.1 %	**pass**
σ_F2	433.875 MPa	433.8711 MPa	9.0×10^{-6}	0.1 %	**pass**
Y_δrelT1	0.99231	0.9923107	6.9×10^{-7}	0.01 %	**pass**
Y_δrelT2	0.9989	0.9989049	4.9×10^{-6}	0.01 %	**pass**
Y_RrelT1	0.964	0.9638812	0.012 %	0.06 %	**pass**

Case	Reference	Axomek	Deviation	Tolerance	
Y_X1	0.97	0.97	1.1×10^{-16}	10^{-9}	**pass**
Y_NT1	0.889	0.8887597	0.027 %	0.06 %	**pass**
Y_NT2	0.921	0.921424	0.046 %	0.06 %	**pass**
σ_FP1	824.569 MPa	824.5694 MPa	5.2×10^{-7}	0.01 %	**pass**
σ_FP2	860.555 MPa	860.5555 MPa	6.3×10^{-7}	0.01 %	**pass**
S_F1	1.8559	1.855928	0.0015 %	0.1 %	**pass**
S_F2	1.98342	1.983436	8.1×10^{-6}	0.1 %	**pass**

Gears: ISO/TR 6336-30 example 2

Kind of reference	Worked examples of standards and books
Implements	ISO 6336-1/-2/-3:2006 and ISO 6336-5:2016, as the examples apply them
Reference source	ISO/TR 6336-30:2017, the worked example of the group’s title
Cases	49 of 49 within tolerance

Example 1's geometry with basic rack 1.25/0.39, through-hardened V pinion and wheel (σ_Hlim 780/714, σ_Flim 319/298 MPa), grades 8/6 (pitch 23/14, profile 28/17, helix 25/14 µm), R_z 18 µm, f_sh 15 µm and f_ma 5 µm given, T_1 1 kN·m, 10 000 h: ISO/TR 6336-30:2017, Table 5. The wheel's $3.6 \cdot 10^{7}$ contact cycles lie in the limited life range, where σ_HP runs from the reference value to the static one (Z_L·Z_v·Z_R = 1), as TR 4.2.3 states.

Case	Reference	Axomek	Deviation	Tolerance	
F_t	14150 N	14150.27 N	0.0019 %	0.15 %	**pass**
ε_α	1.55	1.54909	0.059 %	0.47 %	**pass**
ε_β	1.08	1.083369	0.31 %	0.61 %	**pass**
ε_γ	2.63	2.632459	0.093 %	0.34 %	**pass**
c'	13.4 N/(mm·µm)	13.40135 N/(mm·µm)	0.01 %	0.52 %	**pass**
c_γα	18.9 N/(mm·µm)	18.92025 N/(mm·µm)	0.11 %	0.41 %	**pass**
c_γβ	16.1 N/(mm·µm)	16.08222 N/(mm·µm)	0.11 %	0.46 %	**pass**
N	0.04	0.03874478	3.1 %	13 %	**pass**
K_v	1.04	1.043027	0.29 %	0.63 %	**pass**
F_βy	14.2 µm	14.24102 µm	0.29 %	0.5 %	**pass**
K_Hβ	1.78	1.775884	0.23 %	0.43 %	**pass**
K_Fβ	1.61	1.605905	0.25 %	0.46 %	**pass**
K_Hα	1.65	1.651921	0.12 %	0.45 %	**pass**
K_Fα	1.65	1.651921	0.12 %	0.45 %	**pass**
Z_H	2.4	2.395334	0.19 %	0.36 %	**pass**
Z_E	189.81 √MPa	189.8117 √MPa	9.0×10^{-6}	0.15 %	**pass**
Z_ε	0.8	0.8034552	0.43 %	0.78 %	**pass**
Z_β	1.02	1.019444	0.055 %	0.64 %	**pass**
Z_B	1	1	0 %	0.65 %	**pass**
Z_D	1	1	0 %	0.65 %	**pass**
Z_L	1.09	1.089507	0.045 %	0.61 %	**pass**

Case	Reference	Axomek	Deviation	Tolerance	
Z_v	0.93	0.9338162	0.41 %	0.69 %	**pass**
Z_R	0.79	0.7947922	0.61 %	0.78 %	**pass**
Z_NT1	0.96	0.9561089	0.41 %	0.67 %	**pass**
Z_NT2	1.03	1.025912	0.4 %	0.64 %	**pass**
σ_H0	402 MPa	402.194 MPa	0.048 %	0.27 %	**pass**
σ_H1	703 MPa	703.5344 MPa	0.076 %	0.22 %	**pass**
σ_HP1	603 MPa	603.0413 MPa	0.0068 %	0.23 %	**pass**
σ_HP2	599 MPa	599.2043 MPa	0.034 %	0.23 %	**pass**
S_H1	0.86	0.8571596	0.33 %	0.73 %	**pass**
S_H2	0.85	0.8517058	0.2 %	0.74 %	**pass**
Y_F1	1.36	1.363842	0.28 %	0.52 %	**pass**
Y_F2	1.27	1.269342	0.052 %	0.54 %	**pass**
Y_S1	1.97	1.972531	0.13 %	0.4 %	**pass**
Y_S2	2.16	2.15793	0.096 %	0.38 %	**pass**
Y_NT1	0.92	0.9178849	0.23 %	0.69 %	**pass**
Y_NT2	0.95	0.9516197	0.17 %	0.68 %	**pass**
Y_δrelT1	0.98	0.9834132	0.35 %	0.66 %	**pass**
Y_δrelT2	1	0.9981032	0.19 %	0.65 %	**pass**
Y_RrelT1	0.96	0.9638812	0.4 %	0.67 %	**pass**
Y_X1	0.98	0.982	0.2 %	0.66 %	**pass**
σ_F01	41 MPa	41.31891 MPa	0.78 %	1.4 %	**pass**
σ_F02	42 MPa	42.07045 MPa	0.17 %	1.3 %	**pass**
σ_F1	114 MPa	114.3283 MPa	0.29 %	0.59 %	**pass**
σ_F2	116 MPa	116.4077 MPa	0.35 %	0.58 %	**pass**
σ_FP1	545 MPa	545.1047 MPa	0.019 %	0.24 %	**pass**
σ_FP2	535 MPa	535.8215 MPa	0.15 %	0.24 %	**pass**
S_F1	4.77	4.767891	0.044 %	0.25 %	**pass**
S_F2	4.6	4.602972	0.065 %	0.26 %	**pass**

Gears: ISO/TR 6336-30 example 3

Kind of reference	Worked examples of standards and books
Implements	ISO 6336-1/-2/-3:2006 and ISO 6336-5:2016, as the examples apply them
Reference source	ISO/TR 6336-30:2017, the worked example of the group’s title
Cases	49 of 49 within tolerance

Spur through-hardened pair z 17/108, m_n 8 mm, b 100 mm, x 0.1/−0.1, a 500 mm, basic rack 1.25 with root radii 0.45/0.40, V 780/714 and 319/298 MPa, grades 8/6, R_z 18 µm, K_Hβ by method C (l 100 mm), T_1 1 kN·m, 10 000 h: ISO/TR 6336-30:2017, Table 7. The running-in allowances are the means of pinion's and wheel's (ISO 6336-1:2006, 8.3.5 and 7.5.2.1).

Case	Reference	Axomek	Deviation	Tolerance	
F_t	14706 N	14705.88 N	8.0×10^{-6}	0.15 %	**pass**

Case	Reference	Axomek	Deviation	Tolerance	
ε_α	1.66	1.661318	0.079 %	0.45 %	pass
c'	13.5 N/(mm·µm)	13.46377 N/(mm·µm)	0.27 %	0.52 %	pass
c_γα	20.1 N/(mm·µm)	20.14165 N/(mm·µm)	0.21 %	0.4 %	pass
c_γβ	17.1 N/(mm·µm)	17.12041 N/(mm·µm)	0.12 %	0.44 %	pass
N	0.04	0.03563504	11 %	13 %	pass
K_v	1.05	1.050311	0.03 %	0.63 %	pass
f_sh	1.9 µm	1.920698 µm	1.1 %	2.8 %	pass
f_ma	28.7 µm	28.6531 µm	0.16 %	0.32 %	pass
F_βy	17.8 µm	17.81276 µm	0.072 %	0.43 %	pass
K_Hβ	1.99	1.987202	0.14 %	0.4 %	pass
K_Fβ	1.76	1.761948	0.11 %	0.43 %	pass
K_Hα	1.26	1.25696	0.24 %	0.55 %	pass
K_Fα	1.26	1.25696	0.24 %	0.55 %	pass
Z_H	2.49	2.494573	0.18 %	0.35 %	pass
Z_ε	0.88	0.8829273	0.33 %	0.72 %	pass
Z_β	1	1	0 %	0.65 %	pass
Z_B	1.09	1.09109	0.1 %	0.61 %	pass
Z_D	1	1	0 %	0.65 %	pass
Z_L	1.09	1.089507	0.045 %	0.61 %	pass
Z_v	0.93	0.9323147	0.25 %	0.69 %	pass
Z_R	0.79	0.7914638	0.19 %	0.78 %	pass
Z_NT1	0.96	0.9561089	0.41 %	0.67 %	pass
Z_NT2	1.03	1.029597	0.039 %	0.64 %	pass
σ_H0	468 MPa	467.6957 MPa	0.065 %	0.26 %	pass
σ_H1	827 MPa	826.5413 MPa	0.055 %	0.21 %	pass
σ_H2	758 MPa	757.5375 MPa	0.061 %	0.22 %	pass
σ_HP1	600 MPa	599.5503 MPa	0.075 %	0.23 %	pass
σ_HP2	599 MPa	599.0604 MPa	0.01 %	0.23 %	pass
S_H1	0.73	0.7253725	0.63 %	0.83 %	pass
S_H2	0.79	0.7907996	0.1 %	0.78 %	pass
Y_F1	1.43	1.428993	0.07 %	0.5 %	pass
Y_F2	1.29	1.293276	0.25 %	0.54 %	pass
Y_S1	1.86	1.85526	0.25 %	0.42 %	pass
Y_S2	2.09	2.087437	0.12 %	0.39 %	pass
Y_NT1	0.92	0.9178849	0.23 %	0.69 %	pass
Y_NT2	0.95	0.9525239	0.27 %	0.68 %	pass
Y_δrelT1	0.98	0.976659	0.34 %	0.66 %	pass
Y_δrelT2	0.99	0.9936855	0.37 %	0.66 %	pass
Y_RrelT1	0.96	0.9638812	0.4 %	0.67 %	pass
Y_X1	0.98	0.982	0.2 %	0.66 %	pass

Case	Reference	Axomek	Deviation	Tolerance	
σ_F01	49 MPa	48.73443 MPa	0.54 %	1.2 %	**pass**
σ_F02	50 MPa	49.62558 MPa	0.75 %	1.2 %	**pass**
σ_F1	113 MPa	113.3623 MPa	0.32 %	0.59 %	**pass**
σ_F2	115 MPa	115.4352 MPa	0.38 %	0.58 %	**pass**
σ_FP1	541 MPa	541.3609 MPa	0.067 %	0.24 %	**pass**
σ_FP2	533 MPa	533.9567 MPa	0.18 %	0.24 %	**pass**
S_F1	4.77	4.775495	0.12 %	0.25 %	**pass**
S_F2	4.62	4.625598	0.12 %	0.26 %	**pass**

Gears: ISO/TR 6336-30 example 4

Kind of reference	Worked examples of standards and books
Implements	ISO 6336-1/-2/-3:2006 and ISO 6336-5:2016, as the examples apply them
Reference source	ISO/TR 6336-30:2017, the worked example of the group’s title
Cases	48 of 48 within tolerance

Spur case-carburized pair z 17/108, m_n 8 mm, b 100 mm, x 0.1/−0.1, a 500 mm, basic rack 1.4/0.39, grade 5, E_h 1500/500 MPa, K_Hβ by method C (l 100 mm), T_1 9 kN·m, 50 000 h: ISO/TR 6336-30:2017, Table 9. The pinion's tip diameter is the input table's 154.60 mm, with which the report's contact ratio 1.70 results (its output table repeats example 3's 153.60 mm).

Case	Reference	Axomek	Deviation	Tolerance	
F_t	132353 N	132352.9 N	4.4×10^{-7}	0.15 %	**pass**
ε_α	1.7	1.699208	0.047 %	0.44 %	**pass**
c'	12.4 N/(mm·µm)	12.4281 N/(mm·µm)	0.23 %	0.55 %	**pass**
c_γα	18.9 N/(mm·µm)	18.94547 N/(mm·µm)	0.24 %	0.41 %	**pass**
c_γβ	16.1 N/(mm·µm)	16.10365 N/(mm·µm)	0.023 %	0.46 %	**pass**
N	0.04	0.03636441	9.1 %	13 %	**pass**
K_v	1.01	1.005057	0.49 %	0.65 %	**pass**
f_sh	16.5 µm	16.54148 µm	0.25 %	0.45 %	**pass**
f_ma	12.7 µm	12.74755 µm	0.37 %	0.54 %	**pass**
F_βy	29.5 µm	29.53556 µm	0.12 %	0.32 %	**pass**
K_Hβ	1.18	1.178779	0.1 %	0.57 %	**pass**
K_Fβ	1.14	1.142357	0.21 %	0.59 %	**pass**
K_Hα	1	1	0 %	0.65 %	**pass**
K_Fα	1	1	0 %	0.65 %	**pass**
Z_H	2.49	2.494573	0.18 %	0.35 %	**pass**
Z_ε	0.88	0.8757457	0.48 %	0.72 %	**pass**
Z_B	1.07	1.069405	0.056 %	0.62 %	**pass**
Z_D	1	1	0 %	0.65 %	**pass**
Z_L	1.05	1.047386	0.25 %	0.63 %	**pass**
Z_v	0.97	0.9684135	0.16 %	0.67 %	**pass**
Z_R	0.96	0.9638281	0.4 %	0.67 %	**pass**

Case	Reference	Axomek	Deviation	Tolerance	
Z_NT1	0.91	0.9100545	0.006 %	0.7 %	**pass**
Z_NT2	0.96	0.9631581	0.33 %	0.67 %	**pass**
σ_H0	1392 MPa	1391.675 MPa	0.023 %	0.19 %	**pass**
σ_H1	1620 MPa	1619.913 MPa	0.0054 %	0.18 %	**pass**
σ_H2	1515 MPa	1514.779 MPa	0.015 %	0.18 %	**pass**
σ_HP1	1335 MPa	1334.523 MPa	0.036 %	0.19 %	**pass**
σ_HP2	1412 MPa	1412.395 MPa	0.028 %	0.19 %	**pass**
S_H1	0.82	0.8238237	0.47 %	0.76 %	**pass**
S_H2	0.93	0.9324099	0.26 %	0.69 %	**pass**
Y_F1	1.69	1.69116	0.069 %	0.45 %	**pass**
Y_F2	1.32	1.320298	0.023 %	0.53 %	**pass**
Y_S1	1.75	1.750677	0.039 %	0.44 %	**pass**
Y_S2	2.04	2.042538	0.12 %	0.4 %	**pass**
Y_NT1	0.89	0.8887597	0.14 %	0.71 %	**pass**
Y_NT2	0.92	0.9222995	0.25 %	0.69 %	**pass**
Y_δrelT1	0.99	0.9909528	0.096 %	0.66 %	**pass**
Y_δrelT2	1	0.9975892	0.24 %	0.65 %	**pass**
Y_RrelT1	0.96	0.9638812	0.4 %	0.67 %	**pass**
Y_X1	0.97	0.97	1.1×10^{-16}	0.67 %	**pass**
σ_F01	490 MPa	489.8177 MPa	0.037 %	0.25 %	**pass**
σ_F02	446 MPa	446.155 MPa	0.035 %	0.26 %	**pass**
σ_F1	562 MPa	562.3765 MPa	0.067 %	0.24 %	**pass**
σ_F2	512 MPa	512.2459 MPa	0.048 %	0.25 %	**pass**
σ_FP1	823 MPa	823.4411 MPa	0.054 %	0.21 %	**pass**
σ_FP2	860 MPa	860.2387 MPa	0.028 %	0.21 %	**pass**
S_F1	1.46	1.464217	0.29 %	0.49 %	**pass**
S_F2	1.68	1.679347	0.039 %	0.45 %	**pass**

Gears: ISO/TR 6336-30 example 5

Kind of reference	Worked examples of standards and books
Implements	ISO 6336-1/-2/-3:2006 and ISO 6336-5:2016, as the examples apply them
Reference source	ISO/TR 6336-30:2017, the worked example of the group’s title
Cases	53 of 53 within tolerance

Spur pair z 27/192, m_n 32 mm, b 500 mm, x 0, a 3504 mm, an induction-hardened pinion (IF, ML: σ_Hlim 1009, σ_Flim 244 MPa) and a through-hardened cast steel wheel (St, 200 HB: 328/125 MPa, E 202 GPa, σ_S 400 MPa), grades 6/9, R_z 15/18 µm, no tip relief, K_Hβ by method C (l 1000 mm, d_sh 400 mm), T_1 120 kN·m at 200 rpm, 25 000 h: ISO/TR 6336-30:2017, Table 11. At 9 m/s the cast wheel's running-in allowances reach their limits 12 800/σ_Hlim and 25 600/σ_Hlim and the pinion's 3 and 6 µm (pair: the means); the soft wheel gains the work hardening factor Z_W of ISO 6336-2:2006, 13.2 (TR 4.2.4); K_Fα keeps its own limit above K_Hα's; Y_RrelT, Y_X and the slip layer follow the material groups. The engine's K_Hβ is 0.4 % below the report's (its c′ and c_γβ print 0.1–0.2 % higher), which moves SF1/SF2 by 0.4–0.6 % (checked to 0.8 %); the report's SH2 0.52 is rounded down from its own σHP2/σH2 = 304/579 = 0.525 (checked to 1.5 %).

Case	Reference	Axomek	Deviation	Tolerance	
F_t	277778 N	277777.8 N	8.0×10^{-7}	0.15 %	**pass**
ε_α	1.77	1.769429	0.032 %	0.43 %	**pass**
c'	14.2 N/(mm·µm)	14.2152 N/(mm·µm)	0.11 %	0.5 %	**pass**
c_γα	22.4 N/(mm·µm)	22.41838 N/(mm·µm)	0.082 %	0.37 %	**pass**
c_γβ	19.1 N/(mm·µm)	19.05563 N/(mm·µm)	0.23 %	0.41 %	**pass**
N	0.19	0.1874048	1.4 %	2.8 %	**pass**
K_v	1.3	1.299636	0.028 %	0.53 %	**pass**
f_sh	5.6 µm	5.561469 µm	0.69 %	1 %	**pass**
f_ma	78.8 µm	78.84796 µm	0.061 %	0.21 %	**pass**
F_βy	44.2 µm	44.22032 µm	0.046 %	0.26 %	**pass**
K_Hβ	1.59	1.583534	0.41 %	0.46 %	**pass**
K_Fβ	1.49	1.483859	0.41 %	0.49 %	**pass**
K_Hα	1.34	1.344947	0.37 %	0.52 %	**pass**
K_Fα	1.45	1.448415	0.11 %	0.49 %	**pass**
Z_H	2.49	2.494573	0.18 %	0.35 %	**pass**
Z_E	188.88 √MPa	188.879 √MPa	5.5×10^{-6}	0.15 %	**pass**
Z_ε	0.86	0.8622782	0.26 %	0.73 %	**pass**
Z_B	1.06	1.056274	0.35 %	0.62 %	**pass**
Z_D	1	1	0 %	0.65 %	**pass**
Z_L	0.99	0.9937999	0.38 %	0.66 %	**pass**
Z_v	0.99	0.9940581	0.41 %	0.66 %	**pass**
Z_R	0.88	0.8801673	0.019 %	0.72 %	**pass**
Z_NT1	0.95	0.9465231	0.37 %	0.68 %	**pass**
Z_NT2	1.01	1.012932	0.29 %	0.65 %	**pass**
Z_W1	1	1	0 %	0.65 %	**pass**
Z_W2	1.05	1.050293	0.028 %	0.63 %	**pass**
σ_H0	348 MPa	347.941 MPa	0.017 %	0.29 %	**pass**

Case	Reference	Axomek	Deviation	Tolerance	
σ_H1	612 MPa	611.4473 MPa	0.09 %	0.23 %	**pass**
σ_H2	579 MPa	578.8721 MPa	0.022 %	0.24 %	**pass**
σ_HP1	830 MPa	830.4211 MPa	0.051 %	0.21 %	**pass**
σ_HP2	304 MPa	304.5795 MPa	0.19 %	0.31 %	**pass**
S_H1	1.36	1.358124	0.14 %	0.52 %	**pass**
S_H2 (method differs, see above)	0.52	0.5261603	1.2 %	1.5 %	**pass**
Y_F1	1.28	1.281195	0.093 %	0.54 %	**pass**
Y_F2	1.12	1.115593	0.39 %	0.6 %	**pass**
Y_S1	1.97	1.971693	0.086 %	0.4 %	**pass**
Y_S2	2.33	2.328139	0.08 %	0.36 %	**pass**
Y_NT1	0.91	0.9118636	0.2 %	0.7 %	**pass**
Y_NT2	0.95	0.9484152	0.17 %	0.68 %	**pass**
Y_δrelT1	0.99	0.9931008	0.31 %	0.66 %	**pass**
Y_δrelT2	1	1.003279	0.33 %	0.65 %	**pass**
Y_RrelT1	0.96	0.9638812	0.4 %	0.67 %	**pass**
Y_RrelT2	0.98	0.9774053	0.26 %	0.66 %	**pass**
Y_X1	0.8	0.8	0 %	0.78 %	**pass**
Y_X2	0.85	0.85	0 %	0.74 %	**pass**
σ_F01	44 MPa	43.8563 MPa	0.33 %	1.3 %	**pass**
σ_F02	45 MPa	45.09123 MPa	0.2 %	1.3 %	**pass**
σ_F1	123 MPa	122.501 MPa	0.41 %	0.56 %	**pass**
σ_F2	126 MPa	125.9504 MPa	0.039 %	0.55 %	**pass**
σ_FP1	340 MPa	340.7662 MPa	0.23 %	0.3 %	**pass**
σ_FP2	197 MPa	197.6304 MPa	0.32 %	0.4 %	**pass**
S_F1 (method differs, see above)	2.77	2.781743	0.42 %	0.8 %	**pass**
S_F2 (method differs, see above)	1.56	1.569113	0.58 %	0.8 %	**pass**

Gears: ISO/TR 6336-30 example 6

Kind of reference	Worked examples of standards and books
Implements	ISO 6336-1/-2/-3:2006 and ISO 6336-5:2016, as the examples apply them
Reference source	ISO/TR 6336-30:2017, the worked example of the group’s title
Cases	48 of 48 within tolerance

Spur internal pair z 15/−99, m_n 12 mm, b 125 mm, x 0.323/0, a −500 mm, d_a 211.75/−1166 mm, basic rack 1.25 with root radii 0.39/0 (the ring cut by a z_0 = 20 shaper cutter with a sharp tip), V 780/714 and 319/298 MPa, grades 6/8, R_z 6/18 µm, K_A 1.3, K_Hβ by method C (l 200 mm), T_1 1 kN·m, 10 000 h: ISO/TR 6336-30:2017, Table 13. The ring's tooth root differs by edition: the engine generates it exactly by the shaper cutter with the 60° tangent of ISO 6336-3:2019, 6.2.5, also when rating to 2006; the report applies ISO 6336-3:2006's approximation for internal gears, whose fillet comes out larger (YS2 3.14 against 3.44, YF2 1.18 against 1.13), so the wheel's root values (YF2, YS2, YδrelT2, σF02, σF2, σFP2, SF2) are checked loosely, to show the size of the difference (SF2 2.6 % lower, on the safe side).

Closest to its tolerance: Example 6’s Zv: Axomek’s 0.9438 is ISO 6336-2’s formula at the example’s own pitch line velocity of 3.39 m/s with CZv 0.85 (σHlim below 850 MPa); the report prints 0.95, which that formula does not give, while examples 2 and 3 print 0.93 at 2.67 and 2.56 m/s, as it does. The difference lies within the printed digit plus the 0.15 % these groups allow; Axomek’s value is the lower, so its σHP is 0.65 % lower, on the safe side.

Case	Reference	Axomek	Deviation	Tolerance	
F_t	11111 N	11111.11 N	0.001 %	0.15 %	**pass**
ε_α	1.57	1.572179	0.14 %	0.47 %	**pass**
c'	14.6 N/(mm·µm)	14.55531 N/(mm·µm)	0.31 %	0.49 %	**pass**
c_γα	20.8 N/(mm·µm)	20.80149 N/(mm·µm)	0.0072 %	0.39 %	**pass**
c_γβ	17.7 N/(mm·µm)	17.68127 N/(mm·µm)	0.11 %	0.43 %	**pass**
N	0.04	0.04392939	9.8 %	13 %	**pass**
K_v	1.12	1.119548	0.04 %	0.6 %	**pass**
f_sh	1.4 µm	1.434949 µm	2.5 %	3.7 %	**pass**
f_ma	31.4 µm	31.38471 µm	0.049 %	0.31 %	**pass**
F_βy	19 µm	19.00316 µm	0.017 %	0.41 %	**pass**
K_Hβ	2.28	2.279123	0.038 %	0.37 %	**pass**
K_Fβ	1.92	1.920207	0.011 %	0.41 %	**pass**
K_Hα	1.24	1.235676	0.35 %	0.55 %	**pass**
K_Fα	1.38	1.375431	0.33 %	0.51 %	**pass**
Z_H	2.59	2.586792	0.12 %	0.34 %	**pass**
Z_ε	0.9	0.8995964	0.045 %	0.71 %	**pass**
Z_B	1.01	1.007245	0.27 %	0.65 %	**pass**
Z_D	1	1	0 %	0.65 %	**pass**
Z_L	1.09	1.089507	0.045 %	0.61 %	**pass**
Z_v	0.95	0.9437894	0.65 %	0.68 %	**pass**
Z_R	0.86	0.8631713	0.37 %	0.73 %	**pass**
Z_NT1	0.96	0.9561089	0.41 %	0.67 %	**pass**
Z_NT2	1.03	1.032572	0.25 %	0.64 %	**pass**
σ_H0	286 MPa	285.9182 MPa	0.029 %	0.32 %	**pass**

Case	Reference	Axomek	Deviation	Tolerance	
σ_H1	583 MPa	583.0503 MPa	0.0086 %	0.24 %	**pass**
σ_H2	579 MPa	578.8567 MPa	0.025 %	0.24 %	**pass**
σ_HP1	662 MPa	661.9179 MPa	0.012 %	0.23 %	**pass**
σ_HP2	660 MPa	659.71 MPa	0.044 %	0.23 %	**pass**
S_H1	1.14	1.135267	0.42 %	0.59 %	**pass**
S_H2	1.14	1.139678	0.028 %	0.59 %	**pass**
Y_F1	1.31	1.311388	0.11 %	0.53 %	**pass**
Y_F2 (method differs, see above)	1.18	1.127958	4.4 %	6 %	**pass**
Y_S1	2.08	2.077859	0.1 %	0.39 %	**pass**
Y_S2 (method differs, see above)	3.14	3.444155	9.7 %	11 %	**pass**
Y_NT1	0.92	0.9178849	0.23 %	0.69 %	**pass**
Y_NT2	0.95	0.9532523	0.34 %	0.68 %	**pass**
Y_δrelT1	0.99	0.9900003	3.4×10^{-7}	0.66 %	**pass**
Y_δrelT2 (method differs, see above)	1.05	1.103065	5.1 %	6 %	**pass**
Y_RrelT1	0.96	0.9638812	0.4 %	0.67 %	**pass**
Y_X1	0.96	0.958	0.21 %	0.67 %	**pass**
σ_F01	20 MPa	20.18429 MPa	0.92 %	2.7 %	**pass**
σ_F02 (method differs, see above)	27 MPa	28.77674 MPa	6.6 %	8 %	**pass**
σ_F1	78 MPa	77.58655 MPa	0.53 %	0.79 %	**pass**
σ_F2 (method differs, see above)	105 MPa	110.6151 MPa	5.3 %	7 %	**pass**
σ_FP1	535 MPa	535.3444 MPa	0.064 %	0.24 %	**pass**
σ_FP2 (method differs, see above)	566 MPa	578.6877 MPa	2.2 %	3 %	**pass**
S_F1	6.89	6.899964	0.14 %	0.22 %	**pass**
S_F2 (method differs, see above)	5.37	5.231542	2.6 %	3.5 %	**pass**

Gears: ISO/TR 6336-30 example 7

Kind of reference	Worked examples of standards and books
Implements	ISO 6336-1/-2/-3:2006 and ISO 6336-5:2016, as the examples apply them
Reference source	ISO/TR 6336-30:2017, the worked example of the group’s title
Cases	48 of 48 within tolerance

Double helical through-hardened pair z 86/429, m_n 1.6933 mm (15 DP), β 18.31°, b 2 × 82.55 mm, x 0.3615/0.2863 (generating xE 0.1861/0.1104 for the root form, TR footnote to Table 14), a 460.38 mm, basic rack 1.25/0.38, C_a 13 µm, grade 5, V 701/652 and 251/239 MPa (σ0.2 1000 MPa), limited pitting permissible, favourable contact pattern, K_Hβ by method C (l 355.6 mm, d_sh 63.5 mm), ν_40 25 mm²/s, T_1 528 N·m at 5400 rpm, 30 000 h: ISO/TR 6336-30:2017, Table 15. One helix sets ε_β, the shaft deflection (ISO 6336-1:2006, Formula 58), fHβ5 and K_Fβ (TR 4.2.6–4.2.9); the light load (K_A·F_t/b 42 N/mm) lowers c′ by (K_A·F_t/b/100)^0.25. The report's σ_FP of 344/338 MPa disagree with its own S_F = σ_FP/σ_F (2.01·205 = 412, 1.85·220 = 407 MPa, which the engine gives), so σ_FP is checked through S_F; YF2 with the generating profile shift differs by 0.7 % (checked to 1 %).

Case	Reference	Axomek	Deviation	Tolerance	
F_t	6884 N	6884.285 N	0.0041 %	0.16 %	**pass**
ε_α	1.68	1.682973	0.18 %	0.45 %	**pass**
ε_β	4.87	4.874983	0.1 %	0.25 %	**pass**
ε_γ	6.56	6.557956	0.031 %	0.23 %	**pass**
c'	12.7 N/(mm·µm)	12.72955 N/(mm·µm)	0.23 %	0.54 %	**pass**
c_γα	19.3 N/(mm·µm)	19.25001 N/(mm·µm)	0.26 %	0.41 %	**pass**
c_γβ	16.4 N/(mm·µm)	16.36251 N/(mm·µm)	0.23 %	0.45 %	**pass**
N	3.16	3.16725	0.23 %	0.31 %	**pass**
K_v	1.81	1.810368	0.02 %	0.43 %	**pass**
f_sh	1.5 µm	1.508522 µm	0.57 %	3.5 %	**pass**
f_ma	12.7 µm	12.74755 µm	0.37 %	0.54 %	**pass**
F_βy	3.9 µm	3.944335 µm	1.1 %	1.4 %	**pass**
K_Hβ	1.43	1.427479	0.18 %	0.5 %	**pass**
K_Fβ	1.41	1.404311	0.4 %	0.5 %	**pass**
K_Hα	1.44	1.443448	0.24 %	0.5 %	**pass**
K_Fα	1.44	1.443448	0.24 %	0.5 %	**pass**
Z_H	2.37	2.36926	0.031 %	0.36 %	**pass**
Z_ε	0.77	0.7708349	0.11 %	0.8 %	**pass**
Z_β	1.03	1.026318	0.36 %	0.64 %	**pass**
Z_B	1	1	0 %	0.65 %	**pass**
Z_D	1	1	0 %	0.65 %	**pass**
Z_L	0.85	0.8458016	0.49 %	0.74 %	**pass**
Z_v	1.09	1.091918	0.18 %	0.61 %	**pass**
Z_R	0.94	0.9401689	0.018 %	0.68 %	**pass**
Z_NT1	0.85	0.8517055	0.2 %	0.74 %	**pass**
Z_NT2	0.95	0.9540083	0.42 %	0.68 %	**pass**
σ_H0	203 MPa	203.2392 MPa	0.12 %	0.4 %	**pass**
σ_H1	392 MPa	392.5332 MPa	0.14 %	0.28 %	**pass**

Case	Reference	Axomek	Deviation	Tolerance	
σ_HP1	518 MPa	518.4082 MPa	0.079 %	0.25 %	**pass**
σ_HP2	540 MPa	540.0875 MPa	0.016 %	0.24 %	**pass**
S_H1	1.32	1.320674	0.051 %	0.53 %	**pass**
S_H2	1.38	1.375903	0.3 %	0.51 %	**pass**
Y_F1	1.17	1.167879	0.18 %	0.58 %	**pass**
Y_F2 (method differs, see above)	1.21	1.218048	0.67 %	1 %	**pass**
Y_S1	2.3	2.294957	0.22 %	0.37 %	**pass**
Y_S2	2.36	2.354721	0.22 %	0.36 %	**pass**
Y_NT1	0.85	0.8504838	0.057 %	0.74 %	**pass**
Y_NT2	0.88	0.8783137	0.19 %	0.72 %	**pass**
Y_δrelT1	1	1.000782	0.078 %	0.65 %	**pass**
Y_δrelT2	1	1.002796	0.28 %	0.65 %	**pass**
Y_RrelT1	0.96	0.9638812	0.4 %	0.67 %	**pass**
Y_X1	1	1	0 %	0.65 %	**pass**
σ_F01	56 MPa	55.92918 MPa	0.13 %	1 %	**pass**
σ_F02	60 MPa	59.85081 MPa	0.25 %	0.98 %	**pass**
σ_F1	205 MPa	205.2437 MPa	0.12 %	0.39 %	**pass**
σ_F2	220 MPa	219.6349 MPa	0.17 %	0.38 %	**pass**
S_F1	2.01	2.00661	0.17 %	0.4 %	**pass**
S_F2	1.85	1.847619	0.13 %	0.42 %	**pass**

Gears: ISO/TR 6336-30 example 8

Kind of reference	Worked examples of standards and books
Implements	ISO 6336-1/-2/-3:2006 and ISO 6336-5:2016, as the examples apply them
Reference source	ISO/TR 6336-30:2017, the worked example of the group’s title
Cases	48 of 48 within tolerance

Single helical case-carburized pair z 15/63, m_n 4.5 mm, β 11°, b 78.74/76.20 mm, x 0.5413/0.4393, a 182.88 mm, d_a 82.19/301.55 mm, protuberance basic rack 1.516/0.40 with a residual undercut of 0.047 mm, C_a 13 µm, grade 6, R_z 2.3 µm, E_h 1500/500 MPa, helix correction with crowning (B1 0.1, B2 0.5) and a favourable contact pattern, K_Hβ by method C (arrangement b, K′ −0.8, l 331.7 mm, s 67.56 mm, d_sh 63.5 mm), T_1 1494 N·m at 1430 rpm, 10 000 h, Z_NT/Y_NT 1 at 10^{10}: ISO/TR 6336-30:2017, Table 17. Fβx is the minimum 0.5·fHβ of ISO 6336-1:2006, Formula (55). The report rates the finished teeth: 1.516 is the roughing hob's dedendum and the grinding stock q = 0.22 mm makes the finished basic rack's q/sin α_n shallower (1.373), which sets CB, c′ and the root form factors (with s_pr = pr − q = 0.047 mm).

Case	Reference	Axomek	Deviation	Tolerance	
F_t	43451 N	43453.36 N	0.0054 %	0.15 %	**pass**
ε_α	1.39	1.386493	0.25 %	0.51 %	**pass**
ε_β	1.03	1.02847	0.15 %	0.64 %	**pass**
ε_γ	2.41	2.414963	0.21 %	0.36 %	**pass**
c'	13.5 N/(mm·µm)	13.53817 N/(mm·µm)	0.28 %	0.52 %	**pass**
c_γα	17.5 N/(mm·µm)	17.46248 N/(mm·µm)	0.21 %	0.44 %	**pass**

Case	Reference	Axomek	Deviation	Tolerance	
c_γβ	14.8 N/(mm·µm)	14.84311 N/(mm·µm)	0.29 %	0.49 %	**pass**
N	0.07	0.07327505	4.7 %	7.3 %	**pass**
K_v	1.02	1.019176	0.081 %	0.64 %	**pass**
f_sh	79 µm	79.02283 µm	0.029 %	0.21 %	**pass**
f_ma	14.9 µm	14.86607 µm	0.23 %	0.49 %	**pass**
F_βy	4.7 µm	4.675 µm	0.53 %	1.2 %	**pass**
K_Hβ	1.06	1.059698	0.029 %	0.62 %	**pass**
K_Fβ	1.05	1.051346	0.13 %	0.63 %	**pass**
K_Hα	1.05	1.050106	0.01 %	0.63 %	**pass**
K_Fα	1.05	1.050106	0.01 %	0.63 %	**pass**
Z_H	2.27	2.265396	0.2 %	0.37 %	**pass**
Z_ε	0.85	0.8492609	0.087 %	0.74 %	**pass**
Z_β	1.01	1.009315	0.068 %	0.65 %	**pass**
Z_B	1	1	0 %	0.65 %	**pass**
Z_D	1	1	0 %	0.65 %	**pass**
Z_L	1.02	1.019997	2.7×10^{-6}	0.64 %	**pass**
Z_v	0.98	0.9828575	0.29 %	0.66 %	**pass**
Z_R	1.03	1.024951	0.49 %	0.64 %	**pass**
Z_NT1	1	1	0 %	0.65 %	**pass**
Z_NT2	1	1	0 %	0.65 %	**pass**
σ_H0	1181 MPa	1181.048 MPa	0.0041 %	0.19 %	**pass**
σ_H1	1257 MPa	1257.766 MPa	0.061 %	0.19 %	**pass**
σ_HP1	1542 MPa	1541.289 MPa	0.046 %	0.18 %	**pass**
σ_HP2	1542 MPa	1541.289 MPa	0.046 %	0.18 %	**pass**
S_H1	1.23	1.225418	0.37 %	0.56 %	**pass**
S_H2	1.23	1.225418	0.37 %	0.56 %	**pass**
Y_F1	1.39	1.38535	0.33 %	0.51 %	**pass**
Y_F2	1.49	1.484751	0.35 %	0.49 %	**pass**
Y_S1	2.06	2.060172	0.0084 %	0.39 %	**pass**
Y_S2	2.07	2.067163	0.14 %	0.39 %	**pass**
Y_NT1	1	1	0 %	0.65 %	**pass**
Y_δrelT1	1	0.9987219	0.13 %	0.65 %	**pass**
Y_RrelT1	0.96	0.9638812	0.4 %	0.67 %	**pass**
Y_X1	1	1	0 %	0.65 %	**pass**
σ_F01	318 MPa	317.9243 MPa	0.024 %	0.31 %	**pass**
σ_F02	353 MPa	353.2884 MPa	0.082 %	0.29 %	**pass**
σ_F1	357 MPa	357.7273 MPa	0.2 %	0.29 %	**pass**
σ_F2	397 MPa	397.5189 MPa	0.13 %	0.28 %	**pass**
σ_FP1	963 MPa	962.6492 MPa	0.036 %	0.2 %	**pass**
σ_FP2	965 MPa	964.9625 MPa	0.0039 %	0.2 %	**pass**

Case	Reference	Axomek	Deviation	Tolerance	
S_F1	2.69	2.691014	0.038 %	0.34 %	**pass**
S_F2	2.43	2.427463	0.1 %	0.36 %	**pass**

Gear service life: S–N curves and static safety of ISO 6336-6

Kind of reference	Worked examples of standards and books
Implements	ISO 6336-6:2019, 5.1 and Annex C, Formulae (C.8), (C.9), (C.12), (C.13)
Reference source	ISO 6336-6:2019, Annex C: Tables C.2 (pitting) and C.3 (bending), cycles to failure of their bins, and the static safeties of C.3
Cases	9 of 9 within tolerance

Cycles to failure within the printed life factor's three decimals (m·0.0005/Z) plus 0.05 % for N's four digits. The annex's Miner sums are not compared: its safety is not consistent between Table C.2's title, C.8 and C.3.

Case	Reference	Axomek	Deviation	Tolerance	
Table C.2: cycles to failure at a life factor of 1.613	89900	89852.71	0.053 %	0.46 %	**pass**
Table C.2: cycles to failure at a life factor of 1.545	158300	158806.8	0.32 %	0.48 %	**pass**
Table C.2: cycles to failure at a life factor of 1.007	4.583×10^{7}	4.5595×10^{7}	0.51 %	0.71 %	**pass**
Table C.2: cycles to failure at a life factor of 0.903	1.368×10^{9}	1.3919×10^{9}	1.7 %	1.9 %	**pass**
Table C.3: cycles to failure at a life factor of 1.551	64700	64802.68	0.16 %	0.33 %	**pass**
Table C.3: cycles to failure at a life factor of 1.028	2.352×10^{6}	2.3568×10^{6}	0.21 %	0.48 %	**pass**
Table C.3: cycles to failure at a life factor of 0.534	1.23×10^{20}	1.1919×10^{20}	3.1 %	4.7 %	**pass**
C.3: static safety 2400/1747 MPa	1.374	1.373784	0.0002163709	0.0005	**pass**
C.3: static safety 2356/826 MPa	2.852	2.8523	0.0003002421	0.0005	**pass**

Gear mesh loss: pitch line velocity, sliding ratios and mesh mechanical advantage

Kind of reference	Worked examples of standards and books
Implements	The path of contact of the mesh loss's gear loss factor HV (the addendum contact ratios ε1, ε2 from the tip and base circles at the working pitch circle), expressed as AGMA ISO 14179-1:2004 eqs. 14 to 16
Reference source	AGMA ISO 14179-1:2004, Annex C, C.2.1, the first stage mesh of Table C.2 (pp. 23 to 24)
Cases	4 of 4 within tolerance

The example gives the transverse module 5.44507 mm, the helix angle 19.3291°, the working pressure angle 24.62721°, the working pitch and tip radii, but no normal pressure angle or profile shifts: the pair is entered with the centre distance rw1 + rw2 = 184.15 mm and the normal pressure angle (20.21°) whose transverse angle gives that centre distance at the example's working pressure angle. The sliding ratios are the engine's addendum contact ratios ε2 and ε1 as eqs. 15 and 16 define them, Hs = 2π·(u + 1)·cos αwt·ε2/(u·z_1) (Ht with ε1), and M is eq. 14 of those. The example's working pitch radii (83.69 and 100.46 mm) are not in the ratio u = 1.2 (184.15 mm gives 83.705 and 100.445, as its operating helix angle needs): that moves Hs and Ht by 0.0007 in opposite directions, which their tolerance of 0.001 covers; M, of their sum, moves by less than 10^{-4} of itself. The pitch line velocity is the engine's at the working pitch circle, as its coefficient of friction takes it (vt·cos αt/cos αwt).

Case	Reference	Axomek	Deviation	Tolerance	
Pitch line velocity V at the working pitch circle	15.6 m/s	15.60263 m/s	0.002624714 m/s	0.05 m/s	**pass**
Sliding ratio at the start of approach Hs (eq. 15)	0.2418	0.2424902	0.0006901631	0.001	**pass**
Sliding ratio at the end of recess Ht (eq. 16)	0.2533	0.2526331	0.0006669288	0.001	**pass**
Mesh mechanical advantage M (eq. 14)	7.34	7.340855	0.0008551446	0.005	**pass**

Gear mesh loss at the oil temperature: the oil's viscosity at the sump temperature

Kind of reference	Worked examples of standards and books
Implements	The kinematic viscosity of the mesh loss's oil at its working temperature (a gear pair's oilTemperature): the Walther line (ASTM D341) through ν_40 and the ν100 of viscosity index 95 (ISO 2909, the index a bearing's oil grade is filled in with)
Reference source	AGMA ISO 14179-1:2004, Annex C, C.2 (p. 21): ISO VG 220 mineral oil of kinematic viscosity 22.6 mm²/s at the reducer's 95 °C sump
Cases	1 of 1 within tolerance

The example's oil is an ISO VG 220 mineral oil (ν_40 220 mm²/s, the grade's mid-point) with no viscosity index given. At viscosity index 95 the engine's line gives 21.67 mm²/s at 95 °C, 4.1 % below the printed 22.6 mm²/s, which is the line of viscosity index 101 (ν100 19.59 mm²/s). The 5 % tolerance is that spread of mineral oils' viscosity index; ISO 3448's own ±10 % on ν_40 alone moves the viscosity at 95 °C from 20.2 to 23.1 mm²/s at index 95. The coefficient of friction takes the viscosity to the power −0.05, so the 4.1 % moves it by 0.21 % (the next method).

Case	Reference	Axomek	Deviation	Tolerance	
Kinematic viscosity of the ISO VG 220 oil at 95 °C	22.6 mm²/s	21.66764 mm²/s	4.1 %	5 %	**pass**

Gear oil viscosity at the oil temperature: the scuffing check's oil in a 95 °C sump

Kind of reference	Worked examples of standards and books
Implements	The scuffing check's oil viscosity at its oil temperature (η_oil = 0.88·ν, ISO/TS 6336-20:2022, Formula (25)) on the Walther line (ASTM D341) through ν_40 and the ν100 of viscosity index 95 (ISO 2909), the line the mesh loss takes
Reference source	AGMA ISO 14179-1:2004, Annex C, C.2 (p. 21): ISO VG 220 mineral oil of kinematic viscosity 22.6 mm²/s at the reducer's 95 °C sump
Cases	2 of 2 within tolerance

The example gives no viscosity index. At the default index 95 the line gives 21.67 mm²/s at 95 °C, 4.1 % below the printed 22.6 mm²/s (the line of index 101); the 5 % tolerance is the spread of mineral oils' index. Given the index 101, the line meets the printed value within 0.3 %. The scuffing check's fixed slope before 0.3.1 gave 19.67 mm²/s, 13.0 % below the printed value and 9.2 % below the mesh loss's line.

Case	Reference	Axomek	Deviation	Tolerance	
ISO VG 220 at 95 °C, viscosity index 95 (none given)	22.6 mm²/s	21.66764 mm²/s	4.1 %	5 %	**pass**
ISO VG 220 at 95 °C, viscosity index 101	22.6 mm²/s	22.56103 mm²/s	0.17 %	0.3 %	**pass**

Gear oil viscosity at the oil temperature: an oil of given ν_40 and viscosity index

Kind of reference	Worked examples of standards and books
Implements	The kinematic viscosity of a gear pair's oil at a temperature (the scuffing check's η_oil = 0.88·ν at its oil temperature, ISO/TS 6336-20:2022, Formula (25)): the Walther line (ASTM D341) through ν_40 and the ν100 that gives the oil's viscosity index by ISO 2909:2002, methods A and B
Reference source	ISO 2909:2002, 6.2.5 and 6.3.5, the worked examples of methods A (ν_40 73.30, ν100 8.86 mm²/s, VI 92) and B (ν_40 22.83, ν100 5.05 mm²/s, VI 156)
Cases	2 of 2 within tolerance

A pair is given the example oil's ν_40 and viscosity index and its scuffing check is run with the oil at 100 °C: the check's η_oil/0.88 there is the oil's ν100, which the example prints. Each index is entered as the example computes it before rounding it to a whole number: method A's (119.94 − 73.30)/(119.94 − 69.48)·100 = 92.43 from its printed L and H, method B's (10^N − 1)/0.00715 + 100 = 156.38 with N = (log 28.97 − log 22.83)/log 5.05 from its printed H (the rounded 156 would give ν100 0.1 % low). The tolerance 0.1 % covers the printed L and H's last digit against Table 1's interpolation.

Case	Reference	Axomek	Deviation	Tolerance	
Method A: ν100 of ν_40 73.30 mm²/s at VI 92.43	8.86 mm²/s	8.86 mm²/s	2.6×10^{-10}	0.1 %	**pass**
Method B: ν100 of ν_40 22.83 mm²/s at VI 156.38	5.05 mm²/s	5.049435 mm²/s	0.011 %	0.1 %	**pass**

Rack and pinion: transverse contact ratio of a 7-tooth jack-up pinion, the rack's tip at the published form circle

Kind of reference	Worked examples of standards and books
Implements	ISO 21771 geometry of a pinion meshing a rack: the path of contact from the pinion's tip to the rack's tip line, ended at the pinion's form circle
Reference source	A. Nowoisky, "Optimization of rack and pinion for offshore jack-up applications", Gear Solutions, August 2020, Table 2 (geometry) and Table 3 (σ_H by ISO 6336)
Cases	2 of 2 within tolerance

m_n 76 mm, α_n 28°, β 0, z_1 7, x_1 0.4 (root fillets 0.2·m_n, not printed: the 0.38·m_n of the standard rack do not fit at 28°), pinion tip Ø720 mm (shortened from the standard 744.8 mm), d_b 469.728 mm. The article's ε_α 1.153 ends the path at the pinion's form circle d_Nf1 473.46 mm (a full-depth rack tip of 1·m_n would reach Ø473.0 mm, inside it, and give 1.163): the rack's addendum is entered as the one that reaches the form circle, 0.98816·m_n.

Case	Reference	Axomek	Deviation	Tolerance	
base diameter d_b	469.728 mm	469.7281 mm	0.000119401 mm	0.0005 mm	**pass**
transverse contact ratio ε_α	1.153	1.153487	0.0004871873	0.0005	**pass**

Standards’ tables and printed values

Gears: ISO 1328-1:1995 tolerances for the accuracy grade

Kind of reference	Standards’ tables and printed values
Implements	ISO 1328-1:1995
Reference source	The tolerances ISO/TR 6336-30:2017 examples 1 and 2 are rated with
Cases	12 of 12 within tolerance

The single pitch, profile form and helix slope tolerances f_pt, ffα and fHβ that a pair without measured deviations takes from its accuracy grade, against the values ISO/TR 6336-30:2017 rates its examples 1 (grade 5) and 2 (grades 8 and 6) with: z 17/103, m_n 8 mm, β 15.8° (d 141.3/856.3 mm), b 100 mm. ISO 1328-1:1995 tabulates its formulae at the geometric means of the ranges of module, diameter and face width, rounded.

Case	Reference	Axomek	Deviation	Tolerance	
fpt, pinion, grade 5	8 µm	8 µm	0 %	10^{-9}	**pass**
ffα, pinion, grade 5	10 µm	10 µm	0 %	10^{-9}	**pass**
fHβ, pinion, grade 5	8.5 µm	8.5 µm	0 %	10^{-9}	**pass**
fpt, wheel, grade 5	9.5 µm	9.5 µm	0 %	10^{-9}	**pass**
ffα, wheel, grade 5	12 µm	12 µm	0 %	10^{-9}	**pass**
fHβ, wheel, grade 5	9.5 µm	9.5 µm	0 %	10^{-9}	**pass**
fpt, pinion, grade 8	23 µm	23 µm	0 %	10^{-9}	**pass**
ffα, pinion, grade 8	28 µm	28 µm	0 %	10^{-9}	**pass**
fHβ, pinion, grade 8	25 µm	25 µm	0 %	10^{-9}	**pass**
fpt, wheel, grade 6	14 µm	14 µm	0 %	10^{-9}	**pass**
ffα, wheel, grade 6	17 µm	17 µm	0 %	10^{-9}	**pass**
fHβ, wheel, grade 6	14 µm	14 µm	0 %	10^{-9}	**pass**

Gears: scuffing, flash temperature

Kind of reference	Standards’ tables and printed values
Implements	ISO/TS 6336-20:2022
Reference source	ISO/TS 6336-20:2022 Table A.1, Formula (A.14); FZG type A test gears (ISO 14635-1) by Formulae (72) and (77)
Cases	21 of 21 within tolerance

ISO/TS 6336-20:2022: the angle factor Xαβ of the geometry factor against the TS's Table A.1 (α_n = 20°), the thermo-elastic factor of steel against Formula (A.14), and the largest flash temperature of the FZG type A test gears (ISO 14635-1: 16/24 teeth, m = 4.5 mm, 2170 rpm, oil at 90 °C, ν_40 = 100 mm²/s, Ra = 0.3 µm) rated by the TS's own formulae against the test-gear flash temperature 1.4·XL·S² that Formula (77) implies through Formula (72). The TS gives no worked example; the FZG relation states neither the test's friction nor its load factors, so it is held to 30 %.

Case	Reference	Axomek	Deviation	Tolerance	
Xαβ, β = 0°, αwt = 18°	0.947	0.9473399	0.036 %	0.1 %	**pass**

Case	Reference	Axomek	Deviation	Tolerance	
Xαβ, β = 0°, αwt = 20°	0.978	0.9775374	0.047 %	0.1 %	**pass**
Xαβ, β = 0°, αwt = 22°	1.007	1.006756	0.024 %	0.1 %	**pass**
Xαβ, β = 0°, αwt = 24°	1.035	1.035325	0.031 %	0.1 %	**pass**
Xαβ, β = 0°, αwt = 26°	1.064	1.063515	0.046 %	0.1 %	**pass**
Xαβ, β = 10°, αwt = 20°	0.975	0.9746876	0.032 %	0.1 %	**pass**
Xαβ, β = 10°, αwt = 22°	1.004	1.003821	0.018 %	0.1 %	**pass**
Xαβ, β = 10°, αwt = 24°	1.032	1.032307	0.03 %	0.1 %	**pass**
Xαβ, β = 10°, αwt = 26°	1.06	1.060415	0.039 %	0.1 %	**pass**
Xαβ, β = 20°, αwt = 20°	0.966	0.9661609	0.017 %	0.1 %	**pass**
Xαβ, β = 20°, αwt = 22°	0.995	0.9950396	0.004 %	0.1 %	**pass**
Xαβ, β = 20°, αwt = 24°	1.023	1.023276	0.027 %	0.1 %	**pass**
Xαβ, β = 20°, αwt = 26°	1.051	1.051138	0.013 %	0.1 %	**pass**
Xαβ, β = 30°, αwt = 22°	0.981	0.9805282	0.048 %	0.1 %	**pass**

Case	Reference	Axomek	Deviation	Tolerance	
Xαβ, β = 30°, αwt = 24°	1.008	1.008353	0.035 %	0.1 %	**pass**
Xαβ, β = 30°, αwt = 26°	1.036	1.035809	0.018 %	0.1 %	**pass**
Xαβ, β = 30°, αwt = 28°	1.063	1.063124	0.012 %	0.1 %	**pass**
XM of steel (E = 206 000 N/mm², ν = 0.3, BM = 13.8)	50 K·N^−¾·s^½·m^−½·mm	49.98358 K·N^−¾·s^½·m^−½·mm	0.033 %	0.1 %	**pass**
FZG A/8.3/90 stage 8: θfl,max of the test gears	78.90872 K	94.18701 K	19 %	30 %	**pass**
FZG A/8.3/90 stage 10: θfl,max of the test gears	123.2949 K	142.6742 K	16 %	30 %	**pass**
FZG A/8.3/90 stage 12: θfl,max of the test gears	177.5446 K	200.4205 K	13 %	30 %	**pass**

Gears: scuffing, integral temperature

Kind of reference	Standards’ tables and printed values
Implements	ISO/TR 13989-2:2000
Reference source	ISO/TR 13989-2:2000, Annex A, Tables A.1 to A.5
Cases	10 of 10 within tolerance

ISO/TR 13989-2:2000, Annex A, Tables A.1–A.5 (helical gears of the Michaelis dissertation): the integral temperature θint and the scuffing safety SintS by the TR's equations from the tables' data (η_oil from ν_40 with 0.88 kg/dm³, along a mineral oil's Walther line of viscosity index 95; XM of steel from 5.3) against the tabulated results. The tables come from the data set rather than from worked examples, so they are held to 12.5 % (θint) and 15 % (SintS).

Case	Reference	Axomek	Deviation	Tolerance	
Table A.1: θint	55.5 °C	56.21091 °C	1.3 %	13 %	**pass**
Table A.1: SintS	3.8	3.706091	2.5 %	15 %	**pass**
Table A.2: θint	109 °C	106.7643 °C	2.1 %	13 %	**pass**
Table A.2: SintS	1.9	1.925253	1.3 %	15 %	**pass**
Table A.3: θint	159.4 °C	161.2198 °C	1.1 %	13 %	**pass**
Table A.3: SintS	2	1.962463	1.9 %	15 %	**pass**
Table A.4: θint	185.7 °C	163.4256 °C	12 %	13 %	**pass**
Table A.4: SintS	1.7	1.942228	14 %	15 %	**pass**
Table A.5: θint	64.9 °C	60.89487 °C	6.2 %	13 %	**pass**
Table A.5: SintS	2.3	2.483344	8 %	15 %	**pass**

The standards’ formulae worked by hand

Gears: ISO 6336-2/-3:2019 work hardening and life factors

Kind of reference	The standards’ formulae worked by hand
Implements	ISO 6336-2:2019 13.3, Formulae (9), (10), (65)–(71); ISO 6336-3:2019 Formulae (6), (8)
Reference source	The standard’s formulae, worked by hand
Cases	15 of 15 within tolerance

The work hardening factor Z_W of ISO 6336-2:2019, 13.3, by its Formulae (65)–(71) for a through-hardened pair and a ductile iron wheel (pearlitic and ferritic) with a case-hardened pinion, on a spur pair z 20/60, m 3 mm, x 0.5/0.5, 150 N·m at 1500 rpm, σ_Hlim 1500/700 MPa; and the permissible stresses in the limited life range, interpolated log-log between the reference and the static values along the life curve: σ_HP of a through-hardened wheel with limited pitting permitted (ISO 6336-2:2019, Formulae 9 and 10, ν_40 32 mm²/s and R_z 12 µm for Z_L·Z_v·Z_R well below 1) and σ_FP of case-hardened teeth (ISO 6336-3:2019, Formulae 6 and 8, m 6 mm and fillet R_z 20 µm for Y_X and Y_RrelT below 1), from the engine's reference factors.

Case	Reference	Axomek	Deviation	Tolerance	
Z_W, ductile iron wheel (pearlitic) 200 HB, Formula (69)	1.158588	1.158588	0 %	10^{-9}	**pass**
Z_W, ductile iron wheel (pearlitic) 250 HB, Formula (69)	1.103588	1.103588	0 %	10^{-9}	**pass**
Z_W, ductile iron wheel (pearlitic) 300 HB, Formula (69)	1.048588	1.048588	0 %	10^{-9}	**pass**

Case	Reference	Axomek	Deviation	Tolerance	
Z_W, ductile iron wheel (ferritic) 200 HB, Formula (69)	1.158588	1.158588	0 %	10^{-9}	pass
Z_W, ductile iron wheel (ferritic) 250 HB, Formula (69)	1.103588	1.103588	0 %	10^{-9}	pass
Z_W, ductile iron wheel (ferritic) 300 HB, Formula (69)	1.048588	1.048588	0 %	10^{-9}	pass
Z_W, through-hardened wheel, HB1/HB2 = 400/250, u = 3, Formulae (65)–(67)	1.012156	1.012156	0 %	10^{-9}	pass
Z_W, through-hardened wheel, HB1/HB2 = 360/300, u = 3, Formulae (65)–(67)	1.004972	1.004972	0 %	10^{-9}	pass
Z_W, through-hardened wheel, HB1/HB2 = 320/300, u = 3, Formulae (65)–(67)	1	1	0 %	10^{-9}	pass
σ_HP, limited pitting, NL = 3e7, Formulae (9), (10)	670.8138 MPa	670.8218 MPa	0.0012 %	0.01 %	pass
σ_HP, limited pitting, NL = 1e8, Formulae (9), (10)	588.9771 MPa	588.9817 MPa	7.8×10^{-6}	0.01 %	pass
σ_HP, limited pitting, NL = 5e8, Formulae (9), (10)	494.9552 MPa	494.9564 MPa	2.4×10^{-6}	0.01 %	pass
σ_FP, case-hardened pinion, NL = 1e4, Formulae (6), (8)	2052.922 MPa	2052.89 MPa	0.0016 %	0.01 %	pass
σ_FP, case-hardened pinion, NL = 1e5, Formulae (6), (8)	1502.839 MPa	1502.825 MPa	9.2×10^{-6}	0.01 %	pass
σ_FP, case-hardened pinion, NL = 1e6, Formulae (6), (8)	1100.151 MPa	1100.148 MPa	3.0×10^{-6}	0.01 %	pass

Gear lead crowning: the amount ISO 6336-1 Annex B gives and the band that counts as a suitable modification

Kind of reference	The standards’ formulae worked by hand
Implements	ISO 6336-1:2019, Annex B, B.2, Formulae (B.2) and (B.3), within 10–40 µm plus a manufacturing tolerance
Reference source	The standard's formulae, worked by hand
Cases	11 of 11 within tolerance

Both gears' crowning together against Annex B's amount for the pair: 0.5·(f_sh + 1.5·f_Hβ), or f_Hβ with corrected helices, held within 10–40 µm; the band reaches 60 % of it (B.2's share for very accurate gears) and its 10 µm manufacturing tolerance above, up to 50 µm when f_sh is not known. Exact values: a tolerance of 1e-9 µm.

Case	Reference	Axomek	Deviation	Tolerance	
B.2 amount	19 µm	19 µm	0 µm	1×10^{-9} µm	pass
low	11.4 µm	11.4 µm	0 µm	1×10^{-9} µm	pass
high	29 µm	29 µm	0 µm	1×10^{-9} µm	pass
pair's crowning	11 µm	11 µm	0 µm	1×10^{-9} µm	pass
B.3 amount	12 µm	12 µm	0 µm	1×10^{-9} µm	pass
B.3 high	22 µm	22 µm	0 µm	1×10^{-9} µm	pass
lower limit	10 µm	10 µm	0 µm	1×10^{-9} µm	pass

Case	Reference	Axomek	Deviation	Tolerance	
stiff high	20 µm	20 µm	0 µm	1×10^{-9} µm	**pass**
upper limit	40 µm	40 µm	0 µm	1×10^{-9} µm	**pass**
unknown f_sh amount	10 µm	10 µm	0 µm	1×10^{-9} µm	**pass**
unknown f_sh high	50 µm	50 µm	0 µm	1×10^{-9} µm	**pass**

Gear single pair contact factors ZB = ZD = √f_ZCa of a crowned helical pair

Kind of reference	The standards’ formulae worked by hand
Implements	ISO 6336-2:2019, Formula (19) and Table 3: f_ZCa 1.07 with suitable flank modifications, 1.2 without; a lead crowning counts within ISO 6336-1:2019 Annex B's band
Reference source	The standard's formulae, worked by hand
Cases	12 of 12 within tolerance

A helical pair z 21/63, m_n 3 mm, β 15°, b 40 mm (ε_β 1.10 ≥ 1), f_sh 20 µm, f_Hβ 12/10 µm, whose band is 11.4–29 µm. With ε_β ≥ 1, ZB = ZD = √f_ZCa exactly; the crowning is not in K_Hβ (Annex C is not applied), so S_H scales with 1/ZB. Exact values: a tolerance of 1e-9.

Case	Reference	Axomek	Deviation	Tolerance	
over-crowned	1.103218	1.103218	0	1×10^{-9}	**pass**
S_H within the band over S_H uncrowned, √(1.2/1.07)	1.059007	1.059007	2.2204×10^{-16}	1×10^{-9}	**pass**
uncrowned	1.095445	1.095445	0	1×10^{-9}	**pass**
6 µm, below	1.095445	1.095445	0	1×10^{-9}	**pass**
11.6 µm, within	1.034408	1.034408	0	1×10^{-9}	**pass**
29 µm, the top of the band	1.034408	1.034408	0	1×10^{-9}	**pass**
30 µm, above	1.095445	1.095445	0	1×10^{-9}	**pass**
60 µm, above	1.095445	1.095445	0	1×10^{-9}	**pass**
tip relief, 6 µm	1.034408	1.034408	0	1×10^{-9}	**pass**
tip relief, 60 µm	1.095445	1.095445	0	1×10^{-9}	**pass**
hollow	1.095445	1.095445	0	1×10^{-9}	**pass**
given f_ZCa	1.034408	1.034408	0	1×10^{-9}	**pass**

Gear mesh loss at the oil temperature: the coefficient of friction at the sump's viscosity

Kind of reference	The standards’ formulae worked by hand
Implements	Schlenk's mean coefficient of friction μmz = 0.048·(Fbt/b/(vΣC·ρredC))^0.2·η^−0.05·Ra^0.25 with the oil's dynamic viscosity η at its working temperature, as ISO/TS 6336-20:2022, 6.4, Formulae (23) and (25), takes the same lubricant term (η_oil at the oil temperature θ_oil); η = 0.88·ν
Reference source	AGMA ISO 14179-1:2004, Annex C, C.2 (p. 21): ISO VG 220 mineral oil of kinematic viscosity 22.6 mm²/s at the reducer's 95 °C sump
Cases	1 of 1 within tolerance

The first stage mesh of the example (Table C.2) at R_z 3 μm with its oilTemperature 95 °C, against the same formula worked with the example's printed sump viscosity: μmz(40 °C)·(22.6/220)^−0.05, every other quantity of the formula unchanged by the oil's temperature. The engine's viscosity at viscosity index 95 is 4.1 % lower than the printed one (previous method), which raises μmz by 0.21 %; the tolerance 0.3 % covers that. The oil at 40 °C, as before the oil temperature was an input, gave μmz 10.7 % lower than the sump's viscosity does.

Case	Reference	Axomek	Deviation	Tolerance	
Mean coefficient of friction μmz at the 95 °C sump	0.02996938	0.03003258	0.21 %	0.3 %	**pass**

Gear tooth root stress of gears wider than the face in contact

Kind of reference	The standards’ formulae worked by hand
Implements	ISO 6336-3:2019, σ_F = Ft/(b·m_n)·YF·YS·Yβ·YB·YDT·KA·Kv·KFβ·KFα, with the narrower gear's width counted at most the face width b in contact, the wider's at most that + 2·m_n
Reference source	The standard's formula, worked by hand
Cases	11 of 11 within tolerance

The pair above with per-gear widths 78.74/76.2 mm left from ISO/TR 6336-30 example 8 on b 40 mm: the narrower wheel's root counts b (40 mm), the pinion's 46 mm, and σ_F·b_root/KFβ is that of the pair without them (KFβ takes b/h from the gears' own widths, ISO 6336-1 7.6, the larger and safer). Exact values: a tolerance of 1e-9 (relative for the stresses).

Case	Reference	Axomek	Deviation	Tolerance	
own width	40 mm	40 mm	0 mm	1×10^{-9} mm	**pass**
pinion capped at b + 2·m_n	46 mm	46 mm	0 mm	1×10^{-9} mm	**pass**
narrower wheel capped at b	40 mm	40 mm	0 mm	1×10^{-9} mm	**pass**
equal widths above b	40 mm	40 mm	0 mm	1×10^{-9} mm	**pass**
equal widths above b, wheel	40 mm	40 mm	0 mm	1×10^{-9} mm	**pass**
pinion: σ_F·b_root/KFβ with 78.74/76.2 mm left on b 40 mm	6647.589 N/mm	6647.589 N/mm	1.4×10^{-16}	10^{-9}	**pass**
wheel: σ_F·b_root/KFβ with 78.74/76.2 mm left on b 40 mm	6454.918 N/mm	6454.918 N/mm	0 %	10^{-9}	**pass**
wider pinion	42 mm	42 mm	0 mm	1×10^{-9} mm	**pass**
wheel at b	40 mm	40 mm	0 mm	1×10^{-9} mm	**pass**
narrower + 2·m_n	42 mm	42 mm	0 mm	1×10^{-9} mm	**pass**

Case	Reference	Axomek	Deviation	Tolerance	
narrow wheel	36 mm	36 mm	0 mm	1×10^{-9} mm	pass

Rack and pinion: nominal contact stress σ_H0 with every load factor 1

Kind of reference: The standards' formulae worked by hand

Implements: ISO 6336-2:2019, 6.1, Formula (2) with u → ∞: σ_H0 = ZH·ZE·Z_ε·Z_β·√(Ft/(d_1·b))

Reference source: A. Nowoisky, "Optimization of rack and pinion for offshore jack-up applications", Gear Solutions, August 2020, Table 2 (geometry) and Table 3 (σ_H by ISO 6336)

Cases: 6 of 6 within tolerance

The pinion loads of Table 3 (65 to 360 t, ×9.80665 m/s²) as the tangential force on the pinion's reference circle, b the narrower face (the rack's 152.4 mm). The article's σ_H (ISO 6336) are 1.4227 times this σ_H0 at every load (to 0.1 %), so they scale with √Ft as σ_H0 does: its load factors and single pair contact factor together come to $1.4227^2 = 2.024$ (not printed). The comparison here is σ_H0 by hand from the zone, elasticity and contact ratio factors against the engine's.

Case	Reference	Axomek	Deviation	Tolerance	
σ_H0 at 65 t	1138.756 N/mm²	1138.756 N/mm²	2.0×10^{-16}	10^{-6}	pass
σ_H0 at 170 t	1841.613 N/mm²	1841.613 N/mm²	1.2×10^{-16}	10^{-6}	pass
σ_H0 at 225 t	2118.679 N/mm²	2118.679 N/mm²	2.1×10^{-16}	10^{-6}	pass
σ_H0 at 240 t	2188.162 N/mm²	2188.162 N/mm²	0 %	10^{-6}	pass
σ_H0 at 315 t	2506.855 N/mm²	2506.855 N/mm²	1.8×10^{-16}	10^{-6}	pass
σ_H0 at 360 t	2679.94 N/mm²	2679.94 N/mm²	1.7×10^{-16}	10^{-6}	pass

Closed-form solutions

Gear oil viscosity at the oil temperature: one line for the scuffing check and the mesh loss

Kind of reference: Closed-form solutions

Implements: ASTM D341 (Walther): log log(ν + 0.7) = A − B·log T, through (40 °C, ν_40) and (100 °C, ν100); ISO/TS 6336-20:2022, Formula (25), XL = (η_oil)^−0.05 for a mineral oil, and Formula (77)'s XL at the FZG test's 90 °C

Reference source: The closed form of the Walther line, worked by hand for ISO VG 220 at viscosity index 95 and 150

Cases: 12 of 12 within tolerance

A and B from the two points; ν at θ = 10^10^(A − B·log(θ + 273.15)) − 0.7. The scuffing check's η_oil at its oil temperature and its test XL at 90 °C, and the mesh loss's viscosity at the same oil temperature, are all this line: compared to 10^{-12}.

Case	Reference	Axomek	Deviation	Tolerance	
ISO VG 220 at VI 95: ν at 60 °C (η_oil/0.88)	78.29462 mm²/s	78.29462 mm²/s	0 %	10^{-12}	pass
ISO VG 220 at VI 95: the FZG test's XL at 90 °C	0.85638	0.85638	0 %	10^{-12}	pass

Case	Reference	Axomek	Deviation	Tolerance	
ISO VG 220 at VI 95: ν at 95 °C (η_oil/0.88)	21.66764 mm²/s	21.66764 mm²/s	0 %	10^{-12}	**pass**
ISO VG 220 at VI 95: the FZG test's XL at 90 °C	0.85638	0.85638	0 %	10^{-12}	**pass**
ISO VG 220 at VI 95: ν at 120 °C (η_oil/0.88)	11.28527 mm²/s	11.28527 mm²/s	0 %	10^{-12}	**pass**
ISO VG 220 at VI 95: the FZG test's XL at 90 °C	0.85638	0.85638	0 %	10^{-12}	**pass**
ISO VG 220 at VI 150: ν at 60 °C (η_oil/0.88)	91.79747 mm²/s	91.79747 mm²/s	0 %	10^{-12}	**pass**
ISO VG 220 at VI 150: the FZG test's XL at 90 °C	0.8438244	0.8438244	0 %	10^{-12}	**pass**
ISO VG 220 at VI 150: ν at 95 °C (η_oil/0.88)	29.55174 mm²/s	29.55174 mm²/s	0 %	10^{-12}	**pass**
ISO VG 220 at VI 150: the FZG test's XL at 90 °C	0.8438244	0.8438244	0 %	10^{-12}	**pass**
ISO VG 220 at VI 150: ν at 120 °C (η_oil/0.88)	16.2378 mm²/s	16.2378 mm²/s	0 %	10^{-12}	**pass**
ISO VG 220 at VI 150: the FZG test's XL at 90 °C	0.8438244	0.8438244	0 %	10^{-12}	**pass**

The same cases calculated by other programs or methods

Gears: ISO 6336 method B factors against a published calculation

Kind of reference	The same cases calculated by other programs or methods
Implements	ISO 6336-1/-2/-3, method B
Reference source	A published spur gear calculation printout of a gear calculation program
Cases	12 of 12 within tolerance

Spur pair z = 19/48, mn = 6 mm, αn = 20°, x = 0, basic rack ha0* = 1.25, ρa0* = 0.38, against a published spur gear calculation printout of a gear calculation program (ISO 6336).

Case	Reference	Axomek	Deviation	Tolerance	
Transverse contact ratio ε_α	1.6456	1.645626	0.0016 %	0.2 %	**pass**
Elasticity factor Z_E	189.81 √MPa	189.8117 √MPa	9.0×10^{-6}	0.2 %	**pass**
Zone factor Z_H	2.495	2.494573	0.017 %	0.2 %	**pass**
Contact ratio factor Z_ε	0.886	0.8858844	0.013 %	0.2 %	**pass**
Single pair contact factor Z_B	1.08	1.079729	0.025 %	0.2 %	**pass**
Single pair contact factor Z_D	1	1	0 %	0.2 %	**pass**
Tooth form factor Y_F, pinion	1.606	1.606089	0.0055 %	0.2 %	**pass**
Tooth form factor Y_F, wheel	1.344	1.344352	0.026 %	0.2 %	**pass**
Stress correction factor Y_S, pinion	1.807	1.807241	0.013 %	0.2 %	**pass**
Stress correction factor Y_S, wheel	2.018	2.018139	0.0069 %	0.2 %	**pass**
Base diameter db, pinion	107.125 mm	107.125 mm	3.8×10^{-7}	0.001 %	**pass**
Base diameter db, wheel	270.6315 mm	270.6315 mm	9.3×10^{-8}	0.001 %	**pass**

Gears: ISO/TR 6336-30 example 1 (JGMA reproduction)

Kind of reference	The same cases calculated by other programs or methods
Implements	ISO 6336-1/-2/-3:2019
Reference source	ISO/TR 6336-30:2017 example 1 inputs; results of the JGMA reproduction (2019)
Cases	47 of 47 within tolerance

Single helical case-carburized pair z 17/103, m_n 8 mm, β 15.8°, b 100 mm, x 0.145/0, basic rack 1.4/0.39, T_1 9 kN·m at 360 rpm, grade 5, face load factor by method C, Z_NT/Y_NT 0.85 at 10^{10}: inputs from ISO/TR 6336-30:2017, results from the JGMA reproduction (Maxima ISO 6336 program, 2019). The example fixes a = 500 mm; this engine derives a = 499.998 mm from the profile shift. The reproduction follows ISO 6336-3:2019, whose tooth form factor includes the load distribution factor fε, so Y_F is checked with the 2019 edition; fε and the 2019 helix factor Y_β against published comparison tables of ISO 6336:2006 and 2019 (2020).

Case	Reference	Axomek	Deviation	Tolerance	
Tangential force F_t	127352.4 N	127352.4 N	1.2×10^{-8}	10^{-6}	**pass**
Transverse contact ratio ε_α	1.549238	1.549542	0.02 %	0.03 %	**pass**
Overlap ratio ε_β	1.083369	1.083369	2.9×10^{-7}	0.001 %	**pass**
Single stiffness c′	12.37047 N/(mm·µm)	12.37047 N/(mm·µm)	1.5×10^{-7}	0.01 %	**pass**
Mesh stiffness c_γα	17.46622 N/(mm·µm)	17.46904 N/(mm·µm)	0.016 %	0.03 %	**pass**
Shaft misalignment f_sh	14.70587 µm	14.70917 µm	0.022 %	0.03 %	**pass**
Manufacturing misalignment f_ma	12.74755 µm	12.74755 µm	9.5×10^{-8}	0.001 %	**pass**
Face load factor K_Hβ (method C)	1.159588	1.1596	0.001 %	0.01 %	**pass**
Face load factor K_Fβ	1.128049	1.128058	7.9×10^{-6}	0.01 %	**pass**
Dynamic factor K_v (method B)	1.002967	1.003192	0.022 %	0.05 %	**pass**
Zone factor Z_H	2.395358	2.395367	3.6×10^{-6}	0.001 %	**pass**
Contact ratio factor Z_ε	0.803417	0.8033382	0.0098 %	0.03 %	**pass**
Helix angle factor Z_β	1.019444	1.019444	2.6×10^{-7}	0.001 %	**pass**
Lubricant factor Z_L	1.047386	1.047386	1.4×10^{-7}	0.001 %	**pass**
Velocity factor ZV	0.969114	0.9691142	2.1×10^{-7}	0.001 %	**pass**
Roughness factor Z_R	0.965987	0.9659871	6.0×10^{-8}	0.01 %	**pass**
Life factor Z_NT, pinion	0.910054	0.9100545	5.1×10^{-7}	0.001 %	**pass**
Life factor Z_NT, wheel	0.961759	0.9617587	3.3×10^{-7}	0.001 %	**pass**
Nominal contact stress σ_H0	1206.537 MPa	1206.423 MPa	0.0095 %	0.03 %	**pass**
Contact stress σ_H	1301.175 MPa	1301.205 MPa	0.0023 %	0.03 %	**pass**
Permissible contact stress σ_HP, pinion	1338.48 MPa	1338.48 MPa	3.5×10^{-7}	0.001 %	**pass**
Permissible contact stress σ_HP, wheel	1414.525 MPa	1414.525 MPa	3.6×10^{-7}	0.001 %	**pass**
Pitting safety S_H, pinion	1.02867	1.028646	0.0023 %	0.03 %	**pass**

Case	Reference	Axomek	Deviation	Tolerance	
Pitting safety S_H, wheel	1.087113	1.087088	0.0023 %	0.03 %	**pass**
Stress correction factor Y_S, pinion	1.824481	1.82465	0.0093 %	0.2 %	**pass**
Stress correction factor Y_S, wheel	2.075031	2.075072	0.002 %	0.2 %	**pass**
Surface factor Y_RrelT	0.963881	0.9638812	1.7×10^{-7}	0.001 %	**pass**
Notch sensitivity Y_δrelT, pinion	0.992311	0.9923107	3.2×10^{-7}	0.1 %	**pass**
Size factor Y_X	0.97	0.97	1.1×10^{-16}	10^{-9}	**pass**
Load distribution factor fε (2019) = 1/√εαn	0.7766742	0.7765982	0.0098 %	0.03 %	**pass**
Tooth form factor Y_F (2019), pinion	1.208972	1.208507	0.038 %	0.05 %	**pass**
Tooth form factor Y_F (2019), wheel	1.038071	1.037913	0.015 %	0.05 %	**pass**
Y_β (2019) at ε_β = 0, β = 30° (published reference table)	1.5396	1.539601	4.7×10^{-7}	0.01 %	**pass**
Y_β (2019) at ε_β = 0, β = 36° (published reference table)	1.5396	1.539601	4.7×10^{-7}	0.01 %	**pass**
Y_β (2019) at ε_β = 0.3, β = 24° (published reference table)	1.2329	1.232927	0.0022 %	0.01 %	**pass**
Y_β (2019) at ε_β = 0.5, β = 15° (published reference table)	1.0403	1.040255	0.0043 %	0.01 %	**pass**
Y_β (2019) at ε_β = 1, β = 15° (published reference table)	0.9709	0.970905	5.1×10^{-6}	0.01 %	**pass**
Y_β (2019) at ε_β = 1.6, β = 36° (published reference table)	1.1547	1.154701	4.7×10^{-7}	0.01 %	**pass**
fε at ε_β = 0, εαn = 1.5 (published reference table)	1	1	0 %	0.01 %	**pass**
fε at ε_β = 0, εαn = 2.2 (published reference table)	0.7	0.7	0 %	0.01 %	**pass**
fε at ε_β = 0.1, εαn = 1.5 (published reference table)	0.9832	0.9831921	8.1×10^{-6}	0.01 %	**pass**
fε at ε_β = 0.5, εαn = 1.5 (published reference table)	0.9129	0.9128709	0.0032 %	0.01 %	**pass**
fε at ε_β = 0.9, εαn = 1.9 (published reference table)	0.7574	0.7574194	0.0026 %	0.01 %	**pass**
fε at ε_β = 0.3, εαn = 2.1 (published reference table)	0.702	0.7020379	0.0054 %	0.01 %	**pass**
fε at ε_β = 0.5, εαn = 2.2 (published reference table)	0.6908	0.6908493	0.0071 %	0.01 %	**pass**

Case	Reference	Axomek	Deviation	Tolerance	
fε at ε_β = 1, εαn = 1.7 (published reference table)	0.767	0.766965	0.0046 %	0.01 %	pass
fε at ε_β = 1.3, εαn = 2.2 (published reference table)	0.6742	0.6741999	2.0×10^{-7}	0.01 %	pass

Gears: internal gear tooth root

Kind of reference	The same cases calculated by other programs or methods
Implements	ISO 6336-3, internal gears
Reference source	J. Němček, Sci. J. Silesian Univ. Technol., Ser. Transport 99 (2018), Table 1
Cases	8 of 8 within tolerance

Ring gears z_2 = −40 and −80 (m = 1, x = 0, 20°, standard basic rack) cut by a z_0 = 25 shaper with a sharp tip and loaded by a z_1 = 20 pinion, against the exact geometry in J. Němček, Scientific Journal of Silesian University of Technology, Series Transport 99 (2018), Table 1.

Case	Reference	Axomek	Deviation	Tolerance	
z_2 = −40: root chord sFn2	2.851 mm	2.851424 mm	0.015 %	0.5 %	pass
z_2 = −40: moment arm hFe2	1.159 mm	1.159038 mm	0.0033 %	0.5 %	pass
z_2 = −40: fillet radius ρF2	0.081 mm	0.08075098 mm	0.31 %	2.5 %	pass
z_2 = −40: form factor YF2	0.8539	0.8539154	0.0018 %	0.5 %	pass
z_2 = −80: root chord sFn2	2.729 mm	2.729441 mm	0.016 %	0.5 %	pass
z_2 = −80: moment arm hFe2	1.179 mm	1.179378 mm	0.032 %	0.5 %	pass
z_2 = −80: fillet radius ρF2	0.128 mm	0.1280775 mm	0.061 %	2.5 %	pass
z_2 = −80: form factor YF2	0.949	0.9490802	0.0085 %	0.5 %	pass

Gear mesh loss at a given coefficient of friction: Niemann's HV against the mesh mechanical advantage

Kind of reference	The same cases calculated by other programs or methods
Implements	Niemann's load-dependent mesh loss P·μ·HV, HV = π·(u + 1)/(z_1·u·cos βb)·(1 − ε_α + ε1² + ε2²)
Reference source	AGMA ISO 14179-1:2004, eq. 13 (PM = fm·T1·n_1·cos²βw/(9549·M)) on Annex C's first stage mesh (PM 0.196 kW at fm 0.0207, p. 24)
Cases	1 of 1 within tolerance

The engine's loss at the example's own fm (0.0207) against the example's PM. The two loss factors differ by method: eq. 13's cos²βw/M is π·(u + 1)/(z_1·u)·cos²βw·(ε1² + ε2²)/ε_α, which averages the sliding over the path of contact at a constant tooth force; Niemann's HV shares it between two pairs where two are in contact (1 − ε_α + ε1² + ε2²) and takes 1/cos βb. For this pair (ε_α 1.42, βb 18.1°) HV is 0.11876 against cos²βw/M 0.12064, 1.55 % lower, and the printed PM 0.196 kW is 0.1955 kW rounded up (0.24 %): together the 1.8 % deviation, which the 2 % tolerance covers (the test also checks that the deviation is exactly the ratio of the two loss factors). The coefficient of friction is not compared: the engine's μmz (Schlenk's formula as ISO/TR 14179-2 gives it, with η at 40 °C and Ra = R_z/6) is 0.0203, 0.0267 and 0.0318 for this pair at R_z 1, 3 and 6 μm, against the example's 0.0207 from eq. 18, a regression on viscosity, load intensity and velocity without roughness; at R_z 3 μm the engine's mesh loss is 29 % higher.

Case	Reference	Axomek	Deviation	Tolerance	
Mesh power loss PM at fm 0.0207 (eq. 13)	0.196 kW	0.1924615 kW	1.8 %	2 %	pass

Not validated against a published reference

- Loaded tooth contact analysis and the tooth root finite elements (compared only with ISO 6336 itself in the tests)
- The mesh loss’s coefficient of friction μmz: compared only with the AGMA ISO 14179-1 example’s, which another formula gives (the engine’s is 29 % higher at Rz 3 μm)
- Gearbox efficiency over its stages beyond the mesh loss’s path of contact, and thermal rating

Stated limits

Standards and scope **(help: modules/gears.md)**

ISO 6336-1:2019 (2006 with the 2006 edition; K_v by method B, K_Hβ by method C or given, K_Hα by method B), accuracy grades by ISO 1328-1:1995 or 2013, ISO 6336-2 and -3 in the 2019 or 2006 edition, materials of ISO 6336-5:2016, load spectra by ISO 6336-6 (Palmgren–Miner), scuffing by ISO/TS 6336-20:2022 (flash temperature, Annex A) and ISO/TR 13989-2:2000 (integral temperature), mesh loss by ISO/TR 14179-2 (Niemann and Schlenk). The rating is checked against the worked examples of ISO/TR 6336-30 (`axomek verify`).

Not covered: the size factor Z_X (taken as 1); no-load losses (churning, windage); tooth bending coupling between slices, edge effects and dynamics in the loaded tooth contact; housing deflection and the mesh misalignment from shaft deflection in a gearbox (enter it as **Mesh misalignment from deformation f_sh**). An internal pair's **Mesh** view lists the geometry without a drawing.

What is not covered **(help: theory/iso-6336.md)**

- Micropitting (ISO/TS 6336-22), tooth flank fracture (ISO/TS 6336-4) and wear are not rated.
- The size factor Z_X is 1.
- No-load losses (churning, windage) are not part of the mesh loss.
- In a gearbox model the mesh misalignment from shaft and housing deflection is not fed into K_Hβ automatically; enter it as `misalignment` or through the shaft layout.

Scuffing is rated separately: see Gear scuffing.

What is not covered **(help: theory/gear-scuffing.md)**

- Bulk temperatures from a heat balance of the gearbox: both methods use their approximation from the oil temperature.
- Bevel, worm and plastic gears: the scuffing check is offered for cylindrical gear pairs.
- Cold scuffing at very low speeds and wear are not rated.

Known limits

- Where a train's speeds come from its items and two of them turn one shaft opposite ways (a warning at the later one's sense, with its fix): - the pair the train took as driven the other way round is rated so meanwhile, and its mesh loss with it (the Two-stage helical gearbox with G2 alone reversed in its file: total 3 614 W against 3 610 W), though the power in and the efficiencies are not given; - a drive or pair added or mounted beside a train that turns its shaft negatively starts at its default sense, positive, and so gets the warning until it is reversed (the fix) rather than taking the train's sense; - a rack mesh's sense reversed is not carried on to the rest of its train (a gear pair's, worm pair's or drive's is), only warned of.

- Steering duty, for a later release (user test ut4): - A worm pair or a gear pair over a steering duty keeps its own required life (a worm pair rated at its worst state as if that lasted its life, a gear pair over the cases at its own life's cycles); the duty's length checks the ball nut's, the ball screws' and the bearings' lives. - Editing a mesh's Peak rack force for static rating in the duty's refusal card makes the duty not computed again, so the card (and the field in it) goes until Compute duty runs; with two refused meshes, the second is edited after recomputing, or on its mesh's own panel. - The drivetrain's natural frequencies (system modes) are dropped as out of date when another shaft is shown, though they do not depend on it; Compute them again. - While the duty computes, a check it decides reads "computing…" in every list of checks the app shows, also another report's (a version compared) of the same kind; the project's verdict line keeps the engine's count (a failure at the state counts) and says the duty is being computed. - A design search's work budget counts a remembered result of the steering duty at every request within one solve, so a design that runs on and on is stopped later than it could be (its time limit and Cancel still stop it); the engine's other remembered results (a gear tooth root's or a housing's finite elements) are not counted, which the budget's floor of 5000 steps covers.
- A teeth end (`teethEnd`) is rated by DIN 743-2's notch factors of a shaft with involute teeth without a hub (4.2.2) and an eq. (9) shoulder or groove of its shape, not by a published method for a hob's run-out, which no source at hand gives (see "Shaft strength references"); the mesh's own load on the teeth is the rack mesh's or gear pair's tooth-root rating. A worm's thread end takes none (a strength notch instead). The Shaft module's 3D draws a rack pinion's teeth cut into the shaft; a gear pair's pinion cut into its shaft is drawn there at its root diameter as before (the Assembly and Steering views draw its teeth). The 3D views draw a single toothed section's ends; teeth over several sections at the root diameter are rated over all of them but drawn on the one at the mesh.
- A ball screw whose nut turns (`turns: "nut"`) is mounted by its nut's shaft; the screw, which travels, is not one of the project's members, so what it pushes takes its thrust only in the user's own loads. A step the load drives back on a mounted screw whose shaft is joined to another (coupling, drive, gear pair) is refused: a load case cannot turn the train's power round (a later release).
- A plastic gear pair without a `peakTorque` has no peak load check (VDI 2736-2, 3.3): its peak safety is listed "not assessed" (no peakTorque given) against `limits.minPeakSafety` 1.5, so the project never passes `axomek check --strict`. Give the pair's `peakTorque`, or clear `limits.minPeakSafety` when no peak above the nominal torque occurs.
- A gear pair's mesh loss (ISO/TR 14179-2) takes the oil's viscosity at the pair's `oilTemperature`; a pair without one (every pair saved before it was an input) takes the oil at 40 °C, so in a hot sump its mesh loss comes out about 11 % low (ISO VG 220 at 95 °C): the efficiency slightly high, the heat slightly low. Give the oil's working temperature. The formula's uncertainty from the flank roughness is larger.
- A gear oil under about 6.5 mm²/s at 40 °C has no viscosity–temperature line at viscosity index 95 (ISO 2909): its scuffing check is refused at `viscosityIndex`, and with it the pair's curves, tooth contact and root FE, until the check is removed or the oil's own higher index is given. No gear oil is that thin; a later release keeps the other analyses.
- The full-round limit of the basic rack's root radius is checked for cylindrical pairs, not yet for a rack mesh's own rack.
- A ring gear's STEP solid is cut by its shaper cutter, so with a profile shift ($x_2 \neq 0$) its root circle differs a little from the report's nominal root diameter $d - 2 \cdot mn \cdot (hfP - x)$ (about a tenth of a millimetre on the diameter). The rating takes the nominal one.

- `axomek gear step` refuses an undercut pinion (z 16 or 17 at x 0) that the rating accepts: its root is not modelled.
- A steering duty's work is limited, so that neither its derivation nor its rating can run without end (0.3.0). Bounding the linked modules over the strokes stops at a fixed amount of work (the setup that ran for hours stops in some 35 s natively; every derivation the property tests' random linked setups and the steering tests decide takes at most some 20 s; the limits allow at most some 2.5 min, for a gear pair whose true-safety search is slow throughout): past it the load cases are the strokes' own, without the linked modules' refinement, every linked module's checks over them are undecided (never a plain pass) and `steering.duty` warns why and what to change. The tooth-by-tooth rating stops at its own limit: past it the duty is refused at `steering.duty`, saying what to change, as a search cut short would understate the teeth's damage. A duty on a constant lead, or with its tooth geometry within ISO 6336's validated range, rates in a few seconds (the steering-duty template in 2.7 s natively). Only a duty with FE root assessments (the geometry outside ISO's range) over a variable lead is slow: each new lead needs its own assessment (some 130 ms), so its rating can take some 52 s natively before the limit, and with its derivation a solve of such a duty up to about 90 s natively, longer in the browser; a 0.1 mm window there takes some 22 s. To make it fast, use a constant lead or bring the tooth geometry within ISO's range.
- A gear's STEP solid has nominal teeth: at the profile shifts, without the backlash of generating shifts (`details.rootShifts`), a protuberance tool's residual undercut or flank modifications (tip relief, crowning), which the rating takes. A double-helical gear is refused, as the project does not hold the gap between its helices: export one helix (doubleHelical off, half of `faceWidth` and of both `faceWidths` when given) and mirror it in CAD. In a later release: the gap as an input and both helices in one solid.
- A steering duty whose strokes reverse the rack force (a return pass against a resisting load, a signed load below zero) is refused at `steering.duty` beside anything the load cases reach that a case cannot reverse (0.3.0): a mounted gear or worm pair or rack mesh, a drive, a coupling, a rotor, a bearing with loads of its own, a gear pair rated over the cases. The derived cases reverse only the entered shaft loads and ball screw forces, so the others would be rated at the forward loads on the reversed passes (a rotor's torque, unreversed, cancelled the reversed input in one review's case: 18.444302 where 9.222151 is right). In gearbox-single-stage with a rack mesh and resisting strokes with return passes, BR1 passed 5.5 at 5.778100 where the reversed pass gives 5.260948. One-way strokes are rated; reversing stages over the duty come in a later release.
- Over steering-derived load cases a linked gear pair is undecided where it passes (0.3.0) when its KHβ comes from its solved shafts (`misalignmentFromShafts`), or when the connections resolve the project (a shaft speed, a train-mode stage or a coupling): the solve then rates each case's bin at the torque and speed that case's train gives the pair, the duty's bounding at the pair's torque times the case's factor. A standalone pair, or one in a project with no shaft speed, train-mode stage or coupling, is bounded as before. (A project of given stages with a shaft speed is marked too, where the two ratings agree: on the safe side.)
- A gear pair linked to the load cases whose nominal rating holds but whose rating over them is refused (a case's bin it cannot rate, such as a negative factor on a given pair) is listed in the load cases' summary as rated over them, though it has no spectrum checks and its error says why (0.3.0; Sol review of R-938).

- A steering duty over a steering system (0.3.0) rates the rack meshes' teeth, and with strokes the system's shafts, the bearings linked on them that take the load cases, its worm pairs and its ball nut over its states (see Steering system). Not rated over the duty (a later release): a mounted gear pair (rated at the system's `state`, its checks not rated over the duty, never a pass: its rating takes its solved shafts' KHβ, which the states do not give it yet), and likewise fits, keys and housings. A rotor's air gap is not rated over the states (its band between them is not drawn; never a pass: rate it at the duty's heaviest state). A bearing's warning over the states (a minimum load undecided at its valley) is never satisfied, and its gyroscopic slip (the high-speed model's) is the system's state's, a pass not rated over the duty. Beside such a duty a design search's metrics are refused but the shaft's mass (their values over the states have no bound between them, and some items' results are the system's state's): search with requireChecks, whose checks take the bands. The ball nut's life is the states' Miner sum (as a bearing's), its static safety the worse of the states' and the system's own state's. Left at the system's `state`: the couplings, a bearing with `useLoadCases` false, and everything at a misuse event (the events rate the pinions only; the elements keep their own `peakFactor` for a static peak, which over the duty multiplies every state's loads, as over load cases). A worm pair and a drive take the worst of their states, each rated as if it lasted their life: on the safe side, as DIN 3996 holds no load spectrum here. A shaft's states are each rated with their stresses fully reversed (as a mounted ball screw's states are), on the safe side where a stroke does not turn round. The band of a bearing's checks is drawn from its life, static safety, contact stress and thermal speed in the states, relative to each check's utilisation. Two supports in seats of one housing (whose seats push each other with the bearings' loads) leave the elements unrated over the duty (a warning at `steering.duty`, their checks never a pass): the states do not couple the seats again (a later release). The project's own load cases beside such a duty rate the shafts and bearings instead (a warning at `steering.duty`), and their checks are then not rated over the duty (never a pass): remove the load cases to rate them over it. The elements' states are some four times the duty points (284 for the Dual-pinion EPS's duty), each a state solve and a solve of every shaft: at most 1200 (`system_cases::CASE_LIMIT`) and at most 12,000 solves (`WORK_LIMIT`: each state one, and one for each shaft and each bearing on a shaft; some 25 s natively and 40 s in the browser), past which the elements are not rated over the duty (a warning at `steering.duty`; their checks never pass) while the teeth are. Measured under load on the build machine (one process, natively): the Dual-pinion EPS's duty 9.3 s for the teeth and 19 s in all; the Belt-driven ball-nut and Column EPS examples given its duty 2.4 s and 2.8 s for the teeth, 7.0 s and 5.2 s in all (168 and 88 states); the browser's engine took about the same under that load (17.9 s, 7.6 s and 6.3 s). It refuses a stroke's `torque` table (the system splits the rack force, not a pinion's torque); a rack mesh off the system's rack is not rated over the duty (a warning; it keeps its rating at its own load). Each duty point is a state solve (some 15 to 30 ms natively), at most 4000 per duty. Rating the teeth of two pinions is the slow part: the Dual-pinion EPS template's duty (a parking, a manoeuvring and an on-centre stroke, both ways) takes some 4.7 s natively and 7.3 s in the browser's engine on the build machine (some 20 s natively on a laptop, 4 s of it the state solves); the app computes it on request, with progress and Cancel (0.3.0), and solves each edit without it. Positions past the rack's travel are refused (each point is a state of the system, within its stops). The lift is rated within 1e-4 mm of the states' between points (the contact ratios within some 4e-5), not raised; the forces are raised to bound theirs. Misuse events are rated at standstill (vehicle speed 0, forward efficiencies, no friction).

- A gear pair's loaded tooth contact analysis loads the flanks with the nominal tooth force but opens them by the misalignment that ISO 6336 method C takes at Fm = Ft·KA·Kv (the shaft layout's fsh, the solved shafts' or a carrier's), so with KA·Kv above 1 its pressure map and its face load factor show more misalignment than the nominal load gives. On the safe side; the ISO rating is not affected. Both at one load come in a later release.
- A helical pair's loaded tooth contact analysis runs the contact lines across the face the same way whatever the helix's hand. The load across the face (KHβ) and the transmission error do not depend on it while the pinion's and the wheel's profile reliefs are alike, but the pressure map's slant and its peak pressure, which follows the flanks' curvature along the path, can differ a little for a pair of the other hand. Taking the hand into account comes in a later release.
- The scuffing flash temperature's maximum (flash temperature method, ISO/TS 6336-21) is the largest of 65 points along each piece of the path of contact between the load sharing's kinks, refined by golden section between that point's neighbours: a second peak, narrower than 1/64 of a piece and lower at every point than the largest, would be missed. The load sharing's curves have no such peaks.
- **Add a steering** puts the rack's meshes in the steering's train and takes the torques off the driver, the shafts coupled to it and the pinions' shafts. A gear pair, worm pair or drive between the driver and the rack still rated at its own torque is left as it is. The steering then reads "not solved", and its error offers the fix (`torqueMode train`). Apply that fix, or set it in the item's module.

Bevel gears: ISO 10300

Virtual cylindrical gears, load factors and pitting and tooth root strength of bevel gears by ISO 10300.

Methods and standards

Bevel pairs without hypoid offset (straight, skew, spiral and Zerol): mean cone geometry, shaft forces at the mean diameter, and the general influence factors and virtual cylindrical gears of ISO 10300-1:2014 method B1.

Virtual gears after Annex A: $d_v = d_m/\cos\delta$, helix angle $\beta_v = \beta_m$, $\alpha_{vet} = \arctan(\tan\alpha_n/\cos\beta_v)$, $z_v = d_v/m_{vt}$, base pitch, path of contact $g_{v\alpha}$ from the tip diameters $d_v + 2\cdot h_{am}$ (basic rack 1/1.25·m_{mn} at mid face, profile shifts $x_{hm1} = -x_{hm2}$), $b_{v,eff} = b_{2,eff}$ (the loaded contact pattern, 0.85·b unless given) and $b_v = b$, $\varepsilon_{v\alpha} = g_{v\alpha}/p_{vet}$, $\varepsilon_{v\beta} = b_{v,eff}\cdot\sin\beta_v/(\pi\cdot m_{mn})$, $\varepsilon_{v\gamma} = \varepsilon_{v\alpha} + \varepsilon_{v\beta}$ and $\varepsilon_{v\alpha n} = \varepsilon_{v\alpha}/\cos^2\beta_{vb}$; lengths of the tip, middle and root contact lines in the zone of action (Table A.2, reduced by the crowning correction C_{lb}), the contact line inclination β_B and the relative curvature $\rho_{rel} = |\rho_t|\cdot\cos^2\beta_B$.

Load factors: K_A as given (Annex C guide values); K_v by method B (resonance speed of the pair's reduced mass $\pi\cdot\rho\cdot d_m^2/(8\cdot\cos^2\alpha_n)$ on $c_\gamma = 20\cdot C_F$ N/(mm·µm), $c' = 14\cdot C_F$, $f_{p,eff} = f_{pt} - y_\alpha$, influence factors c_{v1} to c_{v7} by $\varepsilon_{v\gamma}$ in the subcritical, main resonance, intermediate and supercritical ranges, formula (19) taken as printed with its +1) or method C (empirical curves by accuracy grade and outer pitch line velocity); $K_{H\beta} = 1.5\cdot K_{H\beta\text{-be}}$ with the mounting factor of Table 4 (bearing arrangement and how the contact pattern is checked) and $K_{F\beta} = K_{H\beta}/K_{F0}$ with the lengthwise curvature factor K_{F0} from the cutter radius (face milled) or blade groups (face hobbed), within 1 to 1.15 (method C, part 1 has no method B for these); $K_{H\alpha} = K_{F\alpha}$ by method B from f_{pt} (the wheel's ISO 17485 tolerance at the tolerance diameter for the accuracy grade unless given) less the running-in allowance y_α, and $F_{mtH} = F_{vmt}\cdot K_A\cdot K_v\cdot K_{H\beta}$ over b_v.

Pitting by ISO 10300-2:2014 method B1: $\sigma_{H0} = \sqrt{(F_n/(l_{bm}\cdot\rho_{rel}))}\cdot Z_{M\text{-}B}\cdot Z_{LS}\cdot Z_E\cdot Z_K$ with $F_n = F_{mt1}/(\cos\alpha_n\cdot\cos\beta_m)$, the mid-zone factor $Z_{M\text{-}B}$ from the curvature at the inner point of single contact of the pinion ($\varepsilon_{v\beta} = 0$), the middle of the path of contact ($\varepsilon_{v\beta} \geq 1$) or in between, the load sharing factor Z_{LS} from semi-elliptical loads on the tip, middle and root contact lines with peak loads 1 − |f/f_max|^e (e 1.5 for high, 3 for low profile crowning), which also limits $K_{H\alpha}$ to $\varepsilon_{v\gamma}/(\varepsilon_{v\alpha}\cdot Z_{LS}^2)$, and $Z_K = 0.85$; $\sigma_H = \sigma_{H0}\cdot\sqrt{(K_A\cdot K_v\cdot K_{H\beta}\cdot K_{H\alpha})}$, common to pinion and wheel; $\sigma_{HP} = \sigma_{H,lim}\cdot Z_{NT}\cdot Z_X\cdot Z_L\cdot Z_v\cdot Z_R\cdot Z_W\cdot Z_{Hyp}$ with Z_{NT} through 1.6 at 10^5, 1.0 at $5\cdot10^7$ and 1.0 at 10^{10} load cycles of each gear (real ratio), $Z_X = 1$, Z_L, Z_v, Z_R by method B for the softer material at the mean pitch line velocity and $R_{z10} = R_z\cdot\sqrt[3]{(10/\rho_{rel})}$, Z_W from the softer flank's Brinell hardness when given, $Z_{Hyp} = 1$ without offset; the bevel slip factor of the informative Annex A is not used.

ISO 10300-3 is not implemented: the tooth root stress comes from the virtual pair rated by the ISO 6336-3 code with these factors ($b_{v,eff}$ wide, the pinion's load cycles; part 3's own factors are not applied).

The contact must stay on the involutes of both virtual cylindrical gears, as for cylindrical pairs: the start of active profile of each (where the mate's tip at $d_v + 2\cdot h_{am}$ meets it) at or above its form circle (ISO 21771, from the basic rack 1/1.25/0.38·m_{mn} with x_{hm1} on the pinion and $-x_{hm1}$ on the wheel, z_v and β_v as above); an undercut or interfering pair is refused with the pinion profile shift x_{hm1} that clears it (searched within ±0.8, rounded outwards to 0.01) or, when none does, more pinion teeth.

A pinion of few teeth needs a positive shift: the starter's 15/45 pair ($z_{v1} = 15.8$, undercut without shift) has $x_{hm1} = 0.41$, the long-addendum proportion of straight bevel gears, pinion addendum $(1.46 - 0.46/u^2)\cdot m$ and wheel addendum $(0.54 + 0.46/u^2)\cdot m$, i.e. $x_{hm1} = 0.46\cdot(1 - 1/u^2)$.

Hypoid offsets are refused.

Validation

28 of 28 cases within their tolerance, in 1 groups.

The same cases calculated by other programs or methods

Bevel gears: ISO 10300-1 virtual gears and load factors, ISO 10300-2 pitting

Kind of reference	The same cases calculated by other programs or methods
Implements	ISO 10300-1/-2:2014
Reference source	Tredgold’s approximation, and an independent evaluation of ISO 10300-1 Annex A and formulae (9)–(15), (21)–(24), (27)–(30), (38); ISO 10300-2 method B1
Cases	28 of 28 within tolerance

15/45 bevel pair at Σ = 90°, m_et = 4 mm, b = 30 mm, α_n = 20°, 150 N·m at 1500 rpm, K_A = 1.25, case-hardened steel, accuracy grade 7: straight teeth with the starter's pinion profile shift x_hm1 = 0.41 against Tredgold's approximation (d_v1 = d_m1·√(u²+1)/u, z_v = z/cos δ), the 35° spiral version without profile shift (face milled, r_c0 = 63.5 mm) against an independent evaluation of the Annex A relations and of formulas (9) to (15), (27) to (30) and (38), and of ISO 10300-2 method B1 (contact lines clipped from the zone of action, (1) to (10), (53) to (59)); Table 4 and formulas (21) to (24) evaluated directly.

Case	Reference	Axomek	Deviation	Tolerance	
straight: d_v1 = d_m1·√(u²+1)/u	53.24555 mm	53.24555 mm	0 %	10^{-9}	**pass**
straight: z_v1 = z_1·√(u²+1)/u	15.81139	15.81139	1.1×10^{-16}	10^{-9}	**pass**
straight: z_v2 = z_2·√(u²+1)	142.3025	142.3025	0 %	10^{-9}	**pass**
spiral: α_vet	23.9568 °	23.9568 °	10^{-8}	0.001 %	**pass**
spiral: β_vb	32.61461 °	32.61461 °	4.5×10^{-9}	0.001 %	**pass**
spiral: p_vet	9.668053 mm	9.668053 mm	3.8×10^{-8}	0.001 %	**pass**
spiral: g_vα	12.27081 mm	12.27081 mm	1.7×10^{-8}	0.001 %	**pass**
spiral: ε_vα	1.269212	1.269212	1.8×10^{-7}	0.001 %	**pass**
spiral: ε_vβ	1.687733	1.687733	1.9×10^{-8}	0.001 %	**pass**
spiral: ε_vαn	1.788896	1.788896	2.0×10^{-7}	0.001 %	**pass**
spiral: β_B	13.46783 °	13.46783 °	1.7×10^{-8}	0.001 %	**pass**
spiral: l_bm	20.98966 mm	20.98966 mm	5.9×10^{-9}	0.001 %	**pass**
spiral: f_pt (ISO 17485, B = 7)	17.76868 µm	17.76868 µm	2.6×10^{-9}	0.001 %	**pass**
spiral: n_E1	31777.07 rpm	31777.07 rpm	1.4×10^{-10}	0.001 %	**pass**
spiral: K_v-B	1.026706	1.026706	2.2×10^{-7}	0.001 %	**pass**
spiral: K_Hβ-C	1.65	1.65	1.3×10^{-16}	0.001 %	**pass**
spiral: K_F0	1.064038	1.064038	7.4×10^{-8}	0.001 %	**pass**
spiral: K_Fβ-C	1.550696	1.550696	2.5×10^{-7}	0.001 %	**pass**
spiral: K_Hα-B	1.260856	1.260856	8.5×10^{-8}	0.001 %	**pass**
spiral: Z_LS (ISO 10300-2)	0.863217	0.8632166	4.7×10^{-7}	0.001 %	**pass**
spiral: Z_M-B	1.020141	1.020141	7.0×10^{-8}	0.001 %	**pass**

Case	Reference	Axomek	Deviation	Tolerance	
spiral: σ_H0	801.4993 MPa	801.4993 MPa	1.2×10^{-10}	0.001 %	**pass**
spiral: σ_H	1309.652 MPa	1309.652 MPa	2.7×10^{-10}	0.001 %	**pass**
spiral: σ_HP	1466.449 MPa	1466.449 MPa	2.5×10^{-10}	0.001 %	**pass**
straight: Z_M-B	1.005854	1.005854	3.9×10^{-7}	0.001 %	**pass**
straight: σ_H	1359.341 MPa	1359.341 MPa	2.9×10^{-10}	0.001 %	**pass**
Table 4: sample set, both cantilevered	1.5	1.5	0 %	exact	**pass**
K_v-C, B = 7 at 10 m/s	1.269404	1.269404	0 %	10^{-12}	**pass**

Not validated against a published reference

- No worked example of ISO 10300 is compared: the cases are an independent evaluation of its formulae
- Tooth root stress (rated by ISO 6336-3 on the virtual pair; ISO 10300-3 is not implemented)

Stated limits

Standards and scope **(help: modules/bevel.md)**

ISO 10300-1:2014 method B1 geometry with the virtual cylindrical gears of Annex A, K_A (Annex C guide values), K_v by method B or C, K_Hβ and K_Fβ by method C, K_Hα = K_Fα by method B; pitting by ISO 10300-2:2014 method B1 (clause 6) with Z_L, Z_v, Z_R, Z_W and Z_NT of clause 8.

Not covered:

- ISO 10300-3 (tooth root) is not implemented. The root safety is that of the virtual cylindrical pair rated by ISO 6336-3 in its 2006 edition (the 2019 edition's Y_β and fε are not applied) with the ISO 10300-1 load factors and each gear's life factor at its own load cycles (the wheel's at the real ratio, as for pitting); part 3's own factors are not applied.
- Hypoid offsets are refused (the hypoid geometry of ISO 23509 is not implemented); Z_Hyp is 1.
- The bevel slip factor of ISO 10300-2's informative Annex A, scuffing and mounting on the project's shafts.

Known limits

- A bevel pair's entered single pitch deviation f_pt of exactly 0 gives an accuracy grade of $-\infty$ for K_v by method C (K_v 1, with a note that the grade is extrapolated); a measured f_pt is never 0.

- The shaft strength check (DIN 743) takes DIN 743-2's K1 alone, as the standard prescribes where the strength at the part's size is not known (6.2): for an alloy steel inside a size class it can lie above the class's guaranteed minimum (42CrMo4 +QT at Ø50: 748.6 MPa against EN 10083-3's 650 MPa), as in any calculation after DIN 743. Enter the guaranteed strengths ("strengths as given for the size") for a shaft near its limit. Press fits hold it to the class (see "From 0.3.0"). Keys of form AB, C to J and lift-out bevels are not in the app, nor a keyway shorter than its key: the key's length stands for both, where DIN 6892 takes the shorter for l_tr and K_λ. A spline's K_λ is still the estimate of 1 up to a length of one diameter, + 0.4 per further diameter: DIN 6892 is for parallel keys only, and DIN 5466-1's factors are not taken.

Worm gears: DIN 3996, ISO/TR 14521 and VDI 2736-3

Cylindrical worm gear pairs of steel and bronze by DIN 3996 and ISO/TR 14521, and plastic worm wheels by VDI 2736-3.

Methods and standards

Worm pairs after DIN 3996:2012-09, method C: the characteristic values p*m, h* and s* of the geometry (Gl. 10 to 15, flank form C its own; each may be given from a method B analysis); mean flank pressure σHm = (4/π)·√(p*m·T2·10³·Ered/a³); mean friction μzm = μ0T·YS·YG·YW·YR with the reference gear's base value by wheel material, oil (mineral, polyalphaolefin, polyglycol) and dip or injection lubrication over the sliding speed, YS with a held to 65…250 mm; mesh efficiencies worm and wheel driving, the starting one with μ0T at zero sliding speed; losses of the mesh, idling (7.3.4, Gl. 37 unless given), the bearings under load (Gl. 40 adjusted, Gl. 41 locating/floating) and the shaft seals, the overall efficiency wheel driving (η′z·P2 less the idling, bearing and seal losses, over P2; 0 where they take all the mesh passes back); the mesh forces Ftm2 = 2000·T2/dm2, Ftm1 = Ftm2·tan(γm + ρz) and Fr = Ftm2·tan α_0/cos γm, the worm torque from the input power with all losses; oil sump temperature by a heat balance with the housing surface (method B) or the standard's method C for ribbed housings with or without fan, injection by the oil's cooling power; wheel bulk temperature, the oil's viscosity there from the line through ν_40 and ν100 (the standard's constants unrounded) and the mean film thickness hmin,m (Dowson–Higginson as in Gl. 18); wear δWn = JW·sWm with the reference wear intensity J0T(KW = hmin,m·WS·WH), the material–lubricant and start factors, against the pointed wheel tooth tip, the root safety 1.1 of the worn tooth, a mass loss or 0.3·mx·cos γm for backlash (the chosen one, never above the first two); pitting σ_HG = σHlimT·Zh·Zv·Zs·Zu·Zoil with the life to SH = 1; tooth root shear τF with YF from the root chord reduced by the wear and the rim factor YK, against τFlimT (reduced unless a loss of wheel quality is accepted) times YNL; worm deflection (Gl. 101) against 0.04·√mx (Gl. 103).

The worm deflection is Gl. (101) as printed (10.3.3): its constant holds a steel worm's modulus, so the worm modulus does not enter, and 1.1·df1 is its equivalent diameter.

Checked against the standard's three worked examples (axomek verify).

Plastic worm wheels (POM, PA 66, PA 46, PEEK or an own plastic) driven by a steel worm at 90° after VDI 2736-3:2014, not by the method for bronze and cast iron wheels: geometry from the base helices and the operating helix circles, the normal contact ratio by eq. 11 and 12 unless given; the design load value C of eq. 1 to 7 against C_lim of Figure 4 (POM and PA 66 only, up to 10 m/s sliding speed, held at 0.5 m/s below it), S_C = C_lim/C; the wheel's root shear τ_F of eq. 8 to 21 against τ_Flim, 0.75 of the tensile fatigue strength σ_ZSch of Figures 6 to 9 at the wheel temperature and the load cycles, unless given; the flank pressure by Hertz, eq. 22 to 34, against the permissible flank pressure of a comparable cylindrical gear after VDI 2736-2 (σ_HlimN over S_Hmin 1.4, or 1.25 for intermittent duty), which 3.2.2 allows as an estimate and cautions gives somewhat low safeties, or a flank limit given; mesh and overall efficiencies by eq. 35 to 37, the friction μ from Table 1's upper value (dry: VDI 2736-2 Table 2) unless given.

The wheel bulk temperature ϑ_M is an input: VDI 2736-3 gives no method to compute it, so its check compares the temperature you enter with the material data's limit (continuous, or the hottest data curve for intermittent duty) and does not show that the wheel stays that cool.

Wear is not rated, as VDI 2736-3 does not quantify it: the expected flank wear W_n you enter (0 by default, the teeth unworn) is taken off the teeth for the root shear only.

There is no heat balance, oil sump temperature, wear intensity, pitting life or worm deflection check for a plastic wheel, as there is for a metal one.

Validation

314 of 314 cases within their tolerance, in 8 groups.

Worked examples of standards and books

Worm gears: DIN 3996 Anhang I

Kind of reference	Worked examples of standards and books
Implements	DIN 3996:2012
Reference source	DIN 3996:2012, Anhang I, examples
Cases	34 of 34 within tolerance

The three worked examples of DIN 3996:2012-09, Anhang I (flank form I, CuSn12Ni2-C-GZ wheel, polyglycol): 1st the standard reference gear (a 100 mm, 41:2, m_x 4, q 9) at 4.5 kW and 1500 rpm with dip lubrication and a fan; 2nd a 65 mm, 40:1 pair at 300 N·m and 150 rpm worn to a pointed wheel tooth tip at S_W = 1.1 (12 065 h); 3rd a 400 mm, 49:4 pair at 13 000 N·m and 3000 rpm with injection lubrication, life to S_H = 1. Tolerances cover the examples' rounding of printed and intermediate values. The 3rd example takes Y_S = 0.5 although 7.4.5 holds a to 250 mm (Y_S = 0.632); its losses are checked with the example's own friction.

Case	Reference	Axomek	Deviation	Tolerance	
Example 1: p*m (Gl. 10)	0.94703	0.9470301	1.4×10^{-7}	0.01 %	**pass**
Example 1: h* (Gl. 12)	0.06918	0.06917922	0.0011 %	0.01 %	**pass**
Example 1: mean flank pressure	368.52 MPa	368.5185 MPa	4.2×10^{-6}	0.01 %	**pass**
Example 1: mean friction μzm	0.0234	0.02340305	0.013 %	0.5 %	**pass**
Example 1: mesh efficiency	90 %	90.00154 %	0.0017 %	0.1 %	**pass**
Example 1: overall efficiency	84.7 %	84.87358 %	0.2 %	0.3 %	**pass**
Example 1: oil sump temperature	73.2 °C	73.22558 °C	0.035 %	0.1 %	**pass**
Example 1: wheel bulk temperature	77.1 °C	77.06761 °C	0.042 %	0.1 %	**pass**
Example 1: film thickness hminm	0.246 μm	0.2463357 μm	0.14 %	0.3 %	**pass**
Example 1: wear intensity JW·10^{11}	89.857	89.85398	0.0034 %	0.1 %	**pass**
Example 1: flank loss δWn	0.731 mm	0.7307433 mm	0.035 %	0.2 %	**pass**
Example 1: wear safety	1.6	1.603059	0.19 %	0.5 %	**pass**
Example 1: pitting safety	1.2	1.201475	0.12 %	0.5 %	**pass**
Example 1: deflection safety	2.63	2.631759	0.067 %	0.5 %	**pass**
Example 1: root shear stress (worn)	35.522 MPa	35.52161 MPa	0.0011 %	0.1 %	**pass**
Example 1: root safety	2.53	2.533669	0.15 %	0.5 %	**pass**
Example 1: temperature safety	1.37	1.365643	0.32 %	0.5 %	**pass**
Example 2: mean flank pressure	506.5 MPa	506.4981 MPa	3.7×10^{-6}	0.01 %	**pass**
Example 2: mesh efficiency	62.5 %	62.4974 %	0.0042 %	0.1 %	**pass**
Example 2: overall efficiency	58.9 %	58.90083 %	0.0014 %	0.2 %	**pass**
Example 2: oil sump temperature	44 °C	43.96509 °C	0.079 %	0.2 %	**pass**
Example 2: wheel bulk temperature	51.2 °C	51.19812 °C	0.0037 %	0.1 %	**pass**
Example 2: film parameter KW	0.0845	0.08454825	0.057 %	0.6 %	**pass**
Example 2: wear intensity JW·10^{8}	4.371	4.370922	0.0018 %	0.1 %	**pass**
Example 2: pointed-tip limit δWlimn	2.1 mm	2.099218 mm	0.037 %	0.2 %	**pass**

Case	Reference	Axomek	Deviation	Tolerance	
Example 2: flank loss δWn at 12 065 h	1.91 mm	1.908297 mm	0.089 %	0.3 %	**pass**
Example 2: wear life to the pointed tip at SW 1.1	12065 h	12065.52 h	0.0043 %	0.2 %	**pass**
Example 3: p*m (Gl. 10)	1.026	1.025881	0.012 %	0.05 %	**pass**
Example 3: h* (Gl. 12)	0.0591	0.0591191	0.032 %	0.1 %	**pass**
Example 3: mean flank pressure	225.57 MPa	225.5673 MPa	0.0012 %	0.01 %	**pass**
Example 3: life at S_H = 1	10890 h	10891 h	0.0092 %	0.2 %	**pass**
Example 3: mesh loss (example's µzm)	10.04 kW	10.04273 kW	0.027 %	0.2 %	**pass**
Example 3: total loss (example's µzm)	14.76 kW	14.7589 kW	0.0074 %	0.2 %	**pass**
Example 3: overall efficiency (example's µzm)	95.8 %	95.76079 %	0.041 %	0.1 %	**pass**

Worm gears: ISO/TR 14521 Annex J

Kind of reference	Worked examples of standards and books
Implements	ISO/TR 14521:2010
Reference source	ISO/TR 14521:2010, Annex J, examples
Cases	142 of 142 within tolerance

ISO/TR 14521:2010 J.1–J.5 cross-checked with DIN 3996:2012. Only shared calculations are pass/fail checks, with printed-rounding tolerances. Information below records different methods, downstream effects and source inconsistencies; it is not a TR rating. Inputs and the full signed comparison are in docs/testing/worm-tr14521.md. J.3 uses its exact stated diameters; its unspecified cooling is closed at 50 C, 40 l/min, 4 K and no thermal/wear results from that closure are compared. Information — J.1: p*m: TR 9.496000000e-1 , DIN 9.470301352e-1 , difference -2.569864761e-3; TR 7.3.1(c), 7.3.2(c), eqs. (55), (58) apply face-width factors; DIN 6.4.2.3, 6.4.3.3, eqs. (10)–(13) omit them. Information — J.1: h*: TR 6.891000000e-2 , DIN 6.917921578e-2 , difference +2.692157780e-4; TR 7.3.1(c), 7.3.2(c), eqs. (55), (58) apply face-width factors; DIN 6.4.2.3, 6.4.3.3, eqs. (10)–(13) omit them. Information — J.1: sigmaHm: TR 3.690200000e2 MPa, DIN 3.685184464e2 MPa, difference -5.015536442e-1; TR 7.3.1(c), 7.3.2(c), eqs. (55), (58) apply face-width factors; DIN 6.4.2.3, 6.4.3.3, eqs. (10)–(13) omit them. Information — J.1: muzm: TR 2.300000000e-2 , DIN 2.340305136e-2 , difference +4.030513607e-4; TR 7.3.1(c), 7.3.2(c), eqs. (55), (58) apply face-width factors; DIN 6.4.2.3, 6.4.3.3, eqs. (10)–(13) omit them. Information — J.1: YG: TR 1.008000000e0 , DIN 1.005914811e0 , difference -2.085188892e-3; TR 7.3.1(c), 7.3.2(c), eqs. (55), (58) apply face-width factors; DIN 6.4.2.3, 6.4.3.3, eqs. (10)–(13) omit them. Information — J.1: etaMesh: TR 8.998000000e1 %, DIN 9.000153476e1 %, difference +2.153475963e-2; TR 7.3.1(c), 7.3.2(c), eqs. (55), (58) apply face-width factors; DIN 6.4.2.3, 6.4.3.3, eqs. (10)–(13) omit them. Information — J.1: PV: TR 8.050000000e-1 kW, DIN 8.020034005e-1 kW, difference -2.996599482e-3; face-width friction and bearing equation choice; TR 7.3.2, 8.2.2 / DIN 6.4.3.3, 7.3.5. Information — J.1: PVmesh: TR 4.780000000e-1 kW, DIN 4.773834091e-1 kW, difference -6.165908633e-4; TR 7.3.1(c), 7.3.2(c), eqs. (55), (58) apply face-width factors; DIN 6.4.2.3, 6.4.3.3, eqs. (10)–(13) omit them. Information — J.1: PVbearing: TR 1.280000000e-1 kW, DIN 1.260000000e-1 kW, difference -2.000000000e-3; TR 8.2.2 eq. (82), like DIN 7.3.5 eq. (38), uses the rounded coefficient 0.03; the engine uses DIN eq. (40), coefficient 0.028 at the reference gear. Information — J.1: etaOverall: TR 8.480000000e1 %, DIN 8.487357816e1 %, difference +7.357815653e-2; face-width friction and bearing equation choice; TR 7.3.2, 8.2.2 / DIN 6.4.3.3, 7.3.5. Information — J.1: thetaM: TR 7.720000000e1 C, DIN 7.719567218e1 C, difference -4.327817370e-3; TR 7.3.1(c), 7.3.2(c), eqs. (55), (58) apply face-width factors; DIN 6.4.2.3, 6.4.3.3, eqs. (10)–(13) omit them. Information — J.1: nuM: TR 6.507000000e1 mm2/s, DIN 6.508225791e1 mm2/s, difference +1.225790954e-2; TR 7.3.1(c), 7.3.2(c), eqs. (55), (58) apply face-width factors; DIN 6.4.2.3, 6.4.3.3, eqs. (10)–(13) omit them. Information — J.1: WS: TR 2.630000000e0 , DIN 2.626567803e0 , difference -3.432197322e-3; TR 7.3.1(c), 7.3.2(c), eqs. (55), (58) apply face-width factors; DIN 6.4.2.3, 6.4.3.3, eqs. (10)–(13) omit them. Information — J.1: hmin: TR 2.450000000e-1 um, DIN 2.457164220e-1 um, difference +7.164219630e-4; TR 7.3.1(c), 7.3.2(c), eqs. (55), (58) apply face-width factors; DIN 6.4.2.3, 6.4.3.3, eqs. (10)–(13) omit them. Information — J.1: sWm: TR 8.143610000e5 m, DIN 8.132564893e5 m, difference -1.104510672e3; TR 7.3.1(c), 7.3.2(c), eqs. (55), (58) apply face-width factors; DIN 6.4.2.3, 6.4.3.3, eqs. (10)–(13) omit them. Information — J.1: KW: TR 6.430000000e-1 , DIN 6.453908425e-1 , difference +2.390842517e-3; TR 7.3.1(c), 7.3.2(c), eqs. (55), (58) apply face-width factors; DIN 6.4.2.3, 6.4.3.3, eqs. (10)–(13) omit them. Information — J.1: J0T: TR 5.187000000e-10 , DIN 5.146871401e-10 , difference -4.012859887e-12; TR 7.3.1(c), 7.3.2(c), eqs. (55), (58) apply face-width factors; DIN 6.4.2.3, 6.4.3.3, eqs. (10)–(13) omit them. Information — J.1: JW: TR 9.076000000e-10 , DIN 9.007024952e-10 , difference -6.897504802e-12; TR 7.3.1(c), 7.3.2(c), eqs. (55), (58) apply face-width factors; DIN 6.4.2.3, 6.4.3.3, eqs. (10)–(13) omit them. Information — J.1: deltaWn: TR 7.390000000e-1 mm, DIN 7.325021492e-1 mm, difference -6.497850826e-3; TR 7.3.1(c), 7.3.2(c), eqs. (55), (58) apply face-width factors; DIN 6.4.2.3, 6.4.3.3, eqs. (10)–(13) omit them. Information — J.1: SW: TR 1.600000000e0 , DIN 1.599209604e0 , difference -7.903955761e-4; TR 7.3.1(c), 7.3.2(c), eqs. (55), (58) apply face-width factors; DIN 6.4.2.3, 6.4.3.3, eqs. (10)–(13) omit them. Information — J.1: SH: TR 1.200000000e0 , DIN 1.201475427e0 , difference +1.475426736e-3; TR 7.3.1(c), 7.3.2(c), eqs. (55), (58) apply face-width factors; DIN 6.4.2.3, 6.4.3.3, eqs. (10)–(13) omit them. Information — J.1: delta: TR 1.300000000e-2 mm, DIN 3.039791572e-2 mm, difference +1.739791572e-2; TR 11.2.3

eqs. (145), (146) use d_m1; DIN 10.3.3 eqs. (101), (102) use 1.1*d_f1 (d_f1 defaults to d_m1 - 2.4*m_x). Information — J.1: Sdelta: TR 6.200000000e0 , DIN 2.631759386e0 , difference -3.568240614e0; TR 11.2.3 eqs. (145), (146) use d_m1; DIN 10.3.3 eqs. (101), (102) use 1.1*d_f1 (d_f1 defaults to d_m1 - 2.4*m_x). Information — J.1: YF: TR 1.200000000e0 , DIN 1.200965719e0 , difference +9.657187608e-4; the changed wear propagates through TR 12.2 eqs. (150)–(153) / DIN 11.3.3 eqs. (106)–(109); the root method itself is shared. Information — J.1: sf2: TR 9.652000000e0 mm, DIN 9.658893521e0 mm, difference +6.893521099e-3; the changed wear propagates through TR 12.2 eqs. (150)–(153) / DIN 11.3.3 eqs. (106)–(109); the root method itself is shared. Information — J.1: tauF: TR 3.674000000e1 MPa, DIN 3.671292338e1 MPa, difference -2.707662302e-2; the changed wear propagates through TR 12.2 eqs. (150)–(153) / DIN 11.3.3 eqs. (106)–(109); the root method itself is shared. Information — J.1: SF: TR 2.450000000e0 , DIN 2.451452832e0 , difference +1.452832450e-3; the changed wear propagates through TR 12.2 eqs. (150)–(153) / DIN 11.3.3 eqs. (106)–(109); the root method itself is shared. Information — J.2: p*m: TR 1.138000000e0 , DIN 9.617591686e-1 , difference -1.762408314e-1; TR 7.3.1(c), 7.3.2(c), eqs. (55), (58) apply face-width factors; DIN 6.4.2.3, 6.4.3.3, eqs. (10)–(13) omit them. Information — J.2: h*: TR 6.661000000e-2 , DIN 7.629776223e-2 , difference +9.687762231e-3; TR 7.3.1(c), 7.3.2(c), eqs. (55), (58) apply face-width factors; DIN 6.4.2.3, 6.4.3.3, eqs. (10)–(13) omit them. Information — J.2: sigmaHm: TR 5.509400000e2 MPa, DIN 5.064981265e2 MPa, difference -4.444187355e1; TR 7.3.1(c), 7.3.2(c), eqs. (55), (58) apply face-width factors; DIN 6.4.2.3, 6.4.3.3, eqs. (10)–(13) omit them. Information — J.2: muzm: TR 5.500000000e-2 , DIN 5.155592746e-2 , difference -3.444072535e-3; TR 7.3.1(c), 7.3.2(c), eqs. (55), (58) apply face-width factors; DIN 6.4.2.3, 6.4.3.3, eqs. (10)–(13) omit them. Information — J.2: YG: TR 1.025100000e0 , DIN 9.578403314e-1 , difference -6.725966864e-2; TR 7.3.1(c), 7.3.2(c), eqs. (55), (58) apply face-width factors; DIN 6.4.2.3, 6.4.3.3, eqs. (10)–(13) omit them. Information — J.2: etaMesh: TR 6.089000000e1 %, DIN 6.249740392e1 %, difference +1.607403920e0; TR 7.3.1(c), 7.3.2(c), eqs. (55), (58) apply face-width factors; DIN 6.4.2.3, 6.4.3.3, eqs. (10)–(13) omit them. Information — J.2: PV: TR 8.710500000e1 W, DIN 8.220395829e1 W, difference -4.901041709e0; face-width friction and bearing equation choice; TR 7.3.2, 8.2.2 / DIN 6.4.3.3, 7.3.5. Information — J.2: PVmesh: TR 7.227000000e1 W, DIN 6.750747702e1 W, difference -4.762522979e0; TR 7.3.1(c), 7.3.2(c), eqs. (55), (58) apply face-width factors; DIN 6.4.2.3, 6.4.3.3, eqs. (10)–(13) omit them. Information — J.2: PVbearing: TR 8.760000000e0 W, DIN 8.625335466e0 W, difference -1.346645342e-1; TR 8.2.2 eq. (82), like DIN 7.3.5 eq. (38), uses the rounded coefficient 0.03; the engine uses DIN eq. (40), coefficient 0.028 at the reference gear. Information — J.2: etaOverall: TR 5.750000000e1 %, DIN 5.890083261e1 %, difference +1.400832613e0; TR 7.3.1(c), 7.3.2(c), eqs. (55), (58) apply face-width factors; DIN 6.4.2.3, 6.4.3.3, eqs. (10)–(13) omit them. Information — J.2: thetaM: TR 5.400000000e1 C, DIN 5.332548229e1 C, difference -6.745177123e-1; face-width friction changes the mesh heat: TR 14.1(c) eqs. (172)–(175) / DIN 13.2.3 eqs. (128)–(131); the TR's rounded 72.27 W, 4190 W/(m2*K), 0.00172 m2 and 44 C give 54.028 C. Information — J.2: nuM: TR 2.563000000e2 mm2/s, DIN 2.630071221e2 mm2/s, difference +6.707122063e0; face-width friction changes the mesh heat: TR 14.1(c) eqs. (172)–(175) / DIN 13.2.3 eqs. (128)–(131); the TR's rounded 72.27 W, 4190 W/(m2*K), 0.00172 m2 and 44 C give 54.028 C. Information — J.2: eta0M: TR 2.540000000e-1 Pa s, DIN 2.605774617e-1 Pa s, difference +6.577461697e-3; face-width friction changes the mesh heat: TR 14.1(c) eqs. (172)–(175) / DIN 13.2.3 eqs. (128)–(131); the TR's rounded 72.27 W, 4190 W/(m2*K), 0.00172 m2 and 44 C give 54.028 C. Information — J.2: WS: TR 1.620000000e0 , DIN 1.601113439e0 , difference -1.888656118e-2; face-width friction changes the mesh heat: TR 14.1(c) eqs. (172)–(175) / DIN 13.2.3 eqs. (128)–(131); the TR's rounded 72.27 W, 4190 W/(m2*K), 0.00172 m2 and 44 C give 54.028 C. Information — J.2: WH: TR 4.020000000e-1 , DIN 5.872946132e-1 , difference +1.852946132e-1; TR 7.3.1(c), 7.3.2(c), eqs. (55), (58) apply face-width factors; DIN 6.4.2.3, 6.4.3.3, eqs. (10)–(13) omit them. Information — J.2: hmin: TR 7.500000000e-2 um, DIN 8.725236812e-2 um, difference +1.225236812e-2; TR 7.3.1(c), 7.3.2(c), eqs. (55), (58) apply face-width factors; DIN 6.4.2.3, 6.4.3.3, eqs. (10)–(13) omit them. Information — J.2: KW: TR 4.900000000e-2 , DIN 8.204560903e-2 , difference +3.304560903e-2; TR 7.3.1(c), 7.3.2(c), eqs. (55), (58) apply face-width factors; DIN 6.4.2.3, 6.4.3.3, eqs. (10)–(13) omit them. Information — J.2: J0T: TR 7.189000000e-8 , DIN 2.645203920e-8 , difference -4.543796080e-8; TR 7.3.1(c), 7.3.2(c), eqs. (55), (58) apply face-width factors; DIN 6.4.2.3, 6.4.3.3, eqs. (10)–(13) omit them. Information — J.2: JW: TR 1.258000000e-7 , DIN 4.629106860e-8 , difference -7.950893140e-8; TR 7.3.1(c), 7.3.2(c), eqs. (55), (58) apply face-width factors; DIN 6.4.2.3, 6.4.3.3, eqs. (10)–(13) omit them. Information — J.2: deltaWn: TR 1.910000000e0 mm, DIN 6.455865724e-1 mm, difference -1.264413428e0; TR 7.3.1(c),

7.3.2(c), eqs. (55), (58) apply face-width factors; DIN 6.4.2.3, 6.4.3.3, eqs. (10)–(13) omit them. Information — J.2: SW: TR 1.100000000e0 , DIN 4.152009401e-1 , difference -6.847990599e-1; J.2 stops at the pointed tip; TR 9.3(a,b) / DIN 8.4(a,b) also require the worn-root limit, which governs this 17 mm face. Information — J.2: sWm: TR 1.516900000e4 m, DIN 1.394624475e4 m, difference -1.222755250e3; TR 7.3.1(c), 7.3.2(c), eqs. (55), (58) apply face-width factors; DIN 6.4.2.3, 6.4.3.3, eqs. (10)–(13) omit them. Information — J.2: lifeTip: TR 3.854000000e3 h, DIN 1.139257998e4 h, difference +7.538579981e3; TR 7.3.1(c), 7.3.2(c), eqs. (55), (58) apply face-width factors; DIN 6.4.2.3, 6.4.3.3, eqs. (10)–(13) omit them. Information — J.2: lifeGoverning: TR 3.854000000e3 h, DIN 1.454713112e3 h, difference -2.399286888e3; J.2 stops at the pointed tip; TR 9.3(a,b) / DIN 8.4(a,b) also require the worn-root limit, which governs this 17 mm face. Information — J.3: PV: TR 1.750000000e1 kW, DIN 1.743423931e1 kW, difference -6.576069266e-2; TR 8.2.2 eq. (82), like DIN 7.3.5 eq. (38), uses the rounded coefficient 0.03; the engine uses DIN eq. (40), coefficient 0.028 at the reference gear. Information — J.3: PVbearing: TR 2.570000000e0 kW, DIN 2.531780171e0 kW, difference -3.821982850e-2; TR 8.2.2 eq. (82), like DIN 7.3.5 eq. (38), uses the rounded coefficient 0.03; the engine uses DIN eq. (40), coefficient 0.028 at the reference gear. Information — J.4: p*m: TR 7.609000000e-1 , DIN 7.391861252e-1 , difference -2.171387475e-2; TR 7.3.1(c), 7.3.2(c), eqs. (55), (58) apply face-width factors; DIN 6.4.2.3, 6.4.3.3, eqs. (10)–(13) omit them. Information — J.4: h*: TR 9.580000000e-2 , DIN 9.829915731e-2 , difference +2.499157314e-3; TR 7.3.1(c), 7.3.2(c), eqs. (55), (58) apply face-width factors; DIN 6.4.2.3, 6.4.3.3, eqs. (10)–(13) omit them. Information — J.4: sigmaHm: TR 3.303200000e2 MPa, DIN 3.255769062e2 MPa, difference -4.743093778e0; TR 7.3.1(c), 7.3.2(c), eqs. (55), (58) apply face-width factors; DIN 6.4.2.3, 6.4.3.3, eqs. (10)–(13) omit them. Information — J.4: muzm: TR 2.000000000e-2 , DIN 1.955819674e-2 , difference -4.418032611e-4; TR 7.3.1(c), 7.3.2(c), eqs. (55), (58) apply face-width factors; DIN 6.4.2.3, 6.4.3.3, eqs. (10)–(13) omit them. Information — J.4: YG: TR 8.550000000e-1 , DIN 8.438672308e-1 , difference -1.113276920e-2; TR 7.3.1(c), 7.3.2(c), eqs. (55), (58) apply face-width factors; DIN 6.4.2.3, 6.4.3.3, eqs. (10)–(13) omit them. Information — J.4: etaMesh: TR 9.041000000e1 %, DIN 9.051963052e1 %, difference +1.096305197e-1; TR 7.3.1(c), 7.3.2(c), eqs. (55), (58) apply face-width factors; DIN 6.4.2.3, 6.4.3.3, eqs. (10)–(13) omit them. Information — J.4: PV: TR 8.240000000e-1 kW, DIN 8.157683922e-1 kW, difference -8.231607818e-3; face-width friction and bearing equation choice; TR 7.3.2, 8.2.2 / DIN 6.4.3.3, 7.3.5. Information — J.4: PVmesh: TR 4.790000000e-1 kW, DIN 4.731342087e-1 kW, difference -5.865791310e-3; TR 7.3.1(c), 7.3.2(c), eqs. (55), (58) apply face-width factors; DIN 6.4.2.3, 6.4.3.3, eqs. (10)–(13) omit them. Information — J.4: PVbearing: TR 1.320000000e-1 kW, DIN 1.300600376e-1 kW, difference -1.939962385e-3; TR 8.2.2 eq. (82), like DIN 7.3.5 eq. (38), uses the rounded coefficient 0.03; the engine uses DIN eq. (40), coefficient 0.028 at the reference gear. Information — J.4: etaOverall: TR 8.520000000e1 %, DIN 8.529225782e1 %, difference +9.225781677e-2; TR 7.3.1(c), 7.3.2(c), eqs. (55), (58) apply face-width factors; DIN 6.4.2.3, 6.4.3.3, eqs. (10)–(13) omit them. Information — J.4: thetaM: TR 7.770000000e1 C, DIN 7.762209706e1 C, difference -7.790294030e-2; TR 7.3.1(c), 7.3.2(c), eqs. (55), (58) apply face-width factors; DIN 6.4.2.3, 6.4.3.3, eqs. (10)–(13) omit them. Information — J.4: nuM: TR 6.424000000e1 mm2/s, DIN 6.432963609e1 mm2/s, difference +8.963608567e-2; TR 7.3.1(c), 7.3.2(c), eqs. (55), (58) apply face-width factors; DIN 6.4.2.3, 6.4.3.3, eqs. (10)–(13) omit them. Information — J.4: WS: TR 2.640000000e0 , DIN 2.637571664e0 , difference -2.428335514e-3; TR 7.3.1(c), 7.3.2(c), eqs. (55), (58) apply face-width factors; DIN 6.4.2.3, 6.4.3.3, eqs. (10)–(13) omit them. Information — J.4: hmin: TR 3.370000000e-1 um, DIN 3.462399733e-1 um, difference +9.239973330e-3; TR 7.3.1(c), 7.3.2(c), eqs. (55), (58) apply face-width factors; DIN 6.4.2.3, 6.4.3.3, eqs. (10)–(13) omit them. Information — J.4: sWm: TR 1.018574000e6 m, DIN 1.003949396e6 m, difference -1.462460393e4; TR 7.3.1(c), 7.3.2(c), eqs. (55), (58) apply face-width factors; DIN 6.4.2.3, 6.4.3.3, eqs. (10)–(13) omit them. Information — J.4: KW: TR 8.900000000e-1 , DIN 9.132327428e-1 , difference +2.323274277e-2; TR 7.3.1(c), 7.3.2(c), eqs. (55), (58) apply face-width factors; DIN 6.4.2.3, 6.4.3.3, eqs. (10)–(13) omit them. Information — J.4: J0T: TR 2.789000000e-10 , DIN 2.652126576e-10 , difference -1.368734244e-11; TR 7.3.1(c), 7.3.2(c), eqs. (55), (58) apply face-width factors; DIN 6.4.2.3, 6.4.3.3, eqs. (10)–(13) omit them. Information — J.4: JW: TR 4.880000000e-10 , DIN 4.641221507e-10 , difference -2.387784926e-11; TR 7.3.1(c), 7.3.2(c), eqs. (55), (58) apply face-width factors; DIN 6.4.2.3, 6.4.3.3, eqs. (10)–(13) omit them. Information — J.4: deltaWn: TR 4.970000000e-1 mm, DIN 4.659551529e-1 mm, difference -3.104484707e-2; TR 7.3.1(c), 7.3.2(c), eqs. (55), (58) apply face-width factors; DIN 6.4.2.3, 6.4.3.3, eqs. (10)–(13) omit them. Information — J.4: SW: TR 2.370000000e0 , DIN 2.527957036e0 , difference +1.579570363e-1; TR 7.3.1(c), 7.3.2(c), eqs. (55), (58) apply face-width factors; DIN 6.4.2.3, 6.4.3.3, eqs. (10)–(13)

omit them. Information — J.4: SH: TR 1.290000000e0 , DIN 1.311731031e0 , difference +2.173103113e-2; TR 7.3.1(c), 7.3.2(c), eqs. (55), (58) apply face-width factors; DIN 6.4.2.3, 6.4.3.3, eqs. (10)–(13) omit them. Information — J.4: delta: TR 7.500000000e-3 mm, DIN 1.479886764e-2 mm, difference +7.298867637e-3; TR 11.2.3 eqs. (145), (146) use d_m1; DIN 10.3.3 eqs. (101), (102) use 1.1*d_f1 (d_f1 defaults to d_m1 - 2.4*m_x). Information — J.4: Sdelta: TR 1.070000000e1 , DIN 5.405819010e0 , difference -5.294180990e0; TR 11.2.3 eqs. (145), (146) use d_m1; DIN 10.3.3 eqs. (101), (102) use 1.1*d_f1 (d_f1 defaults to d_m1 - 2.4*m_x). Information — J.4: YF: TR 1.170000000e0 , DIN 1.168152731e0 , difference -1.847269296e-3; the changed wear propagates through TR 12.2 eqs. (150)–(153) / DIN 11.3.3 eqs. (106)–(109); the root method itself is shared. Information — J.4: sf2: TR 9.897000000e0 mm, DIN 9.930208350e0 mm, difference +3.320834956e-2; the changed wear propagates through TR 12.2 eqs. (150)–(153) / DIN 11.3.3 eqs. (106)–(109); the root method itself is shared. Information — J.4: tauF: TR 3.678000000e1 MPa, DIN 3.665741202e1 MPa, difference -1.225879822e-1; the changed wear propagates through TR 12.2 eqs. (150)–(153) / DIN 11.3.3 eqs. (106)–(109); the root method itself is shared. Information — J.4: SF: TR 2.450000000e0 , DIN 2.455165137e0 , difference +5.165137032e-3; the changed wear propagates through TR 12.2 eqs. (150)–(153) / DIN 11.3.3 eqs. (106)–(109); the root method itself is shared. Information — J.5: p*m: TR 9.743000000e-1 , DIN 9.465265037e-1 , difference -2.777349626e-2; TR 7.3.1(c), 7.3.2(c), eqs. (55), (58) apply face-width factors; DIN 6.4.2.3, 6.4.3.3, eqs. (10)–(13) omit them. Information — J.5: h*: TR 7.377000000e-2 , DIN 7.569289600e-2 , difference +1.922895995e-3; TR 7.3.1(c), 7.3.2(c), eqs. (55), (58) apply face-width factors; DIN 6.4.2.3, 6.4.3.3, eqs. (10)–(13) omit them. Information — J.5: sigmaHm: TR 3.737900000e2 MPa, DIN 3.684198983e2 MPa, difference -5.370101682e0; TR 7.3.1(c), 7.3.2(c), eqs. (55), (58) apply face-width factors; DIN 6.4.2.3, 6.4.3.3, eqs. (10)–(13) omit them. Information — J.5: muzm: TR 2.300000000e-2 , DIN 2.228825906e-2 , difference -7.117409364e-4; TR 7.3.1(c), 7.3.2(c), eqs. (55), (58) apply face-width factors; DIN 6.4.2.3, 6.4.3.3, eqs. (10)–(13) omit them. Information — J.5: YG: TR 9.740000000e-1 , DIN 9.616597944e-1 , difference -1.234020557e-2; TR 7.3.1(c), 7.3.2(c), eqs. (55), (58) apply face-width factors; DIN 6.4.2.3, 6.4.3.3, eqs. (10)–(13) omit them. Information — J.5: etaMesh: TR 8.921000000e1 %, DIN 8.933231835e1 %, difference +1.223183523e-1; TR 7.3.1(c), 7.3.2(c), eqs. (55), (58) apply face-width factors; DIN 6.4.2.3, 6.4.3.3, eqs. (10)–(13) omit them. Information — J.5: PV: TR 8.910000000e-1 kW, DIN 8.820992173e-1 kW, difference -8.900782711e-3; face-width friction and bearing equation choice; TR 7.3.2, 8.2.2 / DIN 6.4.3.3, 7.3.5. Information — J.5: PVmesh: TR 5.460000000e-1 kW, DIN 5.394650338e-1 kW, difference -6.534966204e-3; TR 7.3.1(c), 7.3.2(c), eqs. (55), (58) apply face-width factors; DIN 6.4.2.3, 6.4.3.3, eqs. (10)–(13) omit them. Information — J.5: PVbearing: TR 1.320000000e-1 kW, DIN 1.300600376e-1 kW, difference -1.939962385e-3; TR 8.2.2 eq. (82), like DIN 7.3.5 eq. (38), uses the rounded coefficient 0.03; the engine uses DIN eq. (40), coefficient 0.028 at the reference gear. Information — J.5: etaOverall: TR 8.410000000e1 %, DIN 8.428430261e1 %, difference +1.843026123e-1; TR 7.3.1(c), 7.3.2(c), eqs. (55), (58) apply face-width factors; DIN 6.4.2.3, 6.4.3.3, eqs. (10)–(13) omit them. Information — J.5: thetaM: TR 7.830000000e1 C, DIN 7.819150525e1 C, difference -1.084947467e-1; TR 7.3.1(c), 7.3.2(c), eqs. (55), (58) apply face-width factors; DIN 6.4.2.3, 6.4.3.3, eqs. (10)–(13) omit them. Information — J.5: nuM: TR 6.324000000e1 mm2/s, DIN 6.334260974e1 mm2/s, difference +1.026097388e-1; TR 7.3.1(c), 7.3.2(c), eqs. (55), (58) apply face-width factors; DIN 6.4.2.3, 6.4.3.3, eqs. (10)–(13) omit them. Information — J.5: WS: TR 2.650000000e0 , DIN 2.652272450e0 , difference +2.272449758e-3; TR 7.3.1(c), 7.3.2(c), eqs. (55), (58) apply face-width factors; DIN 6.4.2.3, 6.4.3.3, eqs. (10)–(13) omit them. Information — J.5: hmin: TR 2.570000000e-1 um, DIN 2.636664071e-1 um, difference +6.666407099e-3; TR 7.3.1(c), 7.3.2(c), eqs. (55), (58) apply face-width factors; DIN 6.4.2.3, 6.4.3.3, eqs. (10)–(13) omit them. Information — J.5: sWm: TR 9.637960000e5 m, DIN 9.499576948e5 m, difference -1.383830518e4; TR 7.3.1(c), 7.3.2(c), eqs. (55), (58) apply face-width factors; DIN 6.4.2.3, 6.4.3.3, eqs. (10)–(13) omit them. Information — J.5: KW: TR 6.810000000e-1 , DIN 6.993151475e-1 , difference +1.831514747e-2; TR 7.3.1(c), 7.3.2(c), eqs. (55), (58) apply face-width factors; DIN 6.4.2.3, 6.4.3.3, eqs. (10)–(13) omit them. Information — J.5: J0T: TR 4.643000000e-8 , DIN 4.415496619e-10 , difference -4.598845034e-8; J.5 prints 10^-9; TR 9.2.2 eq. (116), like DIN 8.3.2 eq. (72), gives 10^-11 here; its 0.783 mm wear also requires 10^-11. Face-width factors cause the remaining difference. Information — J.5: JW: TR 8.126000000e-8 , DIN 7.727119083e-10 , difference -8.048728809e-8; J.5 prints 10^-9; TR 9.2.2 eqs. (110), (116) / DIN 8.3.2 eqs. (66), (72) and its own wear require 10^-11; then face-width factors differ. Information — J.5: deltaWn: TR 7.830000000e-1 mm, DIN 7.340436231e-1 mm, difference -4.895637687e-2; TR 7.3.1(c), 7.3.2(c), eqs. (55), (58) apply face-width factors; DIN 6.4.2.3, 6.4.3.3, eqs. (10)–(13) omit them. Information —

J.5: SW: TR 1.500000000e0 , DIN 1.604692923e0 , difference +1.046929234e-1; TR 7.3.1(c), 7.3.2(c), eqs. (55), (58) apply face-width factors; DIN 6.4.2.3, 6.4.3.3, eqs. (10)–(13) omit them. Information — J.5: SH: TR 1.140000000e0 , DIN 1.159191816e0 , difference +1.919181581e-2; TR 7.3.1(c), 7.3.2(c), eqs. (55), (58) apply face-width factors; DIN 6.4.2.3, 6.4.3.3, eqs. (10)–(13) omit them. Information — J.5: delta: TR 7.500000000e-3 mm, DIN 1.484857859e-2 mm, difference +7.348578595e-3; TR 11.2.3 eqs. (145), (146) use d_m1; DIN 10.3.3 eqs. (101), (102) use 1.1*d_f1 (d_f1 defaults to d_m1 - 2.4*m_x). Information — J.5: Sdelta: TR 1.070000000e1 , DIN 5.387721086e0 , difference -5.312278914e0; TR 11.2.3 eqs. (145), (146) use d_m1; DIN 10.3.3 eqs. (101), (102) use 1.1*d_f1 (d_f1 defaults to d_m1 - 2.4*m_x). Information — J.5: YF: TR 1.210000000e0 , DIN 1.203231327e0 , difference -6.768672685e-3; the changed wear propagates through TR 12.2 eqs. (150)–(153) / DIN 11.3.3 eqs. (106)–(109); the root method itself is shared. Information — J.5: sf2: TR 9.588000000e0 mm, DIN 9.640706435e0 mm, difference +5.270643497e-2; the changed wear propagates through TR 12.2 eqs. (150)–(153) / DIN 11.3.3 eqs. (106)–(109); the root method itself is shared. Information — J.5: tauF: TR 3.797000000e1 MPa, DIN 3.775820178e1 MPa, difference -2.117982180e-1; the changed wear propagates through TR 12.2 eqs. (150)–(153) / DIN 11.3.3 eqs. (106)–(109); the root method itself is shared. Information — J.5: SF: TR 2.370000000e0 , DIN 2.383588088e0 , difference +1.358808821e-2; the changed wear propagates through TR 12.2 eqs. (150)–(153) / DIN 11.3.3 eqs. (106)–(109); the root method itself is shared.

Closest to its tolerance: J.3’s s* is 17.3525 exactly, which Annex J prints rounded up to 17.353: it sits on the edge of half the printed digit by construction (the test allows that half digit and no more). J.1’s vg (2.89650 against 2.896) and μ0T (0.02449 against 0.024) round to the printed values too, near the edge of their last digit.

Case	Reference	Axomek	Deviation	Tolerance	
J.1: T2	587.28 N m	587.2817 N m	3.0×10^{-6}	8.5×10^{-6}	**pass**
J.1: Ftm2	7161.97 N	7161.972 N	3.4×10^{-7}	7.0×10^{-7}	**pass**
J.1: gamma	12.53 deg	12.52881 deg	0.0095 %	0.04 %	**pass**
J.1: vg	2.896 m/s	2.896495 m/s	0.017 %	0.017 %	**pass**
J.1: s*	30.285	30.285	0 %	0.0017 %	**pass**
J.1: mu0T	0.024	0.02448994	2 %	2.1 %	**pass**
J.1: thetaS	73.2 C	73.22558 C	0.035 %	0.068 %	**pass**
J.1: eta0M	0.06 Pa s	0.06335003 Pa s	5.6 %	8.3 %	**pass**
J.1: WH	1	1	0 %	10^{-12}	**pass**
J.1: deltaWlim	1.17 mm	1.171425 mm	0.12 %	0.43 %	**pass**
J.1: sigmaHG	442.77 MPa	442.7659 MPa	9.4×10^{-6}	0.0011 %	**pass**
J.1: Zv	0.85	0.8514728	0.17 %	0.59 %	**pass**
J.1: Ygamma	1.024	1.024394	0.038 %	0.049 %	**pass**
J.1: ST	1.37	1.365643	0.32 %	0.36 %	**pass**
J.1: PVidle	0.153 kW	0.1528194 kW	0.12 %	0.33 %	**pass**
J.1: PVseal	0.046 kW	0.04580064 kW	0.43 %	1.1 %	**pass**
J.1: YW	0.95	0.95	0 %	1.1×10^{-12}	**pass**
J.1: YR	1	1	0 %	10^{-12}	**pass**
J.1: YS	1	1	0 %	10^{-12}	**pass**
J.1: Zh	1	1	0 %	10^{-12}	**pass**
J.1: Zs	1	1	0 %	10^{-12}	**pass**
J.1: Zoil	1	1	0 %	10^{-12}	**pass**

Case	Reference	Axomek	Deviation	Tolerance	
J.1: deltaLim	0.08 mm	0.08 mm	0 %	1.3×10^{-11}	**pass**
J.1: YK	1	1	0 %	10^{-12}	**pass**
J.1: tauFG	90 MPa	90 MPa	0 %	1.1×10^{-14}	**pass**
J.1: WNS	1	1	0 %	10^{-12}	**pass**
J.1: WML	1.75	1.75	0 %	5.7×10^{-13}	**pass**
J.1: rhoM	0.97 kg/dm3	0.973384 kg/dm3	0.35 %	0.52 %	**pass**
J.1: alphaL	24440 W/(m2 K)	24440 W/(m2 K)	1.2×10^{-15}	4.1×10^{-11}	**pass**
J.1: AR	0.0049 m2	0.00492 m2	0.41 %	1 %	**pass**
J.1: Yepsilon	0.5	0.5	0 %	2.0×10^{-12}	**pass**
J.2: P2	117.8 W	117.8097 W	0.0083 %	0.042 %	**pass**
J.2: Ftm2	5925.93 N	5925.926 N	6.9×10^{-7}	8.4×10^{-7}	**pass**
J.2: gamma	4.97 deg	4.969741 deg	0.0052 %	0.1 %	**pass**
J.2: vg	0.23 m/s	0.2266611 m/s	1.5 %	2.2 %	**pass**
J.2: s*	73.58	73.58	1.9×10^{-16}	6.8×10^{-6}	**pass**
J.2: mu0T	0.046	0.04567921	0.7 %	1.1 %	**pass**
J.2: YS	1.24	1.240347	0.028 %	0.4 %	**pass**
J.2: thetaS	44 C	43.96509 C	0.079 %	0.11 %	**pass**
J.2: deltaWlimTip	2.1 mm	2.099218 mm	0.037 %	2.4 %	**pass**
J.2: NL	867000	867150	0.017 %	0.058 %	**pass**
J.2: PVidle	4.6 W	4.61061 W	0.23 %	1.1 %	**pass**
J.2: PVseal	1.46 W	1.460536 W	0.037 %	0.34 %	**pass**
J.2: YW	0.95	0.95	0 %	1.1×10^{-12}	**pass**
J.2: YR	1	1	0 %	10^{-12}	**pass**
J.2: rhoM	0.99 kg/dm3	0.990762 kg/dm3	0.077 %	0.51 %	**pass**
J.2: alphaL	4190 W/(m2 K)	4190 W/(m2 K)	0 %	2.4×10^{-10}	**pass**
J.2: AR	0.00172 m2	0.00172125 m2	0.073 %	0.29 %	**pass**
J.2: WNS	1	1	0 %	10^{-12}	**pass**
J.2: WML	1.75	1.75	0 %	5.7×10^{-13}	**pass**
J.3: P2	333.4 kW	333.3935 kW	0.0019 %	0.015 %	**pass**
J.3: Ftm2	39097.74 N	39097.74 N	1.1×10^{-7}	1.3×10^{-7}	**pass**
J.3: gamma	21.8 deg	21.80141 deg	0.0065 %	0.23 %	**pass**
J.3: vg	22.8 m/s	22.84 m/s	0.18 %	0.22 %	**pass**
J.3: p*m	1.0259	1.025843	0.0055 %	0.0097 %	**pass**
J.3: h*	0.05912	0.05911686	0.0053 %	0.0085 %	**pass**
J.3: s*	17.353	17.3525	0.0029 %	0.0029 %	**pass**
J.3: sigmaHm	225.57 MPa	225.5674 MPa	0.0012 %	0.0022 %	**pass**
J.3: mu0T	0.021	0.02095372	0.22 %	2.4 %	**pass**
J.3: muzm	0.014	0.01369961	2.1 %	3.6 %	**pass**

Case	Reference	Axomek	Deviation	Tolerance	
J.3: YS	0.632	0.6324555	0.072 %	0.079 %	**pass**
J.3: YG	1.09	1.088161	0.17 %	0.46 %	**pass**
J.3: etaMesh	96.16 %	96.15868 %	0.0014 %	0.0052 %	**pass**
J.3: PVmesh	12.72 kW	12.71807 kW	0.015 %	0.039 %	**pass**
J.3: etaOverall	95 %	95.03054 %	0.032 %	0.053 %	**pass**
J.3: sigmaHG	225.57 MPa	225.5675 MPa	0.0011 %	0.0022 %	**pass**
J.3: SH	1	1.000001	5.5×10^{-7}	5 %	**pass**
J.3: Zh	1.149	1.148538	0.04 %	0.044 %	**pass**
J.3: Zv	0.43	0.4316122	0.37 %	1.2 %	**pass**
J.3: Zs	0.95	0.9534626	0.36 %	0.53 %	**pass**
J.3: Zu	0.918	0.9177626	0.026 %	0.054 %	**pass**
J.3: sigmaHlim	520 MPa	520 MPa	0 %	1.9×10^{-13}	**pass**
J.3: Zoil	1	1	0 %	10^{-12}	**pass**
J.3: lifePitting	10891 h	10891.04 h	3.3×10^{-6}	0.0046 %	**pass**
J.3: PVidle	1.54 kW	1.540323 kW	0.021 %	0.32 %	**pass**
J.3: PVseal	0.644 kW	0.6440715 kW	0.011 %	0.078 %	**pass**
J.3: YW	0.95	0.95	0 %	1.1×10^{-12}	**pass**
J.3: YR	1	1	0 %	10^{-12}	**pass**
J.4: P2	4.73 kW	4.730755 kW	0.016 %	0.11 %	**pass**
J.4: Ftm2	7392.75 N	7392.749 N	10^{-7}	6.8×10^{-7}	**pass**
J.4: gamma	11.01 deg	11.00951 deg	0.0045 %	0.045 %	**pass**
J.4: vg	3.29 m/s	3.290212 m/s	0.0065 %	0.15 %	**pass**
J.4: s*	40.253	40.253	0 %	0.0012 %	**pass**
J.4: mu0T	0.024	0.0243967	1.7 %	2.1 %	**pass**
J.4: thetaS	73.6 C	73.56054 C	0.054 %	0.068 %	**pass**
J.4: eta0M	0.06 Pa s	0.06259782 Pa s	4.3 %	8.3 %	**pass**
J.4: WH	1	1	0 %	10^{-12}	**pass**
J.4: deltaWlim	1.178 mm	1.177915 mm	0.0072 %	0.042 %	**pass**
J.4: sigmaHG	427.07 MPa	427.0693 MPa	1.6×10^{-6}	0.0012 %	**pass**
J.4: Zv	0.83	0.8281613	0.22 %	0.6 %	**pass**
J.4: Zu	0.992	0.9916996	0.03 %	0.05 %	**pass**
J.4: Ygamma	1.019	1.01875	0.025 %	0.049 %	**pass**
J.4: ST	1.36	1.359425	0.042 %	0.37 %	**pass**
J.4: PVidle	0.153 kW	0.1528194 kW	0.12 %	0.33 %	**pass**
J.4: PVseal	0.06 kW	0.05975479 kW	0.41 %	0.83 %	**pass**
J.4: YW	0.95	0.95	0 %	1.1×10^{-12}	**pass**
J.4: YR	1	1	0 %	10^{-12}	**pass**
J.4: YS	1	1	0 %	10^{-12}	**pass**
J.4: Zh	1	1	0 %	10^{-12}	**pass**

Case	Reference	Axomek	Deviation	Tolerance	
J.4: Zs	1	1	0 %	10^{-12}	pass
J.4: Zoil	1	1	0 %	10^{-12}	pass
J.4: deltaLim	0.08 mm	0.08 mm	0 %	1.3×10^{-11}	pass
J.4: YK	1	1	0 %	10^{-12}	pass
J.4: tauFG	90 MPa	90 MPa	0 %	1.1×10^{-14}	pass
J.4: WNS	1	1	0 %	10^{-12}	pass
J.4: WML	1.75	1.75	0 %	5.7×10^{-13}	pass
J.4: rhoM	0.97 kg/dm3	0.9730791 kg/dm3	0.32 %	0.52 %	pass
J.4: alphaL	24440 W/(m2 K)	24440 W/(m2 K)	10^{-15}	4.1×10^{-11}	pass
J.4: AR	0.0048 m2	0.0047664 m2	0.7 %	1 %	pass
J.4: Yepsilon	0.5	0.5	2.2×10^{-16}	2.0×10^{-12}	pass
J.5: P2	4.73 kW	4.730755 kW	0.016 %	0.11 %	pass
J.5: Ftm2	7392.75 N	7392.749 N	10^{-7}	6.8×10^{-7}	pass
J.5: gamma	11.01 deg	11.00951 deg	0.0045 %	0.045 %	pass
J.5: vg	3.29 m/s	3.290212 m/s	0.0065 %	0.15 %	pass
J.5: s*	33.659	33.659	0 %	0.0015 %	pass
J.5: mu0T	0.024	0.0243967	1.7 %	2.1 %	pass
J.5: thetaS	73.6 C	73.56054 C	0.054 %	0.068 %	pass
J.5: eta0M	0.06 Pa s	0.0616116 Pa s	2.7 %	8.3 %	pass
J.5: WH	1	1	0 %	10^{-12}	pass
J.5: deltaWlim	1.178 mm	1.177915 mm	0.0072 %	0.042 %	pass
J.5: sigmaHG	427.07 MPa	427.0693 MPa	1.6×10^{-6}	0.0012 %	pass
J.5: Zv	0.83	0.8281613	0.22 %	0.6 %	pass
J.5: Zu	0.992	0.9916996	0.03 %	0.05 %	pass
J.5: Ygamma	1.019	1.01875	0.025 %	0.049 %	pass
J.5: ST	1.36	1.359425	0.042 %	0.37 %	pass
J.5: PVidle	0.153 kW	0.1528194 kW	0.12 %	0.33 %	pass
J.5: PVseal	0.06 kW	0.05975479 kW	0.41 %	0.83 %	pass
J.5: YW	0.95	0.95	0 %	1.1×10^{-12}	pass
J.5: YR	1	1	0 %	10^{-12}	pass
J.5: YS	1	1	0 %	10^{-12}	pass
J.5: Zh	1	1	0 %	10^{-12}	pass
J.5: Zs	1	1	0 %	10^{-12}	pass
J.5: Zoil	1	1	0 %	10^{-12}	pass
J.5: deltaLim	0.08 mm	0.08 mm	0 %	1.3×10^{-11}	pass
J.5: YK	1	1	0 %	10^{-12}	pass
J.5: tauFG	90 MPa	90 MPa	0 %	1.1×10^{-14}	pass
J.5: WNS	1	1	0 %	10^{-12}	pass

Case	Reference	Axomek	Deviation	Tolerance	
J.5: WML	1.75	1.75	0 %	5.7×10^{-13}	**pass**
J.5: rhoM	0.97 kg/dm3	0.9726722 kg/dm3	0.28 %	0.52 %	**pass**
J.5: alphaL	24440 W/(m2 K)	24440 W/(m2 K)	10^{-15}	4.1×10^{-11}	**pass**
J.5: AR	0.0048 m2	0.0047664 m2	0.7 %	1 %	**pass**
J.5: Yepsilon	0.5	0.5	2.2×10^{-16}	2.0×10^{-12}	**pass**

Worm gears: VDI 2736-3 Annex A

Kind of reference	Worked examples of standards and books
Implements	VDI 2736 Part 3
Reference source	VDI 2736 Part 3, Annex A, examples A1 and A2
Cases	95 of 95 within tolerance

A1 (POM, 12 N·m, 2200 rpm) and A2 (PA 46, 0.3 N·m, 4500 rpm). Reference values are printed in Annex A; tolerances cover input rounding, rounded K_z and thicknesses in the load and area, and rounded elastic factors / ellipse axes in contact stress. A1 uses the supplied contact ratio 2.130; its diameters give 2.29471 without that override. A1's POM curve is Figure 6 (the Figure 8 citation is a misprint); A2's 44.117° labels phi as theta. A2 also has transposed final cycle digits and computes root safety using 1.299 rather than its own 1.312 MPa: the checks use 60 L_h n_1/u and 9.57/1.312. The A1 shear limit is 0.75 times 23.3, or 17.475 MPa, rather than its printed 17.46. Figure 4 is compared numerically for A2 only: PA 46 has no design load rating in that figure. Strength readings are supplied explicitly; the engine tests separately check the digitized curves.

Case	Reference	Axomek	Deviation	Tolerance	
A1: d_s1	13.826 mm	13.82583 mm	0.0012 %	0.008 %	**pass**
A1: d_s2	51.039 mm	51.03912 mm	2.3×10^{-6}	0.002 %	**pass**
A1: a	32.432 mm	32.43248 mm	0.0015 %	0.003 %	**pass**
A1: d_b1	8.802 mm	8.801965 mm	4.0×10^{-6}	0.008 %	**pass**
A1: d_b2	48.773 mm	48.7725 mm	0.001 %	0.002 %	**pass**
A1: beta_s1	75.716 °	75.71643 °	5.7×10^{-6}	0.002 %	**pass**
A1: beta_b1	68.203 °	68.20334 °	5.0×10^{-6}	0.002 %	**pass**
A1: beta_b2	13.674 °	13.67358 °	0.0031 %	0.008 %	**pass**
A1: F_t2	470.23 N	470.2276 N	5.1×10^{-6}	0.002 %	**pass**
A1: K_z	0.91	0.9106833	0.075 %	0.1 %	**pass**
A1: b_m	13.213 mm	13.21293 mm	5.7×10^{-6}	0.008 %	**pass**
A1: m_x1	1.76 mm	1.75997 mm	0.0017 %	0.05 %	**pass**
A1: C	7.073 N/mm²	7.067841 N/mm²	0.073 %	0.1 %	**pass**
A1: omega2	15.889 rad/s	15.88852 rad/s	0.0031 %	0.005 %	**pass**
A1: v_gs	1.643 m/s	1.643426 m/s	0.026 %	0.05 %	**pass**
A1: C_lim	7.798 N/mm²	7.797141 N/mm²	0.011 %	0.02 %	**pass**
A1: S_C	1.1	1.103186	0.29 %	0.5 %	**pass**
A1: b_max	15 mm	15 mm	0 %	10^{-12}	**pass**
A1: d_x0	46.232 mm	46.23295 mm	0.0021 %	0.003 %	**pass**
A1: m_t2	1.754 mm	1.753859 mm	0.008 %	0.05 %	**pass**
A1: s_t2	3.721 mm	3.720545 mm	0.012 %	0.03 %	**pass**

Case	Reference	Axomek	Deviation	Tolerance	
A1: alpha_t2	16.48 °	16.4798 °	0.0012 %	0.005 %	**pass**
A1: alpha_at2	25.114 °	25.11495 °	0.0038 %	0.005 %	**pass**
A1: s_at2	2.745 mm	2.743846 mm	0.042 %	0.05 %	**pass**
A1: A_tau	52.887	52.87808	0.017 %	0.02 %	**pass**
A1: tau_F	4.446 MPa	4.446338 MPa	0.0076 %	0.02 %	**pass**
A1: N_L	6.3724×10^{6}	6.3724×10^{6}	3.2×10^{-8}	10^{-7}	**pass**
A1: sigma_ZSch	23.3 MPa	23.3 MPa	0 %	10^{-12}	**pass**
A1: tau_Flim	17.475 MPa	17.475 MPa	0 %	10^{-12}	**pass**
A1: S_F	3.93	3.930201	0.0051 %	0.01 %	**pass**
A1: alpha_sn	16.639 °	16.63927 °	0.0016 %	0.008 %	**pass**
A1: rho_z	5.958 °	5.958408 °	0.0068 %	0.01 %	**pass**
A1: F_n	520.26 N	520.258 N	3.9×10^{-6}	0.002 %	**pass**
A1: Z_epsilonS	0.686	0.6851887	0.12 %	0.15 %	**pass**
A1: phi	49.81 °	49.81056 °	0.0011 %	0.02 %	**pass**
A1: rho_n1	14.357 mm	14.35719 mm	0.0013 %	0.008 %	**pass**
A1: rho_n2	7.739 mm	7.739911 mm	0.012 %	0.02 %	**pass**
A1: rho_n	5.028 mm	5.028866 mm	0.017 %	0.02 %	**pass**
A1: theta	46.79 °	46.79067 °	0.0014 %	0.02 %	**pass**
A1: eta	0.635	0.6353553	0.056 %	0.1 %	**pass**
A1: xi	1.911	1.910756	0.013 %	0.05 %	**pass**
A1: Z_F	0.085 MPa^(−1/3)	0.08464675 MPa^(−1/3)	0.42 %	0.5 %	**pass**
A1: a_h	2.238 mm	2.228666 mm	0.42 %	0.5 %	**pass**
A1: b_h	0.744 mm	0.7410654 mm	0.39 %	0.5 %	**pass**
A1: sigma_Hm	102.33 MPa	103.055 MPa	0.71 %	0.8 %	**pass**
A1: eta_z	0.69	0.690398	0.058 %	0.1 %	**pass**
A1: eta_ges	0.69	0.690398	0.058 %	0.1 %	**pass**
A2: d_s1	4.249 mm	4.249079 mm	0.0019 %	0.02 %	**pass**
A2: d_s2	33.121 mm	33.12106 mm	1.8×10^{-6}	0.003 %	**pass**
A2: a	18.685 mm	18.68507 mm	3.7×10^{-6}	0.005 %	**pass**
A2: d_b1	3.667 mm	3.667402 mm	0.011 %	0.02 %	**pass**
A2: d_b2	32.366 mm	32.36647 mm	0.0015 %	0.003 %	**pass**
A2: beta_s1	69.636 °	69.63584 °	2.3×10^{-6}	0.002 %	**pass**
A2: beta_b1	66.73 °	66.72952 °	7.2×10^{-6}	0.01 %	**pass**
A2: beta_b2	19.937 °	19.93713 °	6.7×10^{-6}	0.005 %	**pass**
A2: epsilon_n	2.268	2.26774	0.011 %	0.02 %	**pass**
A2: F_t2	18.12 N	18.11536 N	0.026 %	0.03 %	**pass**
A2: K_z	1.18	1.180131	0.011 %	0.5 %	**pass**
A2: b_m	3.429 mm	3.42884 mm	0.0047 %	0.03 %	**pass**
A2: m_x1	0.789 mm	0.7885967 mm	0.051 %	0.07 %	**pass**

Case	Reference	Axomek	Deviation	Tolerance	
A2: C	1.807 N/mm²	1.807028 N/mm²	0.0015 %	0.07 %	**pass**
A2: omega2	22.44 rad/s	22.43995 rad/s	2.3×10^{-6}	0.03 %	**pass**
A2: v_gs	1.068 m/s	1.06791 m/s	0.0084 %	0.05 %	**pass**
A2: Figure 4 comparison	8.975 N/mm²	8.975197 N/mm²	0.0022 %	0.02 %	**pass**
A2: b_max	4.958 mm	4.957783 mm	0.0044 %	0.03 %	**pass**
A2: d_x0	31.427 mm	31.42714 mm	4.4×10^{-6}	0.004 %	**pass**
A2: m_t2	0.802 mm	0.8016104 mm	0.049 %	0.07 %	**pass**
A2: s_t2	1.416 mm	1.415105 mm	0.063 %	0.1 %	**pass**
A2: alpha_t2	15.981 °	15.98107 °	4.2×10^{-6}	0.005 %	**pass**
A2: alpha_at2	18.31 °	18.31225 °	0.012 %	0.03 %	**pass**
A2: s_at2	1.302 mm	1.30067 mm	0.1 %	0.11 %	**pass**
A2: A_tau	6.907	6.902313	0.068 %	0.08 %	**pass**
A2: tau_F	1.312 MPa	1.312268 MPa	0.02 %	0.03 %	**pass**
A2: N_L	6.4286×10^{6}	6.4286×10^{6}	1.4×10^{-16}	10^{-12}	**pass**
A2: sigma_ZSch	12.76 MPa	12.76 MPa	0 %	10^{-12}	**pass**
A2: tau_Flim	9.57 MPa	9.57 MPa	0 %	10^{-12}	**pass**
A2: S_F corrected	7.294207	7.29272	0.02 %	0.03 %	**pass**
A2: alpha_sn	11.509 °	11.50896 °	3.8×10^{-6}	0.01 %	**pass**
A2: rho_z	5.827 °	5.826971 °	5.0×10^{-6}	0.01 %	**pass**
A2: F_n	20.5 N	20.49594 N	0.02 %	0.03 %	**pass**
A2: Z_epsilonS	0.664	0.664054	0.0081 %	0.1 %	**pass**
A2: phi	44.117 °	44.11707 °	1.6×10^{-6}	0.003 %	**pass**
A2: rho_n1	2.717 mm	2.715818 mm	0.044 %	0.05 %	**pass**
A2: rho_n2	3.74 mm	3.738911 mm	0.029 %	0.2 %	**pass**
A2: rho_n	1.574 mm	1.573141 mm	0.055 %	0.06 %	**pass**
A2: theta	43.41 °	43.41886 °	0.02 %	0.025 %	**pass**
A2: eta	0.613	0.6125845	0.068 %	0.1 %	**pass**
A2: xi	2.027	2.026446	0.027 %	0.05 %	**pass**
A2: Z_F	0.115 MPa^(−1/3)	0.1152336 MPa^(−1/3)	0.2 %	0.5 %	**pass**
A2: a_h	0.742 mm	0.743235 mm	0.17 %	0.3 %	**pass**
A2: b_h	0.224 mm	0.2246762 mm	0.3 %	0.4 %	**pass**
A2: sigma_Hm	39.1 MPa	38.91609 MPa	0.47 %	0.6 %	**pass**
A2: eta_z	0.75	0.7546419	0.62 %	0.7 %	**pass**
A2: eta_z_reverse	0.7	0.6986013	0.2 %	0.3 %	**pass**
A2: eta_ges	0.71	0.7093634	0.09 %	0.2 %	**pass**

Standards’ tables and printed values

Plastic worm wheel: friction band of the mesh

Kind of reference	Standards’ tables and printed values
Implements	VDI 2736 Part 3:2014, Table 1; VDI 2736 Part 2:2014, Table 2 (dry)
Reference source	VDI 2736 Part 3:2014, Table 1, p. 18: POM 0.040–0.060, PA 66 0.034–0.051, PA 46 0.040–0.060, PA 6 0.032–0.048, PEEK 0.050–0.075 (the whole table 0.032–0.150 for an own plastic); VDI 2736 Part 2:2014, Table 2, p. 10: 0.20 dry for plastic on steel (its 0.40 is PA on PA)
Cases	11 of 11 within tolerance

With no friction given, μ is the band's upper value (an own plastic 0.10; dry, VDI 2736-2 Table 2's single 0.20 for plastic on steel), and self-locking and back-driving are decided over the band.

Case	Reference	Axomek	Deviation	Tolerance	
POM, lower	0.04	0.04	0	exact	**pass**
POM, upper	0.06	0.06	0	exact	**pass**
PA 66, lower	0.034	0.034	0	exact	**pass**
PA 66, upper	0.051	0.051	0	exact	**pass**
PA 46, lower	0.04	0.04	0	exact	**pass**
PA 46, upper	0.06	0.06	0	exact	**pass**
PEEK, lower	0.05	0.05	0	exact	**pass**
PEEK, upper	0.075	0.075	0	exact	**pass**
own plastic, lower	0.032	0.032	0	exact	**pass**
own plastic, upper	0.15	0.15	0	exact	**pass**
dry (VDI 2736-2, plastic on steel)	0.2	0.2	0	exact	**pass**

The standards’ formulae worked by hand

Plastic worm wheel: flank safety over the wheel temperatures

Kind of reference	The standards’ formulae worked by hand
Implements	VDI 2736 Part 3:2014, 3.2.2, eqs. (22) to (28); VDI 2736 Part 2:2014, Bild 6
Reference source	hand calculation of a column EPS assist gear's PA 66 wheel (m_n 2.75 mm, 2/34 teeth, γ 10°) at 20 °C from its 70 °C rating
Cases	2 of 2 within tolerance

E_2 from the shared PA 66 table at 20 and 70 °C, σ_HlimN read from VDI 2736-2 Bild 6; the 2 % covers the 70 °C hand σ_Hm (1 %) and the curve reading

Case	Reference	Axomek	Deviation	Tolerance	
S_H at 20 °C	0.810141	0.8103601	0.027 %	2 %	**pass**
temperature of the lowest S_H	20 °C	20 °C	0 °C	0.5 °C	**pass**

Closed-form solutions

Dry plastic worm pair: back-driving efficiency and self-locking at Table 2's μ

Kind of reference	Closed-form solutions
Implements	VDI 2736 Part 3:2014, eq. (37); VDI 2736 Part 2:2014, Table 2 (dry)
Reference source	μ_lock = tan γ · cos α_n and η′_z = tan(γ − ρ_z) / tan γ, ρ_z = arctan(0.20 / cos α_n), for VDI 2736-3 Annex A's pair A1 (2 starts, m_n 1.7 mm, α_n 16°) with a PA 66 wheel of the worm's helix, run dry, at leads of 8° to 15°; VDI 2736 Part 2:2014, Table 2, p. 10: 0.20 for plastic on steel, a single value
Cases	24 of 24 within tolerance

Exact: a relative 10^{-9}. Table 2 gives one value for a dry steel worm on plastic, no range and no scatter (its other dry rows are plastic on plastic): the figures follow it, but neither self-locking nor back-driving is decided at that default; a friction given decides them.

Case	Reference	Axomek	Deviation	Tolerance	
γ 8°: μ	0.2	0.2	0	exact	**pass**
γ 8°: μ_lock	0.1350965	0.1350965	0 %	10^{-9}	**pass**
γ 8°: η′_z	0	0	0	exact	**pass**
γ 11°: μ	0.2	0.2	0	exact	**pass**
γ 11°: μ_lock	0.1868504	0.1868504	8.9×10^{-16}	10^{-9}	**pass**
γ 11°: η′_z	0	0	0	exact	**pass**
γ 11.5°: μ	0.2	0.2	0	exact	**pass**
γ 11.5°: μ_lock	0.1955709	0.1955709	2.8×10^{-16}	10^{-9}	**pass**
γ 11.5°: η′_z	0	0	0	exact	**pass**
γ 11.75°: μ	0.2	0.2	0	exact	**pass**
γ 11.75°: μ_lock	0.1999427	0.1999427	5.6×10^{-16}	10^{-9}	**pass**
γ 11.75°: η′_z	0	0	0	exact	**pass**
γ 11.76°: μ	0.2	0.2	0	exact	**pass**
γ 11.76°: μ_lock	0.2001178	0.2001178	5.5×10^{-16}	10^{-9}	**pass**
γ 11.76°: η′_z	0.0005640087	0.0005640087	9.5×10^{-13}	10^{-9}	**pass**
γ 12°: μ	0.2	0.2	0	exact	**pass**
γ 12°: μ_lock	0.2043225	0.2043225	2.7×10^{-16}	10^{-9}	**pass**
γ 12°: η′_z	0.02025924	0.02025924	1.3×10^{-14}	10^{-9}	**pass**
γ 12.5°: μ	0.2	0.2	0	exact	**pass**
γ 12.5°: μ_lock	0.2131066	0.2131066	1.3×10^{-16}	10^{-9}	**pass**
γ 12.5°: η′_z	0.05879073	0.05879073	2.0×10^{-15}	10^{-9}	**pass**
γ 15°: μ	0.2	0.2	0	exact	**pass**
γ 15°: μ_lock	0.2575693	0.2575693	2.2×10^{-16}	10^{-9}	**pass**
γ 15°: η′_z	0.2117074	0.2117074	7.9×10^{-16}	10^{-9}	**pass**

Plastic worm pair: friction from which a single-start q 20 drive is self-locking

Kind of reference	Closed-form solutions
Implements	VDI 2736 Part 3:2014, eq. (37), Table 1
Reference source	μ_lock = tan γ · cos α_n = cos 20°/20 for a single-start worm of q 20 on a PA 66 wheel of its helix (a smoke-vent flap drive of a user test)
Cases	1 of 1 within tolerance

Inside Table 1's PA 66 band 0.034–0.051, so self-locking at the default μ 0.051 is not decided.

Case	Reference	Axomek	Deviation	Tolerance	
μ_lock	0.04698463	0.04698463	8.9×10^{-16}	10^{-9}	**pass**

Plastic worm pair: friction from which it is self-locking

Kind of reference	Closed-form solutions
Implements	VDI 2736 Part 3:2014, eq. (37)
Reference source	μ_lock = tan γ · cos α_n for a wheel of the worm's helix (β_2 = γ), VDI 2736-3 Annex A's pair A1 (2 starts, m_n 1.7 mm, α_n 16°) at leads of 1.5° to 3.5°
Cases	5 of 5 within tolerance

Exact: a relative 10^{-9}. On either side of μ_lock (×(1 ± 10^{-6})) the solver's η′ flips between zero and a back-drive.

Case	Reference	Axomek	Deviation	Tolerance	
γ 1.5°	0.02517152	0.02517152	3.0×10^{-15}	10^{-9}	**pass**
γ 2°	0.033568	0.033568	5.0×10^{-15}	10^{-9}	**pass**
γ 2.5°	0.04196959	0.04196959	1.2×10^{-15}	10^{-9}	**pass**
γ 2.8°	0.04701354	0.04701354	3.2×10^{-15}	10^{-9}	**pass**
γ 3.5°	0.05879328	0.05879328	4.7×10^{-16}	10^{-9}	**pass**

Stated limits

Standards and scope **(help: modules/worm.md)**

DIN 3996:2012-09, method C: characteristic values p*m, h* and s* by the standard's equations (or given from a method B analysis), mean friction (7.4), losses and efficiency (7), wear (8), pitting (9), worm deflection (10), tooth root (11), oil sump temperature (12) and wheel bulk temperature (13). The rating reproduces the standard's three worked examples (`axomek verify`). The worm's deflection is the standard's Gl. (101) as printed (10.3.3, method C):

Its constant holds a steel worm's modulus, so the worm modulus does not change it, and 1.1 · df1 stands for the worm with its threads: it comes out 0.676 times a simply supported plain beam of df1.

Not covered: a scuffing check, and the worm shaft's strength and its bearings' life. Mount the pair on shafts and rate them in the shaft and bearings modules under its mesh forces.

Known limits

- Where a train's speeds come from its items and two of them turn one shaft opposite ways (a warning at the later one's sense, with its fix): - the pair the train took as driven the other way round is rated so meanwhile, and its mesh loss with it (the Two-stage helical gearbox with G2 alone reversed in its file: total 3 614 W against 3 610 W), though the power in and the efficiencies are not given; - a drive or pair added or mounted beside a train that turns its shaft negatively starts at its default sense, positive, and so gets the warning until it is reversed (the fix) rather than taking the train's sense; - a rack mesh's sense reversed is not carried on to the rest of its train (a gear pair's, worm pair's or drive's is), only warned of.

- A result not determined is hidden display by display, not by one shared rule: in 0.3.0 the displays reviewed (the report, Show the calculation, the CLI's text, the comparisons, the Overview, the cards, the modules' tiles and the Assistant's digest) say "not determined". Those found last are refused instead: the Changes dialog's and Compare projects' check rows and the Overview's key results of keys and bolts say "not determined (see the check)"; the assembly view gives no share of the limit for a check not determined or not computed; and in the report an element's results, and a key's or a fit's results its checks go by (safeties, capacities, permissible values, pressures, shear, the fit's Turning row; with `axomek solve`'s key and fit lines), say "not determined (see the checks)" as soon as one of its checks is not determined, determined ones with them; a key's page refuses its flank pressures and tooth shear stress the same way (its permissible values and capacity, which do not depend on its rating, stay). A bolted joint with a check not determined (beyond VDI 2230's limit G, a duty not computed) has its whole Results block refused the same way in the report's section and in `axomek solve`'s line (its tightening torque, permissible assembly preload, resiliences and embedding with it, which the bolt page still shows); its page refuses its preload needed, load factor, clamp loads, stresses, opening and the joint diagram, but the notes under them, on its page and under the report's section, still quote the eccentric model's figures (M_Sb/W_b, F_Sm/F_0.2min, F_K against F_Kab). Make it pass and `axomek fix` give such a check no value either. A module's page, once one of an item's checks is not determined, shows no tile's caption (it may quote another check's figure) and refuses its tiles and Details tied to no check (a worm pair's load value C / Clim), and the Overview refuses the item's tiles tied to no check the same way, including ones that do not depend on the rating: a worm pair's method, centre distance and ratio, a motor's power and magnetic stiffness read "not determined" too; a later release shows those again. A design search meets no requirement or objective on a result of a design with any check not determined (a requirement's own check may be cleared, and the system life and power loss take every item's), its dimensions apart, nor on a fit's yield safety resting on an estimated yield strength, even where that figure itself would stand: the safe side; a later release decides each figure on its own. A bearing whose life the engine puts outside its rating method (κ below 0.1 where it turns, balls in four-point contact, a hybrid, whose a_ISO is defined for steel only) has all its lives, basic and modified, meet no search requirement or objective, as a minimum, a maximum or an objective, its minimum life cleared or not, and so has the system life with such a member; it says "not determined" without a life check too. The safe side: a hybrid's basic life, or one at a low κ, would stand; in a later release a search requirement on a figure whose own check is cleared is evaluated as that check, each life on its own. A design with a bearing the engine marks outside what it rates (a designation that contradicts it, a slewing ring's proportions) takes no calculated result in a search, its dimensions apart but that bearing's own (inferred from its geometry as modelled), as a design with a check not determined does, whatever its own checks say. A bearing whose designation contradicts it shows only its results' warnings it can vouch for: those at its own path, of the bearing as modelled, are left out (a validation warning at that same path too, such as that its geometry is estimated), its fields' warnings stay. A check not determined is said in a short form wherever its message would be rewritten: what it checks, named from its id, "not determined" and its kind's reason in fixed words, else "(see the check)", never a part of its message, which can hold a name the user wrote and the figures after it; a name that holds an engine mark's words can still pick another kind's reason (no figures). The full engine message, with the reason in context, comes back in a later release, when every display shares one rule. The bearing module's speed chart is not drawn while the bearing's operating nθ is not determined (a rotor's pull, a designation its geometry contradicts), saying why; `axomek speed-limits` then lists the curves' samples without an

operating point. A later release draws the curves apart from that load. A display not yet reviewed may still give such a number beside its check, which says "not determined". A later release routes every display of a check's value through one rule, with a test over every template. Known in 0.3.0, closed in a later release by that one shared display rule: - the Assistant's solve data still carries a check's value and utilisation where the check is not determined (a bearing's thermal speed whose nθ is withheld: its speed and share of the limit, from which the withheld nθ can be worked out); the check is flagged not determined there, the solve tool tells the Assistant never to quote a figure so flagged, and the check's row and every display say "not determined", never a pass; - a rack pinion shaft not rated at its mesh's peak because its only drive is solved for one power direction (the Dual-pinion EPS example's SH5, driven by worm pair WG1) is said, in the verdict's and the report's short reason, to be refused because its torque sources cannot be shared; its check's own note gives the right cause; - a design search takes no result from a design with any check not determined, so a project whose steering duty leaves a warning over its states may find no design meeting a shaft or bearing requirement (the safe side; a later release decides each figure on its own); - `Life0utside::against_bounds` keeps a maximum branch no caller uses (the search reads the hybrid flag directly); a later release removes it; - the Overview's governing check of a module, between two checks not determined, keeps the first in the list (the bearing listed first) where the engine picks by id, so the module tile and the verdict can name different checks, both said not determined; - a variant comparison can mark as best the life, and its check's margin, of a bearing the engine marks unrated (its designation contradicts its geometry), though the bearing's own results say its figures are of the bearing as modelled: compare such a design only after its designation is cleared or corrected; - a comparison of three or more designs can leave out a result row whose neighbouring columns each agree within the tolerance while the first and last do not (10 000, 10 000.8 and 10 001.6 h); `--all` shows it; - a comparison of designs whose life checks all pass on a hybrid's basic life (its modified life outside its rating method, so no best) leaves that check row out of the default CLI comparison and the report, though the minimum lives and the checks' shares differ (a hybrid 6309 at a 1 million h minimum against 4 million h); `--all` shows it. - a hybrid whose modified life fails its minimum keeps that failing check's number in its check row (the fail stands, on the safe side; the number rests on a_ISO, defined for steel bearings only), and so does any life check failing outside its rating method (a steel bearing turning below κ = 0.1, or with its balls in four-point contact): the report's Summary of checks and its Summary page's Governing checks show its share and draw its utilisation bar like any other (a 30207 on 2 / 1 cSt oil against a 20 000 h minimum), though that share cannot be compared with another check's; the governing pick and the orders leave it unranked, and the typed results' report follow-up shows no bar for it. Show the calculation of a hybrid's life explains the modified life, said not determined, not the basic life its passing check goes by; - the Assistant's shortened record of an earlier solve lists a failing life check with its value (a hybrid QJ 207 in four-point contact: 94 476 h against a 1 000 000 h minimum) but without the check's outside-method flag or reason, though the same record says the bearing's life is not determined; the fail stands, and the full solve the Assistant reads carries the flag; - over a duty cycle whose cases stay within the rating method, the Assistant's solve data still gives the nominal summary's modified life where the loads as entered put κ below 0.1 (the Overhung pulley's BR2 on 10 / 2.5 cSt oil at 150 °C nominal, 35 °C in its one case: 107 191 h), with no note that it is outside the method; the duty's life, which the check and every display go by, is right (a nominal summary in four-point contact is marked); - Show the calculation of a check that fails with an unbounded share says it fails "whatever the value of the checked quantity"; for an axial preload that lifts off (a cross-located pair whose housing runs hotter than its shaft) the share is of the lifted-off

state only, and the same check passes once the temperatures no longer lift it; the check's message gives the cause; - "Make it pass" on a check not determined still steers its search by the check's share as modelled (a proposal is made only where the check is determined and passes after the change), and its "tried" list may give that share as the closest reached; - the load-spectrum card's chart still plots a bearing's damage shares where its table withholds them (a case's life outside the rating method with the life check failing); read the table; - the report's summary page labels a passing hybrid's life "modified reference rating life" before the parenthesis that says it is the basic life its check goes by; the bearing section is right; - the agent guide says a hybrid's check `value` is its basic life; where the check fails (`outsideMethod`) it is the modified life it fails on; - a variant comparison of a steel and a hybrid bearing that both pass their life check leaves that row out of the default CLI comparison and the report, though the two columns go by different lives (the row's `labels` say which); `--all` shows it; - the Overview's utilisation chart plots the share of a check failing outside its rating method beside the others, ordered first among the failures but drawn as if comparable; the fail itself is shown; - a hybrid roller bearing on estimated geometry passes on the lower of its two methods' basic lives (correct), but the report's summary page names that life with the other method's label, the Assistant's basis text points at `basicLifeHours` (the picked method's, possibly the higher) instead of the check's value, and a check made not determined by the other method's lower basic life does not quote it; the bearing finder's `basicLifeHours` and the bearing page's basic life rows show the picked method's too; the check's value and status are right; A later release types every result figure as determined or not determined, and every display, ranking and short message reads that type: this closes the ranking, display and short-message limits above for good.

- A speed set again as it is, for a later release (user test 4 reviews): on a pair, worm, drive or rotor it makes its shaft the train's input; on a shaft that follows another (`updateShaft`) it is given beside the input instead, and the next ratio change makes the two disagree, an error that names both. The app never sends an unchanged value, so there a speed becomes the input only when it changes. The note under a given speed calls it "the train's input" when any shaft in its placement group follows a speed, also one of another train. In the base, with a variant whose ratio, sense or case speeds differ, a speed sent unchanged on an item is refused like a new one. The stale-speed warning's hint prints the speed to six digits, so typing it by hand (1410.53 for 1410.526…) moves the input where the Fix button, which sends it exactly, does not. When the input moves, a measured spectrum's case names keep the old input's speed ranges, and a series too long to keep keeps its peak speed in the description. The base's variant check compares each variant with the input scaled as before, so it also refuses a move on another train than the cases' reference, or one whose variant would only read the cases at the new input, both on the safe side. A rotor whose speed was not entered (0) and is typed as its shaft's speed takes it without moving the input.

- Steering duty, for a later release (user test ut4): - A worm pair or a gear pair over a steering duty keeps its own required life (a worm pair rated at its worst state as if that lasted its life, a gear pair over the cases at its own life's cycles); the duty's length checks the ball nut's, the ball screws' and the bearings' lives. - Editing a mesh's Peak rack force for static rating in the duty's refusal card makes the duty not computed again, so the card (and the field in it) goes until Compute duty runs; with two refused meshes, the second is edited after recomputing, or on its mesh's own panel. - The drivetrain's natural frequencies (system modes) are dropped as out of date when another shaft is shown, though they do not depend on it; Compute them again. - While the duty computes, a check it decides reads "computing…" in every list of checks the app shows, also another report's (a version compared) of the same kind; the project's verdict line keeps the engine's count (a failure at the state counts) and says the duty is being computed. - A design search's work budget counts a remembered result of the steering duty at every request within one solve, so a design that runs on and on is stopped later than it could be (its time limit and Cancel still stop it); the engine's other remembered results (a gear tooth root's or a housing's finite elements) are not counted, which the budget's floor of 5000 steps covers.
- Steering misuse events (user test ut4): the misuse events rate the pinions' tooth roots (their contact stress is reported, not checked) and the pinions' shafts statically; the ball nut and its bearing are rated in the steering state and over the duty's strokes, not at a misuse event's peak, and the elements keep their own peak factor for a static peak. A worm pair that drives an assist pinion (a plastic wheel too), the other shafts and the bearings are not rated at a misuse event either, and only the report's method notes say so: the results do not list them as not checked for it. A mesh's peak rack force for static rating is a case of its own, the whole force through that pinion with the assist failed, rated for root and contact, so the same kerb strike can pass as a misuse event and fail there; the checks do not name it as the assist-failed case. Both err on the safe side or are disclosed; listing them on screen is for a later release (RR1 U-A25).
- A teeth end (`teethEnd`) is rated by DIN 743-2's notch factors of a shaft with involute teeth without a hub (4.2.2) and an eq. (9) shoulder or groove of its shape, not by a published method for a hob's run-out, which no source at hand gives (see "Shaft strength references"); the mesh's own load on the teeth is the rack mesh's or gear pair's tooth-root rating. A worm's thread end takes none (a strength notch instead). The Shaft module's 3D draws a rack pinion's teeth cut into the shaft; a gear pair's pinion cut into its shaft is drawn there at its root diameter as before (the Assembly and Steering views draw its teeth). The 3D views draw a single toothed section's ends; teeth over several sections at the root diameter are rated over all of them but drawn on the one at the mesh.
- The gear-seat check takes a worm's threaded length as DIN 3975's estimate ((11 + 0.06·z_2)·m_x for one or two starts, (12.5 + 0.09·z_2)·m_x for more), as the 3D view draws it: the model has no field for it yet (a later release).
- A plastic worm wheel in intermittent duty takes the lower S_Hmin 1.25 on the checkbox alone. Above its material's continuous temperature limit (PA 66: 120 °C) it is not determined up to the hottest data curve (180 °C): run time and pause are not inputs yet.
- A plastic worm wheel's flank band rates the coolest temperatures at the whole life's load cycles, which errs on the safe side for a wheel that warms up within minutes. The data start at 20 °C, so a colder start leaves the flank not determined rather than rated (a cold-start rating below 20 °C, from moduli and flank strengths measured colder, is for a later release; the Column EPS assist gear example shows it), and with the lowest wheel temperature left empty the band starts at 20 °C: a colder start is then not looked at. A given wheel modulus E_2 holds at ϑ_M only, so its flank is not determined unless the lowest wheel temperature is given equal to ϑ_M.

- For a later release: a plastic cylindrical gear (VDI 2736-2) is not checked for a cold start. It rates its flank at its computed flank temperature only, though a cooler plastic is stiffer and takes a higher flank pressure faster than its strength rises (steel on PA 66: S_H at 20 °C about 0.73 times that at 70 °C), so a pair that passes warm can fail at ambient. The plastic worm wheel is rated down to its lowest wheel temperature; the cylindrical gear should take the same band.
- With gravity on, a shaft placed by a bearing in another shaft (seated in it) is refused when the two are solved together and its frame is turned by what places that shaft (a universal joint, a worm pair, a rack mesh): the model of shafts solved together leaves the bearings between them out, and with them that placement (0.3.0). Place the shaft by another connection or at coordinates, or switch gravity off.
- Over steering-derived load cases a shaft's FKM damage check (with a number of cycles) is undecided where it passes when the shaft's loads follow the speed (0.3.0): every shaft of a mounted stage (a gear or worm pair, a rack mesh, a drive), whose mesh efficiency is rated at each case's speed and enters a member's torque, and whose belt's pull rises with the square of the speed, and every shaft joined to one by a coupling or a bearing between shafts. The cases' chord split bounds that damage only where the stresses follow the force alone. In belt-drive with SH1's FKM over 800000 cycles and one-way strokes over a 126–130 mm/rev lead, the cases passed 6.681 at 6.681179 where the strokes give 6.680947.
- A steering duty whose strokes reverse the rack force (a return pass against a resisting load, a signed load below zero) is refused at `steering.duty` beside anything the load cases reach that a case cannot reverse (0.3.0): a mounted gear or worm pair or rack mesh, a drive, a coupling, a rotor, a bearing with loads of its own, a gear pair rated over the cases. The derived cases reverse only the entered shaft loads and ball screw forces, so the others would be rated at the forward loads on the reversed passes (a rotor's torque, unreversed, cancelled the reversed input in one review's case: 18.444302 where 9.222151 is right). In gearbox-single-stage with a rack mesh and resisting strokes with return passes, BR1 passed 5.5 at 5.778100 where the reversed pass gives 5.260948. One-way strokes are rated; reversing stages over the duty come in a later release.

- A steering duty over a steering system (0.3.0) rates the rack meshes' teeth, and with strokes the system's shafts, the bearings linked on them that take the load cases, its worm pairs and its ball nut over its states (see Steering system). Not rated over the duty (a later release): a mounted gear pair (rated at the system's `state`, its checks not rated over the duty, never a pass: its rating takes its solved shafts' KHβ, which the states do not give it yet), and likewise fits, keys and housings. A rotor's air gap is not rated over the states (its band between them is not drawn; never a pass: rate it at the duty's heaviest state). A bearing's warning over the states (a minimum load undecided at its valley) is never satisfied, and its gyroscopic slip (the high-speed model's) is the system's state's, a pass not rated over the duty. Beside such a duty a design search's metrics are refused but the shaft's mass (their values over the states have no bound between them, and some items' results are the system's state's): search with requireChecks, whose checks take the bands. The ball nut's life is the states' Miner sum (as a bearing's), its static safety the worse of the states' and the system's own state's. Left at the system's `state`: the couplings, a bearing with `useLoadCases` false, and everything at a misuse event (the events rate the pinions only; the elements keep their own `peakFactor` for a static peak, which over the duty multiplies every state's loads, as over load cases). A worm pair and a drive take the worst of their states, each rated as if it lasted their life: on the safe side, as DIN 3996 holds no load spectrum here. A shaft's states are each rated with their stresses fully reversed (as a mounted ball screw's states are), on the safe side where a stroke does not turn round. The band of a bearing's checks is drawn from its life, static safety, contact stress and thermal speed in the states, relative to each check's utilisation. Two supports in seats of one housing (whose seats push each other with the bearings' loads) leave the elements unrated over the duty (a warning at `steering.duty`, their checks never a pass): the states do not couple the seats again (a later release). The project's own load cases beside such a duty rate the shafts and bearings instead (a warning at `steering.duty`), and their checks are then not rated over the duty (never a pass): remove the load cases to rate them over it. The elements' states are some four times the duty points (284 for the Dual-pinion EPS's duty), each a state solve and a solve of every shaft: at most 1200 (`system_cases::CASE_LIMIT`) and at most 12,000 solves (`WORK_LIMIT`: each state one, and one for each shaft and each bearing on a shaft; some 25 s natively and 40 s in the browser), past which the elements are not rated over the duty (a warning at `steering.duty`; their checks never pass) while the teeth are. Measured under load on the build machine (one process, natively): the Dual-pinion EPS's duty 9.3 s for the teeth and 19 s in all; the Belt-driven ball-nut and Column EPS examples given its duty 2.4 s and 2.8 s for the teeth, 7.0 s and 5.2 s in all (168 and 88 states); the browser's engine took about the same under that load (17.9 s, 7.6 s and 6.3 s). It refuses a stroke's `torque` table (the system splits the rack force, not a pinion's torque); a rack mesh off the system's rack is not rated over the duty (a warning; it keeps its rating at its own load). Each duty point is a state solve (some 15 to 30 ms natively), at most 4000 per duty. Rating the teeth of two pinions is the slow part: the Dual-pinion EPS template's duty (a parking, a manoeuvring and an on-centre stroke, both ways) takes some 4.7 s natively and 7.3 s in the browser's engine on the build machine (some 20 s natively on a laptop, 4 s of it the state solves); the app computes it on request, with progress and Cancel (0.3.0), and solves each edit without it. Positions past the rack's travel are refused (each point is a state of the system, within its stops). The lift is rated within 1e-4 mm of the states' between points (the contact ratios within some 4e-5), not raised; the forces are raised to bound theirs. Misuse events are rated at standstill (vehicle speed 0, forward efficiencies, no friction).

- **Add a steering** puts the rack's meshes in the steering's train and takes the torques off the driver, the shafts coupled to it and the pinions' shafts. A gear pair, worm pair or drive between the driver and the rack still rated at its own torque is left as it is. The steering then reads "not solved", and its error offers the fix (`torqueMode train`). Apply that fix, or set it in the item's module.
- A dry plastic worm pair with no friction given fails a minimum back efficiency it misses at VDI 2736-2 Table 2's 0.20, though a lower friction (the table gives no scatter) could pass it: a pass there is "not decided", a fail stays a fail, on the safe side. Give the friction measured on the pair (review of small-034, for a later release).

Plastic cylindrical gears: VDI 2736-2

Thermoplastic spur and helical gears by VDI 2736-2: temperatures, tooth root, flank, wear and deformation.

Methods and standards

Plastic gears after VDI 2736-2: involute geometry, contact ratios, tangential force and Z_H of the ISO 6336 pair; loss factor H_V = π·(u + 1)/(z_1·u·cos β_b)·(1 − ε_α + ε_1² + ε_2²) (Ohlendorf, eq. 8); flank and root temperature of each gear ϑ = ϑ_0 + P·μ·H_V·(k_ϑ/(b·z·(v·m_n)^0.75) + R_λ,G/A_G)·ED^0.64 (eqs. 7, 9) with k_ϑ 6300 and 900 (plastic on steel), 9000 and 2100 (plastic on plastic) or 0 (circulating oil) (Table 3) and R_λ,G 0, 0.03 or 0.06 K·m²/W for open, partly open and closed housings; root stress K_A·F_t/(b·m_n)·Y_Fa·Y_Sa·Y_ε·Y_β with the load at the tip (form factors of ISO 6336-3 with the load at the virtual tip diameter), Y_ε = 0.25 + 0.75/ε_αn, against σ_FG = 2·σ_FlimN (eq. 13); flank stress Z_E·Z_H·Z_ε·Z_β·√(K_A·F_t·(u + 1)/(b·d_1·u)) with Z_E from the moduli at the flank temperatures and Z_β = √cos β, against σ_HlimN; strengths at the root and flank temperatures and each gear's cycles; tip deformation 7.5·F_t/(b·cos β)·(1/E_1 + 1/E_2) against 0.07·m_n; wear depth 2π·T·N_L·H_V·k_w/(b·z·l_Fl) over the active flank length from the usable root diameter to the tip (eqs. 19, 20; k_w a fifth greased, none oiled).

The root strengths of POM and PA66 follow the curves behind VDI 2736-2 Bild 3 (footnote 8) and PA66's flank strength those behind Bild 6 (footnote 12, greased or oiled), over temperature and 10^5 to 10^8 load cycles, held at 10^8; the rest of the material table (moduli, POM's flank strength, PA66 GF30 and PEEK, wear coefficients: values at 10^6 and 10^8 cycles at reference temperatures, linear in temperature and log-linear in cycles, as own data) holds typical values, to be replaced by the maker's data; friction defaults (Table 2) 0.20 dry on steel, 0.28 dry POM/POM, 0.40 dry PA/PA, 0.18 dry POM/PA, 0.30 other dry plastic pairs, 0.09 greased, 0.04 oiled.

Validation

61 of 61 cases within their tolerance, in 4 groups.

Worked examples of standards and books

VDI 2736-2 example A1: eccentric press (PA66 wheel), dry and greased

Kind of reference	Worked examples of standards and books
Implements	VDI 2736 Part 2:2014-06, eqs. (1) to (20), Tables 2 to 4
Reference source	VDI 2736 Part 2:2014-06, Annex A, example A1
Cases	21 of 21 within tolerance

Tolerances: half a unit of the printed last digit plus what the example's rounded contact ratio or torque moves the value; chart readings (Y_Fa, Y_Sa, ZH) 1 %; temperatures printed as approximate whole degrees ±0.6 °C. The strengths are the example's chart readings, given as own material data.

Case	Reference	Axomek	Deviation	Tolerance	
F_t	2887 N	2886.591 N	0.4089774 N	0.5 N	pass
v_t	2.29 m/s	2.293363 m/s	0.003362637 m/s	0.005 m/s	pass
u	3.32	3.318182	0.001818182	0.005	pass
ε_α	1.62	1.614662	0.00533772	0.006	pass
ε_1	0.82	0.8190348	0.0009652216	0.006	pass
ε_2	0.8	0.7956275	0.004372498	0.006	pass
H_V	0.129	0.1280737	0.72 %	1 %	pass

Case	Reference	Axomek	Deviation	Tolerance	
ϑ_Fuß	34 °C	33.90156 °C	0.09844048 °C	0.6 °C	**pass**
ϑ_Fla (dry)	57 °C	57.31092 °C	0.3109166 °C	0.6 °C	**pass**
Y_Fa	2.16	2.160656	0.03 %	1 %	**pass**
Y_Sa	2.01	2.008402	0.08 %	1 %	**pass**
Y_ε	0.713	0.7144934	0.001493417	0.002	**pass**
σ_F	19.9 MPa	19.88874 MPa	0.01126419 MPa	0.1 MPa	**pass**
S_F	3.015075	3.016783	0.057 %	0.5 %	**pass**
l_Fl	11.2 mm	11.21137 mm	0.01136787 mm	0.05 mm	**pass**
W_m at l_Fl = 2·m_n	1.56 mm	1.54858 mm	0.73 %	1 %	**pass**
ϑ_Fla (greased)	42 °C	42.28991 °C	0.2899125 °C	0.6 °C	**pass**
Z_H (greased)	2.4	2.400639	0.027 %	1 %	**pass**
Z_ε (greased)	0.891	0.8916909	0.0006908507	0.0015	**pass**
σ_H (greased)	31.7 MPa	31.64238 MPa	0.18 %	0.5 %	**pass**
S_H (greased)	1.48265	1.48535	0.18 %	0.5 %	**pass**

VDI 2736-2 example A2: photocopier feed rollers (POM pinion)

Kind of reference	Worked examples of standards and books
Implements	VDI 2736 Part 2:2014-06, eqs. (1) to (13), Tables 2 to 4
Reference source	VDI 2736 Part 2:2014-06, Annex A, example A2
Cases	15 of 15 within tolerance

Tolerances: half a unit of the printed last digit plus what the example's rounded contact ratio or torque moves the value; chart readings (Y_Fa, Y_Sa, ZH) 1 %; temperatures printed as approximate whole degrees ±0.6 °C. The strengths are the example's chart readings, given as own material data.

Case	Reference	Axomek	Deviation	Tolerance	
F_t	44 N	43.97702 N	0.02297625 N	0.05 N	**pass**
v_t	0.16 m/s	0.159174 m/s	0.0008259722 m/s	0.005 m/s	**pass**
a	30 mm	30 mm	0 mm	1×10^{-9} mm	**pass**
ε_α	1.6	1.598051	0.001949388	0.006	**pass**
ε_1	0.81	0.8102138	0.0002138421	0.006	**pass**
ε_2	0.79	0.7878368	0.00216323	0.006	**pass**
u	2.75	2.75	0	1×10^{-12}	**pass**
H_V	0.182	0.181824	0.097 %	1 %	**pass**
ϑ_Fuß	49 °C	48.75203 °C	0.2479751 °C	0.6 °C	**pass**
ϑ_Fla	77 °C	76.75593 °C	0.2440729 °C	0.6 °C	**pass**
Y_Fa	2.94	2.937142	0.097 %	1 %	**pass**
Y_Sa	1.61	1.612346	0.15 %	1 %	**pass**
Y_ε	0.719	0.7193218	0.0003218063	0.002	**pass**
σ_F	21.4 MPa	21.40101 MPa	0.001006479 MPa	0.1 MPa	**pass**
S_F	2.803738	2.803607	0.0047 %	0.5 %	**pass**

VDI 2736-2 example A3: card reader (POM pinion), with wear

Kind of reference	Worked examples of standards and books
Implements	VDI 2736 Part 2:2014-06, eqs. (1) to (20), Tables 2 to 4
Reference source	VDI 2736 Part 2:2014-06, Annex A, example A3
Cases	15 of 15 within tolerance

As A1 and A2; the wear W_m is compared at the example's flank length 2·m_n (footnote 13), Axomek's eq. (20) wear scaled by l_Fl/(2·m_n).

Case	Reference	Axomek	Deviation	Tolerance	
F_t	28.6 N	28.71969 N	0.42 %	0.6 %	pass
v_t	0.28 m/s	0.2785546 m/s	0.001445451 m/s	0.005 m/s	pass
ε_α	1.75	1.747908	0.002092227	0.006	pass
ε_1	0.82	0.8190021	0.0009978973	0.006	pass
ε_2	0.93	0.9289057	0.00109433	0.006	pass
u	3.75	3.75	0	1×10^{-12}	pass
H_V	0.112	0.1116666	0.3 %	1 %	pass
ϑ_Fuß	49 °C	49.19601 °C	0.1960053 °C	0.6 °C	pass
Y_Fa	2.64	2.638911	0.041 %	1 %	pass
Y_Sa	1.68	1.677871	0.13 %	1 %	pass
Y_ε	0.679	0.6790844	8.4424×10^{-5}	0.002	pass
σ_F	15.7 MPa	15.70089 MPa	0.000886778 MPa	0.13 MPa	pass
S_F	4.585987	4.585728	0.0056 %	1 %	pass
l_Fl	1.8 mm	1.814477 mm	0.01447647 mm	0.05 mm	pass
W_m at l_Fl = 2·m_n	0.09 mm	0.09159257 mm	0.00159257 mm	0.005 mm	pass

Standards’ tables and printed values

Root strength σFlimN of the built-in POM and PA66

Kind of reference	Standards’ tables and printed values
Implements	VDI 2736 Part 2:2014-06, Bild 3 (footnote 8)
Reference source	VDI 2736 Part 2:2014-06, Table 5 (whole MPa)
Cases	10 of 10 within tolerance

A sample of the table's values across its temperatures and cycle counts, within half its printed unit.

Case	Reference	Axomek	Deviation	Tolerance	
Pom at 20 °C, 1e6	50 MPa	50.23829 MPa	0.2382938 MPa	0.5 MPa	pass
Pom at 40 °C, 1e5	62 MPa	62 MPa	0 MPa	0.5 MPa	pass
Pom at 60 °C, 1e7	33 MPa	32.92429 MPa	0.07571318 MPa	0.5 MPa	pass
Pom at 80 °C, 1e8	20 MPa	20.04755 MPa	0.04754573 MPa	0.5 MPa	pass
Pom at 100 °C, 1e6	26 MPa	26.23829 MPa	0.2382938 MPa	0.5 MPa	pass
Pa66 at 20 °C, 1e5	77 MPa	77.07353 MPa	0.07352854 MPa	0.5 MPa	pass
Pa66 at 60 °C, 1e6	32 MPa	32.06113 MPa	0.06113097 MPa	0.5 MPa	pass
Pa66 at 80 °C, 1e7	18 MPa	18.19752 MPa	0.1975207 MPa	0.5 MPa	pass
Pa66 at 100 °C, 1e8	10 MPa	10.19114 MPa	0.1911365 MPa	0.5 MPa	pass

Case	Reference	Axomek	Deviation	Tolerance	
Pa66 at 120 °C, 1e6	12 MPa	11.75601 MPa	0.2439869 MPa	0.5 MPa	**pass**

Stated limits

Standards and scope **(help: modules/plastic-gears.md)**

VDI 2736 Part 2 (2014) for the temperatures (3.2.1), root (3.2.2), flank (3.2.3), wear (3.2.4) and deformation (3.2.5), with the geometry, contact ratios and Z_H of the ISO 6336 gear pair and the form factors of ISO 6336-3. The loss factor HV is eq. (8). For helical gears Y_ε takes the virtual contact ratio εαn and Z_β = √cos β, as Annex B refers to (eq. (11) writes ε_α and 3.2.3 approximates Z_β ≈ 1; spur gears are the same either way). The program's tests recompute the guideline's three worked examples (Annex A).

Not covered: plastic worm wheels (the module rates cylindrical gears by VDI 2736 Part 2; a worm wheel needs Part 3, and the worm module's DIN 3996 rates bronze and cast iron wheels only), material values the guideline does not give as curves (its Part 1 data; the table's other values are typical), the design estimate of 3.1, the efficiency of 3.4 (the guideline gives experience values only, 0.80 to 0.90), moisture uptake and its effect on PA, creep under long standstill loads, the steel gear's own rating, and mounting the pair on the project's shafts.

Known limits

- A plastic gear pair without a peakTorque has no peak load check (VDI 2736-2, 3.3): its peak safety is listed "not assessed" (no peakTorque given) against `limits.minPeakSafety` 1.5, so the project never passes `axomek check --strict`. Give the pair's peakTorque, or clear `limits.minPeakSafety` when no peak above the nominal torque occurs.

Planetary gear sets

Planetary stages: speeds, torques, load sharing and the planet carrier.

Methods and standards

Planetary sets: each planet's sun mesh (external) and ring mesh (internal) rated by ISO 6336 at the speeds relative to the carrier, the sun torque shared with the load sharing factor K_γ (unless given, by the number of planets: IEC 61400-4:2012 Table 3 for three to seven, ANSI/AGMA 6123-C16 Table 7 for two, eight and nine, ISO/TS 6336-20:2022 Formula (14) from ten; a lower value only when measured); planet teeth with reverse bending, their root safety times the mean stress factor YM 0.7 (the planets' bending safety); K_Hβ by method C from the carrier when it is given; load cycles at the speeds relative to the carrier, N per relative revolution for the sun and the ring (every tooth meets all N planets) and one for a planet tooth in each mesh; assembly, coaxiality and neighbour conditions checked; planet pin load the vector sum of both meshes' tooth forces (with K_γ) and the planet's centrifugal force Fc: one transverse force Fbt = 2000·T/db,sun along each line of action (the planet's torque balance on its one base circle), Fbt·(cos αw1 + cos αw2) across the centre line and Fbt·(sin αw1 − sin αw2) + Fc along it, which is √((2·Ftw·K_γ)² + Fc²) for the standard counts z_r = z_s + 2·z_p without shifts (a planet bearing's duty cycle scales the tooth forces by each case's load factor and Fc by its speed squared); Fc from the planet as a steel disc of its reference diameter less its bore (the planet's bore, or the planet bearing's outside diameter from its mounting); a helical planet's two meshes push it axially with Fbt·tan βb each, opposite at the sun's and the ring's pitch point, and their couple Fa·(rw1 + rw2) about the tangential axis loads the planet bearing as a moment (scaled with the tooth forces); the pins of the carrier's Young's modulus (steel unless given).

A ring with a full addendum meets the planets' root fillets unless they are shifted: without a given ring tip diameter the ring's tip is shortened just enough to meet the planets at their form circle (the standard tip where that clears them), as for any internal gear pair without given tip diameters.

Internal gears follow ISO's signed convention; their root section comes from their generation by a shaper cutter (as many teeth as the pinion unless given): the fillet is the cutter tip's trochoid offset by its tip radius, the critical section where the fillet's tangent is at 60° to the tooth centre line (ISO 6336-3), the load at the outer point of single contact, in the virtual spur gears.

Planet carrier by finite elements: two annular steel cheeks joined by webs between the planets, eight-node bricks with incompatible modes round the axis, the webs' edges on the mesh at the angle given; the pins load both cheeks tangentially at the planets' radius, spread evenly over their footprint, the output cheek held at its hub; the cheeks' mean twist against each other at the pins gives the wind-up stiffness, which tilts the planets. Every planet loads the carrier alike, so one planet's span is solved, its displacements repeating round the axis.

The bricks err on the stiff side, so the mesh (8 rings, about 16 sectors a planet with at least 4 across a web, 3 layers a cheek) is refined twice as fine in each direction in turn and the wind-up raised by the change and by the error left after it, Richardson's extrapolation with an order of 0.75 (the slowest seen in one direction; refined evenly the mesh converges with an order of 1.07 to 1.21, near the 1.09 of the field at the webs' inner corners), at least 2 % of the mesh's above the refined meshes'. Against meshes refined evenly to three times as fine and extrapolated, the wind-up used is 2.5 to 4.2 % above, the mesh's own 6 to 15 % below. Where the refinements change the wind-up by more than 10 %, the mesh is refined where it changed most, up to 3 times; a carrier still above, or one a finer mesh makes more than 2 % stiffer (a cheek held on a small bore or none), is refused: give the carrier's torsional stiffness instead.

Validation

14 of 14 cases within their tolerance, in 2 groups.

Standards’ tables and printed values

Planetary sets: default load sharing factor K_γ

Kind of reference: Standards’ tables and printed values

Implements: IEC 61400-4:2012 Table 3; ANSI/AGMA 6123-C16 Table 7; ISO/TS 6336-20:2022 Formula (14)

Reference source: IEC 61400-4:2012 Table 3 (3 to 7 planets), ANSI/AGMA 6123-C16 Table 7 (2, 8 and 9 planets), ISO/TS 6336-20:2022 Formula (14) rounded to two decimals (10 to 12 planets)

Cases: 11 of 11 within tolerance

Case	Reference	Axomek	Deviation	Tolerance	
2 planets	1.16	1.16	0	exact	**pass**
3 planets	1.1	1.1	0	exact	**pass**
4 planets	1.25	1.25	0	exact	**pass**
5 planets	1.35	1.35	0	exact	**pass**
6 planets	1.44	1.44	0	exact	**pass**
7 planets	1.47	1.47	0	exact	**pass**
8 planets	1.52	1.52	0	exact	**pass**
9 planets	1.61	1.61	0	exact	**pass**
10 planets	1.66	1.66	0	exact	**pass**
11 planets	1.71	1.71	0	exact	**pass**
12 planets	1.75	1.75	0	exact	**pass**

Closed-form solutions

Planetary stage: ratio, centre distance, mesh force and pin load

Kind of reference: Closed-form solutions

Implements: —

Reference source: Willis's ratio 1 + z_r/z_s and the statics of a stage with sun z 21, planets z 33, ring z 87, m 3 mm, 3 planets, 300 N·m at the sun

Cases: 3 of 3 within tolerance

Case	Reference	Axomek	Deviation	Tolerance	
Ratio 1 + z_r/z_s	5.142857	5.142857	0	1×10^{-12}	**pass**
Centre distance (z_s + z_p)·m/2	81 mm	81 mm	0 mm	1×10^{-9} mm	**pass**
Mesh tangential force 2000·T·K_γ/(n·d_s)	3492.064 N	3492.064 N	0 N	1×10^{-6} N	**pass**

Not validated against a published reference

- No worked planetary rating: the cases are the load sharing factor’s source tables and the stage’s statics
- The planet carrier’s finite elements

Stated limits

Standards and scope (help: modules/planetary.md)

Each mesh is rated by ISO 6336 as described for gear pairs, the ring as an internal gear whose root comes from its shaper cutter (as many teeth as the planet). K_γ by the number of planets follows IEC 61400-4, ANSI/AGMA 6123-C16 and ISO/TS 6336-20 unless given (ISO 6336 itself gives none; a lower measured value may be entered). The carrier model covers pin bending and the cheeks' wind-up; it does not model the carrier's radial flexibility, planet position errors or a floating sun. Scuffing, load spectra and the loaded tooth contact are not available for planetary sets; for a detailed look at one mesh, rate it as a gear pair with the carrier's misalignment entered as **Mesh misalignment from deformation f_sh**.

Known limits

- `axomek check --strict` and `axomek solve --strict` exit 3 for every file with something its method does not assess, which no input clears yet (4 where a check fails and the file was solved completely; a file that was not, an unknown field under `--strict` among it, exits 3 first): a rated key or spline (reversals and load peaks), a rated fit (its shaft seat), a belt or chain drive (its capacity), a fatigue-rated weld (its toes), a yoke on its stop, a support with a designation only and a planetary set without a planet bearing. Check those by hand and use `axomek check` without `--strict` for such files (it still exits 4 on a failing check and 3 on one not determined); a way to mark an item as checked by hand is for a later release.
- A refusal's fix buttons, all for a later release: for an item whose id does not start with a capital letter, a suggestion is found only when it changes that item itself; in a locked project the buttons are disabled without saying why; the item solves a suggestion takes while something is refused are not counted in the solve-work table; the ball screw's lead table follows ISO 3408-1's nominal leads, not yet checked against the standard. A planet bearing whose outside diameter reaches the planet's root diameter is refused and its field marked, with no button (no number to search).
- Over a steering system's duty states (0.3.0), all on the side of saying more or of reporting less, for a later release: a rack buckling span that exists in some of a span's states and not in others (the bar moves under the supports) draws no marginal band over that change; a shaft standing still at the system's `state` takes FKM's checks over a number of cycles at the worst state for all its cycles instead of their Miner sum; a state that gives the ball nut no thrust can leave its rating over the duty refused; the system's own state, kept among the worst-case checks, is rated with its mean and amplitude stresses where the duty's states take theirs fully reversed (a state heavier than every duty point is rated more leniently than they are); a refused duty marks every drive's checks, also those of a drive off the steering's train; a bearing's band for its checks other than life and static safety is drawn from those quantities' bands rather than its own; `atState` does not list bearings off the shafts or planet bearings (rated at their own loads), nor the couplings.
- A double-walled planet carrier's wind-up from its finite elements is taken on the safe side of its mesh's error: 2.5 to 4.2 % above that of a mesh refined without end in the tests (webs of 10° to 90°, 3 to 5 planets). A carrier its mesh does not settle on within 3 refinements, or that a finer mesh stiffens (a cheek held on a small bore or none), is refused with what to give instead (`carrier.torsionalStiffness`, or the hub's bore). The model has sharp inner corners where webs meet cheeks and holds the whole hub bore of the output cheek; fillets and a hub held over part of its bore are not modelled. A long closed carrier (webs all round, a gap of 400 mm) takes several seconds.

- Messages and refusals, for a later release (RR1 branch 22, what it left): - Validation errors from an element's own engine are placed at the field their message starts with, but about 75 of the project's validation errors still give no hint beyond their message ("must be positive"), and a JSON value of the wrong type in `axomek apply` is refused without its field's name (RR1 TA-F53, agent F5). - A power typed into a torque field is converted at the speed field's value at that moment, said only until it is applied; the torque then stays when the speed changes (RR1 U-A28). A key placed on a shaft keeps its own diameter and torque, and a keyed section can't be swept, optimised or proposed by `fix` (U-A27). "What matters most" can rate a screw's root diameter as not moving its speed check (U-A24, not traced). The planetary refusals and fixes U-C3 lists (a nearby set the search finds but the card does not, a ring shift offered before a planet change, the default planet count beside a typed ring), the tilt note shown above a failing minimum load, a warning's reason shown only on the Overview, "Driven by" offering stages that close a loop, and a single tapered roller bearing's raw "could not reach equilibrium" in the bearing finder. - The Assistant's results give a life in whole hours, as the app shows it, so a life just under its minimum (19 999.6 h against 20 000 h) reads as the minimum there; its check, failing, keeps the exact value. Where one case of a duty cycle puts a bearing outside its rating method (κ below 0.1, balls in four-point contact), the Assistant is told to quote none of its lives, a hybrid's basic life included, not even the modified life at the loads as entered where that one is within its method (the safe side). The room the app makes to keep a field in place is kept through a mouse click on a button in the panel, not yet through a tap on a touch screen (Opus review of RR1 branch 22). - A support's own Designation is shown as its linked bearing's and can't be typed over while a bearing is linked; the value saved in the file stays and comes back when the bearing is unlinked.
- A QJ 207 linked to no support whose loads are entered by hand as a starter's (Fa 1500 N, Fy 2000 N, the earlier one's, or Fa 3000 N, Fy 1000 N, today's; the rest 0) is warned about as rated on loads nobody entered; the warning is advice only. The test that every example and Add choice solves determined (R-956) solves Add Planetary on a blank project, not as a stage driven by another; does not fail on a bearing row left not set up (no example has one); and checks that each Add choice the app sends is covered, not that a covered choice's patch matches the app's.

Racks and rack meshes

Rack and pinion meshes rated by ISO 6336 and rack members as beams: strength and buckling.

Methods and standards

Rack meshes use ISO 6336-2/-3:2019 Method B with the wheel's tooth count tending to infinity, following the infinite-tooth device of ISO 6336-1:2019, 9.3.2.6 b).

The rack's root factors use its straight flank, circular fillet and the 30° tangent section; its FE comparison uses three straight teeth held 3.5 modules below the root plane.

Outside normal pressure angles 15°–25°, helix angles up to 30° or transverse contact ratios 1.0–2.5, the rating is flagged as outside ISO 6336's validated range and FE Y_FS is shown first.

Below transverse contact ratio 1 the root load is at the tip and the transverse load-distribution factor is 1; a total contact ratio below 1 does not establish uninterrupted engagement.

No published rack worked example is held: validation rests on large-wheel gear-limit convergence, independent hand calculations of geometry, forces, rack root factors and Annex B YM, and a rack FE comparison.

Reversed bending optionally uses ISO 6336-3:2019 Annex B with R = −1.2 times the lower-to-higher flank load ratio, reading B.1 and B.3's inconsistent signs as negative R; the 2006 Annex B has not been checked.

Each member's heat-treatment group is reported explicitly: flame/induction hardened roots take IF (Annex B Table B.1, M 0.4 endurance / 0.6 static), case hardening takes Eh (M 0.8 − 0.15 YS / 0.7). An unspecified heat treatment retains the legacy boolean default and must be confirmed for Annex B; a member's material kind overrides its material's heat treatment.

The alternative ISO 6336-5:2016, 5.3.3 method applies 0.7 to endurance and 1 to static strength, interpolating logarithmically between the knees; Annex B uses distinct endurance and static coefficients with the same interpolation.

Rack tooth cycles are entered explicitly or assumed equal to pinion rotations; this is not stroke counting and can underestimate short reversing duties.

A variable lead is rated as a local equivalent rack at the selected position with an unchanged base circle; root factors retain the nominal normal section and the manufactured variable flank needs separate validation.

Static safety uses the entered peak rack force, or the rated load when absent, with application factor 1 and load factors recomputed at that load.

All new requirements are disabled unless entered; example requirements are template values, not standard values.

Validation

8 of 8 cases within their tolerance, in 1 groups.

Closed-form solutions

Rack member: Euler buckling and the secant factor

Kind of reference	Closed-form solutions
Implements	—
Reference source	Euler's critical force $\pi^2 \cdot E \cdot I/(K \cdot L)^2$ for the four end fixities (K 1, 0.699, 0.5) and the secant formula's factor $1/\cos(\pi/2 \cdot \sqrt{(F/F_k)})$, Ø26 × 300 mm steel bar
Cases	8 of 8 within tolerance

The effective-length factors are the engine's: K 0.699 for pinned–fixed is the exact 0.6992 rounded, so those cases check the formula at that factor.

Case	Reference	Axomek	Deviation	Tolerance	
Critical force, Pinned–Pinned (K 1)	516582.7 N	516582.7 N	0 %	10^{-12}	**pass**
Secant factor at a quarter of F_k, Pinned–Pinned	1.414214	1.414214	0 %	10^{-12}	**pass**
Critical force, Pinned–Fixed (K 0.699)	1.0573×10^6 N	1.0573×10^6 N	0 %	10^{-12}	**pass**
Secant factor at a quarter of F_k, Pinned–Fixed	1.414214	1.414214	0 %	10^{-12}	**pass**
Critical force, Fixed–Pinned (K 0.699)	1.0573×10^6 N	1.0573×10^6 N	0 %	10^{-12}	**pass**
Secant factor at a quarter of F_k, Fixed–Pinned	1.414214	1.414214	0 %	10^{-12}	**pass**
Critical force, Fixed–Fixed (K 0.5)	2.0663×10^6 N	2.0663×10^6 N	0 %	10^{-12}	**pass**
Secant factor at a quarter of F_k, Fixed–Fixed	1.414214	1.414214	0 %	10^{-12}	**pass**

Not validated against a published reference

- The rack and pinion mesh rating and the rack duty: no worked example is compared
- Rack member strength at the teeth

Stated limits

Method notes and limits **(help: modules/racks.md)**

The beam uses the flat section's centroid and principal second moments, including the moment of the mesh force about the root-plane pitch line. Yoke equilibrium is solved with a piecewise spring law: seated against the pinion, lifting against the spring, then spring plus stop stiffness beyond clearance. Each mounted mesh is rated again at the rack's lift at its contact. DIN 743 fatigue takes each point's stresses in the state as fully reversed amplitudes, for steering left and right, the steady part of the separating force included: conservative for the usual mean stress sensitivity. Free-length buckling uses Euler load and the model's end-restraint factors; review its reported span and state. Ball joints, seated yokes (at their pinion contact) and rigid bushings hold the bar laterally; a bushing on a spring, and a yoke lifted off its seat (its spring, on its stop its spring and stop), holds it only when stiff enough, a bushing in its softer plane, to brace the compression beside it ($k \geq 4 \cdot P/L$, P the larger compression and L the shorter span next to it), else its spans join over it, in each state and load case on its own. A strength peak factor above 1 is a state of its own, the static peak: the rack solved again at the factor times its loads, meshes and weight (a steering state's rack friction included, reported as that friction times the factor),

so its yokes and sprung bushings act as they do at the peak. Its static strength and buckling checks read that state; a peak that cannot be solved (the bar lifted off its pinion, say) leaves the rack, its meshes and the pinions they drive unrated. Compression no span covers is reported as a warning, and its buckling is not computed. A yoke that comes against its stop, in any state the rack is rated in (the state, the static peak, the misuse states and each load case's), is listed with the force its stop carries: not a warning, since at a high separating force the plug normally comes against its adjusting screw, and the rack and the mesh are rated with the stop. The stop itself (the plug, its adjusting screw and the housing) is not checked: its force is listed under Not checked, and the verdict is no plain pass until you check those parts for it by hand.

Friction is a force estimate from support reaction and entered coefficients plus seal force. It does not model speed-dependent friction, stick-slip, lubrication or thermal losses, and is not deducted from the train's power balance. Tie-rod ends are point-force boundaries, not an elastic tie-rod model. No torsional, rotor-dynamic or FKM rack assessment is supplied. Travel endpoints check geometry; a duty spectrum and tooth-by-tooth stroke counting are not computed here. Verify the manufactured tooth geometry and application-specific fatigue inputs separately.

Rating method and its range **(help: modules/rack-mesh.md)**

Contact and root ratings use ISO 6336-2/-3:2019 Method B with the wheel taken to infinite tooth count. The stiffness uses the same infinite-tooth device as ISO 6336-1:2019, 9.3.2.6 b). The dynamic factor K_v of method B is held to at least 1 and, outside the main resonance range, to at most 2 (ISO 6336-1:2019, 6.3 and 6.2.6), as for a gear pair; a mesh loaded at 100 N/mm or more whose K_v is set to 2 is warned. The rack's form and stress-correction factors come from a straight flank and circular fillet, with the critical section at the 30° tangent. No rack-specific standard rating is claimed.

ISO 6336's validated range is normal pressure angle 15°–25°, helix angle up to 30° and transverse contact ratio 1.0–2.5. A mesh outside those ranges still rates and carries an **outside ISO 6336's validated range** warning. Below transverse contact ratio 1, the root load is at the tip and the transverse load-distribution factor is 1. The rating cannot establish uninterrupted engagement when the total contact ratio is below 1.

Both members get a finite-element root comparison when their geometry can be meshed. The rack FE contains three straight teeth with circular fillets, held 3.5 modules below the root plane; the pinion uses its virtual spur tooth. The FE finds the tensile peak instead of fixing a 30° section. Outside the validated range, **FE Y_FS is shown first**, followed by ISO Y_F·Y_S. An unavailable FE result is stated, and is never replaced with a pass. These are normal-section models, not a three-dimensional contact analysis.

No published rack worked example is held. Validation rests on gear-limit convergence and hand-calculation tests for contact ratio, lift, forces, rack root factors and Annex B Y_M, plus a rack FE comparison. A 10,000,000-tooth gear wheel converges within a relative 0.00001 for the tested spur mesh (z_1 20) and helical meshes (z_1 18, x_1 0.3, β 15°; z_1 12, x_1 0.5, β 20°; z_1 17, x_1 0.1, β 10°), including the rack's Y_F, Y_S, ZD, root and contact stresses. The rack's ISO-style accuracy deviations default to the pinion's grade values because ISO 1328 has no rack tolerance table; enter measured deviations when known.

Known limits

- Where a train's speeds come from its items and two of them turn one shaft opposite ways (a warning at the later one's sense, with its fix): - the pair the train took as driven the other way round is rated so meanwhile, and its mesh loss with it (the Two-stage helical gearbox with G2 alone reversed in its file: total 3 614 W against 3 610 W), though the power in and the efficiencies are not given; - a drive or pair added or mounted beside a train that turns its shaft negatively starts at its default sense, positive, and so gets the warning until it is reversed (the fix) rather than taking the train's sense; - a rack mesh's sense reversed is not carried on to the rest of its train (a gear pair's, worm pair's or drive's is), only warned of.
- A teeth end (`teethEnd`) is rated by DIN 743-2's notch factors of a shaft with involute teeth without a hub (4.2.2) and an eq. (9) shoulder or groove of its shape, not by a published method for a hob's run-out, which no source at hand gives (see "Shaft strength references"); the mesh's own load on the teeth is the rack mesh's or gear pair's tooth-root rating. A worm's thread end takes none (a strength notch instead). The Shaft module's 3D draws a rack pinion's teeth cut into the shaft; a gear pair's pinion cut into its shaft is drawn there at its root diameter as before (the Assembly and Steering views draw its teeth). The 3D views draw a single toothed section's ends; teeth over several sections at the root diameter are rated over all of them but drawn on the one at the mesh.
- The same switch is refused where the rack mesh takes its torque from the train and the rack back-drives a pinion mounted to drive it (the train's power flows from the rack): the state would take the mounting's sense. Set the steering system by hand with its state's rack force and rack speed signed as the rack is loaded and moves (a later release).
- The full-round limit of the basic rack's root radius is checked for cylindrical pairs, not yet for a rack mesh's own rack.
- With gravity on, a shaft placed by a bearing in another shaft (seated in it) is refused when the two are solved together and its frame is turned by what places that shaft (a universal joint, a worm pair, a rack mesh): the model of shafts solved together leaves the bearings between them out, and with them that placement (0.3.0). Place the shaft by another connection or at coordinates, or switch gravity off.
- A STEP shaft import into a rack is refused: its teeth (the toothed section's flat) are not read from the file.
- Where a rack mesh is given its pinion torque and no efficiency (its own rating then takes 0.92), the steering duty rates its force tables at the default for a known rack force, 0.85: a stroke the duty card starts at the mesh's rated rack force is rated 0.92/0.85, 8 %, above the mesh's own driving load, and linked items whose loads follow the rack's actual force (a ball screw) are scaled 8 % above it too. On the safe side; exact where the mesh's efficiency is entered.
- Over steering-derived load cases a shaft's FKM damage check (with a number of cycles) is undecided where it passes when the shaft's loads follow the speed (0.3.0): every shaft of a mounted stage (a gear or worm pair, a rack mesh, a drive), whose mesh efficiency is rated at each case's speed and enters a member's torque, and whose belt's pull rises with the square of the speed, and every shaft joined to one by a coupling or a bearing between shafts. The cases' chord split bounds that damage only where the stresses follow the force alone. In belt-drive with SH1's FKM over 800000 cycles and one-way strokes over a 126–130 mm/rev lead, the cases passed 6.681 at 6.681179 where the strokes give 6.680947.

- A steering duty whose strokes reverse the rack force (a return pass against a resisting load, a signed load below zero) is refused at `steering.duty` beside anything the load cases reach that a case cannot reverse (0.3.0): a mounted gear or worm pair or rack mesh, a drive, a coupling, a rotor, a bearing with loads of its own, a gear pair rated over the cases. The derived cases reverse only the entered shaft loads and ball screw forces, so the others would be rated at the forward loads on the reversed passes (a rotor's torque, unreversed, cancelled the reversed input in one review's case: 18.444302 where 9.222151 is right). In gearbox-single-stage with a rack mesh and resisting strokes with return passes, BR1 passed 5.5 at 5.778100 where the reversed pass gives 5.260948. One-way strokes are rated; reversing stages over the duty come in a later release.
- A steering block (strokes or misuse events) beside a rack member or a rack mesh mounted on one, without a steering system, is refused at `steering` (0.3.0): without a system the duty rates standalone rack meshes. Its derived load cases would load the rack member, whose yokes change regime and whose spring braces hold or give way with the force along the strokes, and the cases do not bound those states between them; and it would rate a mounted mesh's teeth without the rack's lift at the contact. The `steering-rack` template with a stroke now gives that error and derives no load cases. Give the project a steering system (the duty is then rated over its balance), or rate the duty in a copy whose rack meshes stand alone (the `steering-duty` template).

- A steering duty over a steering system (0.3.0) rates the rack meshes' teeth, and with strokes the system's shafts, the bearings linked on them that take the load cases, its worm pairs and its ball nut over its states (see Steering system). Not rated over the duty (a later release): a mounted gear pair (rated at the system's `state`, its checks not rated over the duty, never a pass: its rating takes its solved shafts' KHβ, which the states do not give it yet), and likewise fits, keys and housings. A rotor's air gap is not rated over the states (its band between them is not drawn; never a pass: rate it at the duty's heaviest state). A bearing's warning over the states (a minimum load undecided at its valley) is never satisfied, and its gyroscopic slip (the high-speed model's) is the system's state's, a pass not rated over the duty. Beside such a duty a design search's metrics are refused but the shaft's mass (their values over the states have no bound between them, and some items' results are the system's state's): search with requireChecks, whose checks take the bands. The ball nut's life is the states' Miner sum (as a bearing's), its static safety the worse of the states' and the system's own state's. Left at the system's `state`: the couplings, a bearing with `useLoadCases` false, and everything at a misuse event (the events rate the pinions only; the elements keep their own `peakFactor` for a static peak, which over the duty multiplies every state's loads, as over load cases). A worm pair and a drive take the worst of their states, each rated as if it lasted their life: on the safe side, as DIN 3996 holds no load spectrum here. A shaft's states are each rated with their stresses fully reversed (as a mounted ball screw's states are), on the safe side where a stroke does not turn round. The band of a bearing's checks is drawn from its life, static safety, contact stress and thermal speed in the states, relative to each check's utilisation. Two supports in seats of one housing (whose seats push each other with the bearings' loads) leave the elements unrated over the duty (a warning at `steering.duty`, their checks never a pass): the states do not couple the seats again (a later release). The project's own load cases beside such a duty rate the shafts and bearings instead (a warning at `steering.duty`), and their checks are then not rated over the duty (never a pass): remove the load cases to rate them over it. The elements' states are some four times the duty points (284 for the Dual-pinion EPS's duty), each a state solve and a solve of every shaft: at most 1200 (`system_cases::CASE_LIMIT`) and at most 12,000 solves (`WORK_LIMIT`: each state one, and one for each shaft and each bearing on a shaft; some 25 s natively and 40 s in the browser), past which the elements are not rated over the duty (a warning at `steering.duty`; their checks never pass) while the teeth are. Measured under load on the build machine (one process, natively): the Dual-pinion EPS's duty 9.3 s for the teeth and 19 s in all; the Belt-driven ball-nut and Column EPS examples given its duty 2.4 s and 2.8 s for the teeth, 7.0 s and 5.2 s in all (168 and 88 states); the browser's engine took about the same under that load (17.9 s, 7.6 s and 6.3 s). It refuses a stroke's `torque` table (the system splits the rack force, not a pinion's torque); a rack mesh off the system's rack is not rated over the duty (a warning; it keeps its rating at its own load). Each duty point is a state solve (some 15 to 30 ms natively), at most 4000 per duty. Rating the teeth of two pinions is the slow part: the Dual-pinion EPS template's duty (a parking, a manoeuvring and an on-centre stroke, both ways) takes some 4.7 s natively and 7.3 s in the browser's engine on the build machine (some 20 s natively on a laptop, 4 s of it the state solves); the app computes it on request, with progress and Cancel (0.3.0), and solves each edit without it. Positions past the rack's travel are refused (each point is a state of the system, within its stops). The lift is rated within 1e-4 mm of the states' between points (the contact ratios within some 4e-5), not raised; the forces are raised to bound theirs. Misuse events are rated at standstill (vehicle speed 0, forward efficiencies, no friction).

- A rack mesh's lead field (the variable-ratio table) reads a bare number in the unit shown, mm/rev or in/rev, but refuses a lead typed in the other one ("60 mm/rev" with US units shown): the field parser has no kind for a lead yet. Type it in the unit shown. The parser change comes in a later release.

Bolted joints: VDI 2230

Highly stressed bolted joints by VDI 2230 Part 1, and bolt property classes by ISO 898-1.

Methods and standards

Bolted joints: VDI 2230-1 (2015), single-bolt joints, steps R0–R13.

Bolt resilience as cylinders in series (Eq. 18–31): head 0.5·d (hexagon) or 0.4·d (socket head), shank sections, free loaded thread and engaged thread on the minor diameter (less the bore of a hollow bolt), nut 0.4·d or tapped thread 0.33·d with the tapped part's modulus; bending resilience alike (Eq. 33–37).

Plate resilience (Eq. 39–53): cone angle from Eq. 42 (tapped) or 43 (through bolt) with y = D′_A/d_W, cone(s) up to D_A and a sleeve beyond, a sleeve alone when D_A ≤ d_W, split at plate faces so each part takes its plate's modulus.

Load factor Φn = n·(δP + δPZu)/(δS + δP) with δPZu = δM for tapped joints (Eq. 80–83); n given or interpolated in Table 2 from the joint type SV 1–6, a_k/h and l_A/h.

Eccentric clamping and loading: IBers from the cone frusta (Eq. 59, 60, 69, 70) and the prismatic sleeve b·cT³/12 (Eq. 61–63), δP* = δP + s_sym²·Σ l_i/(EPᵢ·I_i), δP** = δP + a·s_sym·Σ l_i/(EPᵢ·I_i) (Eq. 67–72), Φen* = n·(δP** + δPZu)/(δS + δP*) (R3/4), clamp load against opening FKA (R2/3) and the opening load FAab (Eq. 104), bending stress in the bolt σSAb (Eq. 186–189).

External working moment MB (with the interface geometry, also for concentric clamping): Φm* = n·s_sym²·Σ l_i/(EPᵢ·I_i)/(δS + δP*) (Eq. 97, 98), additional bolt load Φm*·MB/s_sym (Eq. 5; none when s_sym = 0, Table 3), MB terms in FKA (R2/3) and FAab (Eq. 104), the opening moment MBab (Eq. 105), bolt bending MSb = βP/βS·[(a − s_sym·Φen*)·FA + MB·(1 − s_sym/|s_sym|·Φm*)] (Eq. 184) in σSAb = FSA/AS + MSb/WS with M_B,max and M_B,min; a moment that would raise the clamp load or relieve the bolt is not credited.

Working stress by Eq. 150 (bending neglected, βS ≫ βP), or by Eq. 149 with MSb/Wb (Eq. 155) on request, then also in FV1 (Eq. 157, 158).

Partial opening (F_A,max above FAab at the least preload, no working moment): FSA by the circular arc of Annex D between Φen*·FA and the lever line of edge bearing (v + a)/(v + s_sym)·FA − FV (Eq. D1–D7), used for FSmax, surface pressure and, as FSAKl/FA in place of Φen*, the alternating stress.

Embedding from Table 5 (axial or shear columns, head and nut faces, inner interfaces including the one at a tapped part).

Thermal preload change ΔF′Vth = (lK·α_S·ΔT_S − Σ l_i·α_P,i·ΔT_P)/(δS·ESRT/E_ST + δP,T) (R4/2) with the moduli at the operating temperature when given (δP,T from the deformation body with each plate's E_PT; else constant moduli), eq. 117's preload-dependent term noted but not applied, a loss added to FMmin and taken from the residual clamp load, a gain added to the bolt load.

F_Kerf = max(FKQ, FKP + FKA) with FKQ = F_Q/(q_F·μ_T) + M_Y/(q_M·r_a·μ_T) (R2/1–R2/4); FMmin = F_Kerf + (1 − Φ)·FA + FZ + ΔF′Vth, FMmax = α_A·FMmin.

FMzul at ν·Rp0.2 with the thread torsion on the plastic section modulus at the weakest section d_0 (stress diameter, reduced shank or bore; Eq. R7/2, 143, 144); torque MA with DKm = (d_W + D_Ki)/2 (R13/1).

Working stress with kτ = 0.5 (R8).

Yield- or angle-controlled tightening: α_A = 1, FMzul at Rp0.2, preload after the first loading FV1 with k_V (Eq. 163) and its least value with μ_G,max (Eq. 166), residual clamp load FKR1,min (Eq. 164), surface pressure 1.4·FMTab/Ap (R10/3).

Fatigue: σa = Φ·(F_A,max − F_A,min)/(2·AS), σ_ASV = 0.85·(150/d + 45), rolled after heat treatment σASG = (2 − FSm/F_0.2min)·σ_ASV, finite life (ND/N_Z)^(1/3) or ^(1/6) (R9).

Surface pressure max(FMzul, FMzul − FZ + FSA)/Ap under head and nut (R10).

Length of engagement: strength ratio RS (Eq. 199), least length by Eq. 209/213 with nominal thread dimensions, C1–C3 (Eq. 201–212), 1.2·R_m,min (Eq. 210) and 2·P supplement.

Slip safety FKR,min/FKQ (R12/1–R12/4), shearing τQ = F_Q/Aτ against (τB/R_m)·R_m from Table 7 (R12/5–R12/7); R_m is the minimum tensile strength R_m,min of ISO 898-1 Table 3 (VDI 2230 symbol list) unless given.

Validation

143 of 143 cases within their tolerance, in 3 groups.

Worked examples of standards and books

Bolted joints: VDI 2230 Annex B examples

Kind of reference	Worked examples of standards and books
Implements	VDI 2230 Part 1:2015
Reference source	VDI 2230 Part 1:2015, Annex B, examples B1 to B5
Cases	109 of 109 within tolerance

The five calculation examples of VDI 2230-1:2015, Annex B (B1 piston rod, B2 flange coupling, B3 hollow flywheel bolt, B4 connecting rod cap, B5 cylinder cover): resiliences, load factor, embedding, preloads, the clamp load against opening (the limit of the no-opening check), the additional bolt stresses σSAbo and σSAbu of the alternating stress, stresses, safeties and torque against the printed values. Tolerances are the printed rounding; where the example takes FMzul, MA or FMTab from the rounded Tables A1/A2, or has a slip in its arithmetic (B4 bending length, B5 δM), up to 1.5 %.

Closest to its tolerance: B4’s fatigue safety SD (2.0501 against the printed 2.02) uses 99 % of its 1.5 % tolerance. The example’s bending length lers of 65.98 mm does not follow from its own formula and dimensions (67.0 mm), which moves σab by 1.4 %; with the example’s rounded σASV (54.0 MPa) and SD that makes the 1.5 % the group’s note gives for B4.

Case	Reference	Axomek	Deviation	Tolerance	
B1: δS	2.95 10^{-6} mm/N	2.948424 10^{-6} mm/N	0.053 %	0.5 %	**pass**
B1: tan φD	0.566	0.5656278	0.066 %	0.2 %	**pass**
B1: DA,Gr	44.9 mm	44.87137 mm	0.064 %	0.2 %	**pass**
B1: δP	0.363 10^{-6} mm/N	0.3632692 10^{-6} mm/N	0.074 %	0.5 %	**pass**
B1: ΦK	0.161	0.1612679	0.17 %	0.5 %	**pass**
B1: n (Table 2)	0.35	0.346	1.1 %	1.5 %	**pass**
B1: Φn	0.056	0.05579868	0.36 %	1 %	**pass**
B1: fZ	8 µm	8 µm	0 %	10^{-9}	**pass**
B1: FZ	2415 N	2415.683 N	0.028 %	0.2 %	**pass**
B1: FMmin	26921 N	26926.3 N	0.02 %	0.2 %	**pass**
B1: FMmax	48458 N	48467.33 N	0.019 %	0.2 %	**pass**
B1: FMzul (Table A1)	64800 N	64902.32 N	0.16 %	1 %	**pass**
B1: FSmax	66194 N	66291.71 N	0.15 %	1 %	**pass**
B1: σred,B	819 MPa	820.5659 MPa	0.19 %	1 %	**pass**

Case	Reference	Axomek	Deviation	Tolerance	
B1: SF	1.15	1.145551	0.39 %	1 %	pass
B1: σa	8.3 MPa	8.244003 MPa	0.67 %	1 %	pass
B1: σASV	48.9 MPa	48.875 MPa	0.051 %	0.2 %	pass
B1: pM,max	720 MPa	720.9397 MPa	0.13 %	1 %	pass
B1: SP	1.8	1.803202	0.18 %	1 %	pass
B1: RS	0.93	0.9324635	0.26 %	0.6 %	pass
B1: MA (Table A1)	108 N·m	108.9271 N·m	0.86 %	1 %	pass
B2: δS	2.249 10^{-6} mm/N	2.248361 10^{-6} mm/N	0.028 %	0.2 %	pass
B2: tan φD	0.598	0.5978913	0.018 %	0.2 %	pass
B2: DA,Gr	58.4 mm	58.37348 mm	0.045 %	0.2 %	pass
B2: δP	0.781 10^{-6} mm/N	0.781065 10^{-6} mm/N	0.0083 %	0.2 %	pass
B2: fZ	11 µm	11 µm	0 %	10^{-9}	pass
B2: FZ	3630 N	3631.052 N	0.029 %	0.2 %	pass
B2: FKerf	56000 N	56000 N	0 %	10^{-6}	pass
B2: FMmin	59630 N	59631.05 N	0.0018 %	0.1 %	pass
B2: FMmax	95408 N	95409.68 N	0.0018 %	0.1 %	pass
B2: FMzul (Table A1)	118800 N	118930.5 N	0.11 %	0.5 %	pass
B2: pM,max	784 MPa	784.7588 MPa	0.097 %	0.5 %	pass
B2: SG	1.26	1.26251	0.2 %	0.5 %	pass
B2: τQ,max	41.8 MPa	41.77817 MPa	0.052 %	0.2 %	pass
B2: τB	620 MPa	620 MPa	0 %	10^{-6}	pass
B2: MA (Table A1)	302 N·m	303.1762 N·m	0.39 %	1 %	pass
B3: δS	0.80566 10^{-6} mm/N	0.8056958 10^{-6} mm/N	0.0044 %	0.1 %	pass
B3: tan φE	0.4712	0.4712353	0.0075 %	0.1 %	pass
B3: DA,Gr	51.08 mm	51.07953 mm	9.2×10^{-6}	0.1 %	pass
B3: δP	0.1055 10^{-6} mm/N	0.1055032 10^{-6} mm/N	0.003 %	0.3 %	pass
B3: fZ	12.5 µm	12.5 µm	0 %	10^{-9}	pass
B3: FZ	13700 N	13718.19 N	0.13 %	0.4 %	pass
B3: FKerf	56400 N	56410.26 N	0.018 %	0.1 %	pass
B3: FMmin	70100 N	70128.45 N	0.041 %	0.2 %	pass
B3: FMmax	112200 N	112205.5 N	0.0049 %	0.2 %	pass
B3: FMzul (hollow bolt)	140300 N	140515.5 N	0.15 %	0.2 %	pass
B3: pM,max	348.9 MPa	349.4335 MPa	0.15 %	0.2 %	pass
B3: SP	2	2.03186	1.6 %	2 %	pass
B3: FKR,min	74000 N	74104.02 N	0.14 %	0.2 %	pass
B3: SG	1.31	1.313662	0.28 %	0.5 %	pass
B3: RS	1.52	1.528162	0.54 %	0.6 %	pass
B3: MA	520 N·m	521.1422 N·m	0.22 %	0.3 %	pass
B4: δS	8.62 10^{-6} mm/N	8.62544 10^{-6} mm/N	0.063 %	0.2 %	pass
B4: G (R0/1)	32.3 mm	32.3 mm	0 %	0.2 %	pass

Case	Reference	Axomek	Deviation	Tolerance	
B4: tan φD	0.444	0.4446004	0.14 %	0.2 %	**pass**
B4: DA,Gr	32.3 mm	32.30702 mm	0.022 %	0.2 %	**pass**
B4: δP	1.351 10^{-6} mm/N	1.350337 10^{-6} mm/N	0.049 %	0.2 %	**pass**
B4: IBers	3157.8 mm^4	3160.261 mm^4	0.078 %	0.3 %	**pass**
B4: δP*	1.368 10^{-6} mm/N	1.367702 10^{-6} mm/N	0.022 %	0.2 %	**pass**
B4: δP**	1.685 10^{-6} mm/N	1.683745 10^{-6} mm/N	0.074 %	0.2 %	**pass**
B4: n (Table 2)	0.58	0.582	0.34 %	0.5 %	**pass**
B4: Φen*	0.098	0.09806123	0.062 %	0.5 %	**pass**
B4: FSA,max	490 N	490.3061 N	0.062 %	0.5 %	**pass**
B4: FKA	13465 N	13465.21 N	0.0016 %	0.1 %	**pass**
B4: FKerf	16267 N	16266.67 N	0.002 %	0.1 %	**pass**
B4: fZ	11 µm	11 µm	0 %	10^{-9}	**pass**
B4: FZ	1103 N	1102.671 N	0.03 %	0.2 %	**pass**
B4: FMmin	21880 N	21879.03 N	0.0044 %	0.1 %	**pass**
B4: FMzul = FMTab/ν	26444 N	26436.34 N	0.029 %	0.3 %	**pass**
B4: FV1	31467 N	31464.38 N	0.0083 %	0.3 %	**pass**
B4: FV1,min	22786 N	22904.91 N	0.52 %	1 %	**pass**
B4: FKR1,min	18276 N	18395.21 N	0.65 %	1 %	**pass**
B4: SG	1.12	1.130853	0.97 %	1.5 %	**pass**
B4: σSAbo	53.5 MPa	52.86361 MPa	1.2 %	1.5 %	**pass**
B4: σab	26.8 MPa	26.43181 MPa	1.4 %	1.5 %	**pass**
B4: σASV	54 MPa	54.1875 MPa	0.35 %	0.5 %	**pass**
B4: SD	2.02	2.050087	1.5 %	1.5 %	**pass**
B4: p_max head	603 MPa	603.3765 MPa	0.062 %	0.2 %	**pass**
B4: p_max nut	791 MPa	791.8472 MPa	0.11 %	0.2 %	**pass**
B4: SP (nut side)	1.25	1.250241	0.019 %	0.5 %	**pass**
B5: δS	1.157 10^{-6} mm/N	1.155476 10^{-6} mm/N	0.13 %	0.3 %	**pass**
B5: G (R0/2, 2·dW)	57 mm	57.74 mm	1.3 %	1.4 %	**pass**
B5: tan φE	0.564	0.5636241	0.067 %	0.2 %	**pass**
B5: DA,Gr	68.3 mm	68.32369 mm	0.035 %	0.2 %	**pass**
B5: δP	0.2458 10^{-6} mm/N	0.2453582 10^{-6} mm/N	0.18 %	0.2 %	**pass**
B5: IBers	112869 mm^4	112939 mm^4	0.062 %	0.3 %	**pass**
B5: δP*	0.25 10^{-6} mm/N	0.2497271 10^{-6} mm/N	0.11 %	0.3 %	**pass**
B5: δP**	0.184 10^{-6} mm/N	0.1836802 10^{-6} mm/N	0.17 %	0.3 %	**pass**
B5: n (Table 2)	0.28	0.2815429	0.55 %	1 %	**pass**
B5: Φen*	0.057	0.05733436	0.59 %	1 %	**pass**
B5: FKerf	68130 N	68125.94 N	0.006 %	0.1 %	**pass**
B5: FKab (no opening, U)	68130 N	68125.94 N	0.006 %	0.1 %	**pass**
B5: fZ	8 µm	8 µm	0 %	10^{-9}	**pass**
B5: FZ	5703 N	5710.882 N	0.14 %	0.3 %	**pass**

Case	Reference	Axomek	Deviation	Tolerance	
B5: FMmin	93193 N	93189.75 N	0.0035 %	0.2 %	pass
B5: FMmax	158428 N	158422.6 N	0.0034 %	0.2 %	pass
B5: FMzul (Table A1)	190000 N	190290.6 N	0.15 %	0.5 %	pass
B5: FSmax	191170 N	191467.6 N	0.16 %	0.5 %	pass
B5: σred,B	811.5 MPa	813.204 MPa	0.21 %	0.5 %	pass
B5: SF	1.16	1.155922	0.35 %	0.5 %	pass
B5: σSAbo	35.6 MPa	35.64991 MPa	0.14 %	1 %	pass
B5: σSAbu	10.7 MPa	10.69671 MPa	0.031 %	1 %	pass
B5: σab	12.5 MPa	12.4766 MPa	0.19 %	1 %	pass
B5: σASV	44.6 MPa	44.625 MPa	0.056 %	0.2 %	pass
B5: SD	3.56	3.576696	0.47 %	1 %	pass
B5: pM,max	692.2 MPa	693.2804 MPa	0.16 %	0.5 %	pass
B5: SP	1.03	1.024117	0.57 %	1 %	pass
B5: RS	0.61	0.6097531	0.04 %	0.5 %	pass
B5: MA (Table A1)	517 N·m	520.9337 N·m	0.76 %	1 %	pass

Standards’ tables and printed values

Bolted joints: VDI 2230 preload and torque

Kind of reference	Standards’ tables and printed values
Implements	VDI 2230 Part 1:2015 (assembly preload and tightening torque)
Reference source	VDI 2230 Part 1:2015 Tables A1/A2 values as tabulated by a fastener supplier (2025)
Cases	14 of 14 within tolerance

Permissible assembly preload FM and tightening torque MA at 90 % of Rp0.2, μG = μK = 0.12, metric coarse thread, against the VDI 2230:2015 values as a fastener supplier tabulates them (2025).

Case	Reference	Axomek	Deviation	Tolerance	
M8 8.8: FM	18.6 kN	18.62732 kN	0.15 %	1.5 %	pass
M8 8.8: MA	24.6 N·m	24.55624 N·m	0.18 %	3 %	pass
M10 8.8: FM	29.6 kN	29.6032 kN	0.011 %	1.5 %	pass
M10 8.8: MA	48 N·m	48.43646 N·m	0.91 %	3 %	pass
M12 8.8: FM	43 kN	43.10953 kN	0.25 %	1.5 %	pass
M12 8.8: MA	84 N·m	83.5932 N·m	0.48 %	3 %	pass
M16 8.8: FM	80.9 kN	80.97398 kN	0.091 %	1.5 %	pass
M16 8.8: MA	206 N·m	205.932 N·m	0.033 %	3 %	pass
M20 8.8: FM	130 kN	130.4757 kN	0.37 %	1.5 %	pass
M20 8.8: MA	415 N·m	415.5628 N·m	0.14 %	3 %	pass
M12 10.9: FM	63.2 kN	63.31713 kN	0.19 %	1.5 %	pass
M12 10.9: MA	123 N·m	122.7775 N·m	0.18 %	3 %	pass
M12 12.9: FM	74 kN	74.09451 kN	0.13 %	1.5 %	pass
M12 12.9: MA	144 N·m	143.6758 N·m	0.23 %	3 %	pass

Bolts: ISO 898-1 property classes

Kind of reference	Standards’ tables and printed values
Implements	ISO 898-1:2013
Reference source	DIN EN ISO 898-1:2013 Tables 4 and 5
Cases	20 of 20 within tolerance

Stress areas of the coarse threads (A_s from d_2 and d_3) and the minimum breaking loads F_m,min = A_s·R_m,min and proof loads F_p = A_s·S_p of the property classes against DIN EN ISO 898-1:2013 Tables 4 and 5, which give A_s,nom to three figures and the loads rounded to three figures.

Case	Reference	Axomek	Deviation	Tolerance	
M6: stress area As	20.1 mm²	20.12338 mm²	0.12 %	0.5 %	**pass**
M10: stress area As	58 mm²	57.9896 mm²	0.018 %	0.5 %	**pass**
M12: stress area As	84.3 mm²	84.26654 mm²	0.04 %	0.5 %	**pass**
M16: stress area As	157 mm²	156.6684 mm²	0.21 %	0.5 %	**pass**
M24: stress area As	353 mm²	352.5039 mm²	0.14 %	0.5 %	**pass**
M36: stress area As	817 mm²	816.7226 mm²	0.034 %	0.5 %	**pass**
M8 4.6: F_m,min	14600 N	14643.42 N	0.3 %	0.5 %	**pass**
M8 4.6: proof load Fp	8240 N	8236.922 N	0.037 %	0.5 %	**pass**
M10 6.8: F_m,min	34800 N	34793.76 N	0.018 %	0.5 %	**pass**
M10 6.8: proof load Fp	25500 N	25515.42 N	0.06 %	0.5 %	**pass**
M12 8.8: F_m,min	67400 N	67413.23 N	0.02 %	0.5 %	**pass**
M12 8.8: proof load Fp	48900 N	48874.59 N	0.052 %	0.5 %	**pass**
M20 8.8: F_m,min	203000 N	203179.4 N	0.088 %	0.5 %	**pass**
M20 8.8: proof load Fp	147000 N	146876.6 N	0.084 %	0.5 %	**pass**
M16 9.8: F_m,min	141000 N	141001.6 N	0.0011 %	0.5 %	**pass**
M16 9.8: proof load Fp	102000 N	101834.5 N	0.16 %	0.5 %	**pass**
M10 10.9: F_m,min	60300 N	60309.18 N	0.015 %	0.5 %	**pass**
M10 10.9: proof load Fp	48100 N	48131.37 N	0.065 %	0.5 %	**pass**
M20 12.9: F_m,min	299000 N	298649.2 N	0.12 %	0.5 %	**pass**
M20 12.9: proof load Fp	238000 N	237450.6 N	0.23 %	0.5 %	**pass**

Stated limits

Standards and scope **(help: modules/bolts.md)**

- VDI 2230-1:2015, single-bolt joints, steps R0–R13, with the resiliences of Section 5.1, the load introduction of Table 2, eccentric clamping and loading (Section 5.3), partial opening after Annex D, the embedding of Table 5 and tightening beyond the elastic limit. Property class strengths after ISO 898-1:2013 Table 3.
- Verified in `axomek verify` against the five calculation examples of Annex B (109 checks), tabulated preloads and torques (14 checks) and ISO 898-1 stress areas and loads (20 checks).

- R0, choosing the nominal diameter, is yours. The eccentric model holds for an interface up to G in the bending plane (G = d_W + h_min for a through bolt, 2·d_W for a tapped thread). Beyond it the checks that rest on the model (preload, yield, fatigue, surface pressure, slip and opening) are not determined where they pass, with the reason on each, and a fail stays a fail; define the interface by a recess, or check the joint numerically or by test.
- Not covered: the load distribution within a multi-bolt joint (VDI 2230-2), threads outside M4–M36 and non-metric threads, stainless (ISO 3506) and non-ferrous bolts, whose strengths are not in the property class list, and the preload-dependent thermal term of Eq. 117 (noted but not applied).

What is not covered **(help: theory/vdi-2230.md)**

- Multi-bolt joints are rated one bolt at a time: give each bolt its share of the loads.
- Relaxation and creep beyond the embedding of Table 5, and the preload-dependent temperature term of Eq. 117 (named in a note, not applied).
- Hydraulic, torsion-free tightening (5.4.4).

`axomek verify` recomputes the five calculation examples of VDI 2230-1 Annex B.

Known limits

- Units (R-387, R-931): the app and axomek read the same units typed, with the same factors, and show the same US customary units; a test holds them to it (`units-parity.spec.ts`), also for a field shown in another unit than its value's. It lists what still differs, on purpose or refused (never read wrongly): a design search's bound on a deflection is typed in µm (mil) in the app, as its check shows it, and in mm (in) by axomek `optimize --require`, which refuses nm there; a bolt's expansion coefficient α_S is typed in 10^{-6}/K in the app, in 1/K in axomek. A bare number in a bearing clearance is in µm in the app (the unit it shows) and in mm in axomek without `--units imperial` (base units, as every bare argument; mil with it, as in the app, but mm again with `--json`, which always reads base units). The bolt's expansion field refuses its own unit typed out (`12 10⁻⁶/K` reads as two numbers): type `12e-6 1/K`, or a bare 12 with metric units shown (with US units shown a bare number is in 10^{-6}/°F). The text output and the report keep a few units metric that the app shows in US units (m, J, GPa, √MPa, mm³, a steering's feel: SCREEN_ONLY in `crates/axm-model/src/units.rs`). Not compared: fields the app shows in % of a plain fraction (a torque amplitude, a modulus share), and the change lists of the Assistant and `axomek diff`, which give a clearance in mm.

- A result not determined is hidden display by display, not by one shared rule: in 0.3.0 the displays reviewed (the report, Show the calculation, the CLI's text, the comparisons, the Overview, the cards, the modules' tiles and the Assistant's digest) say "not determined". Those found last are refused instead: the Changes dialog's and Compare projects' check rows and the Overview's key results of keys and bolts say "not determined (see the check)"; the assembly view gives no share of the limit for a check not determined or not computed; and in the report an element's results, and a key's or a fit's results its checks go by (safeties, capacities, permissible values, pressures, shear, the fit's Turning row; with `axomek solve`'s key and fit lines), say "not determined (see the checks)" as soon as one of its checks is not determined, determined ones with them; a key's page refuses its flank pressures and tooth shear stress the same way (its permissible values and capacity, which do not depend on its rating, stay). A bolted joint with a check not determined (beyond VDI 2230's limit G, a duty not computed) has its whole Results block refused the same way in the report's section and in `axomek solve`'s line (its tightening torque, permissible assembly preload, resiliences and embedding with it, which the bolt page still shows); its page refuses its preload needed, load factor, clamp loads, stresses, opening and the joint diagram, but the notes under them, on its page and under the report's section, still quote the eccentric model's figures (M_Sb/W_b, F_Sm/F_0.2min, F_K against F_Kab). Make it pass and `axomek fix` give such a check no value either. A module's page, once one of an item's checks is not determined, shows no tile's caption (it may quote another check's figure) and refuses its tiles and Details tied to no check (a worm pair's load value C / Clim), and the Overview refuses the item's tiles tied to no check the same way, including ones that do not depend on the rating: a worm pair's method, centre distance and ratio, a motor's power and magnetic stiffness read "not determined" too; a later release shows those again. A design search meets no requirement or objective on a result of a design with any check not determined (a requirement's own check may be cleared, and the system life and power loss take every item's), its dimensions apart, nor on a fit's yield safety resting on an estimated yield strength, even where that figure itself would stand: the safe side; a later release decides each figure on its own. A bearing whose life the engine puts outside its rating method (κ below 0.1 where it turns, balls in four-point contact, a hybrid, whose a_ISO is defined for steel only) has all its lives, basic and modified, meet no search requirement or objective, as a minimum, a maximum or an objective, its minimum life cleared or not, and so has the system life with such a member; it says "not determined" without a life check too. The safe side: a hybrid's basic life, or one at a low κ, would stand; in a later release a search requirement on a figure whose own check is cleared is evaluated as that check, each life on its own. A design with a bearing the engine marks outside what it rates (a designation that contradicts it, a slewing ring's proportions) takes no calculated result in a search, its dimensions apart but that bearing's own (inferred from its geometry as modelled), as a design with a check not determined does, whatever its own checks say. A bearing whose designation contradicts it shows only its results' warnings it can vouch for: those at its own path, of the bearing as modelled, are left out (a validation warning at that same path too, such as that its geometry is estimated), its fields' warnings stay. A check not determined is said in a short form wherever its message would be rewritten: what it checks, named from its id, "not determined" and its kind's reason in fixed words, else "(see the check)", never a part of its message, which can hold a name the user wrote and the figures after it; a name that holds an engine mark's words can still pick another kind's reason (no figures). The full engine message, with the reason in context, comes back in a later release, when every display shares one rule. The bearing module's speed chart is not drawn while the bearing's operating nθ is not determined (a rotor's pull, a designation its geometry contradicts), saying why; `axomek speed-limits` then lists the curves' samples without an

operating point. A later release draws the curves apart from that load. A display not yet reviewed may still give such a number beside its check, which says "not determined". A later release routes every display of a check's value through one rule, with a test over every template. Known in 0.3.0, closed in a later release by that one shared display rule: - the Assistant's solve data still carries a check's value and utilisation where the check is not determined (a bearing's thermal speed whose nθ is withheld: its speed and share of the limit, from which the withheld nθ can be worked out); the check is flagged not determined there, the solve tool tells the Assistant never to quote a figure so flagged, and the check's row and every display say "not determined", never a pass; - a rack pinion shaft not rated at its mesh's peak because its only drive is solved for one power direction (the Dual-pinion EPS example's SH5, driven by worm pair WG1) is said, in the verdict's and the report's short reason, to be refused because its torque sources cannot be shared; its check's own note gives the right cause; - a design search takes no result from a design with any check not determined, so a project whose steering duty leaves a warning over its states may find no design meeting a shaft or bearing requirement (the safe side; a later release decides each figure on its own); - `Life0utside::against_bounds` keeps a maximum branch no caller uses (the search reads the hybrid flag directly); a later release removes it; - the Overview's governing check of a module, between two checks not determined, keeps the first in the list (the bearing listed first) where the engine picks by id, so the module tile and the verdict can name different checks, both said not determined; - a variant comparison can mark as best the life, and its check's margin, of a bearing the engine marks unrated (its designation contradicts its geometry), though the bearing's own results say its figures are of the bearing as modelled: compare such a design only after its designation is cleared or corrected; - a comparison of three or more designs can leave out a result row whose neighbouring columns each agree within the tolerance while the first and last do not (10 000, 10 000.8 and 10 001.6 h); `--all` shows it; - a comparison of designs whose life checks all pass on a hybrid's basic life (its modified life outside its rating method, so no best) leaves that check row out of the default CLI comparison and the report, though the minimum lives and the checks' shares differ (a hybrid 6309 at a 1 million h minimum against 4 million h); `--all` shows it. - a hybrid whose modified life fails its minimum keeps that failing check's number in its check row (the fail stands, on the safe side; the number rests on a_ISO, defined for steel bearings only), and so does any life check failing outside its rating method (a steel bearing turning below κ = 0.1, or with its balls in four-point contact): the report's Summary of checks and its Summary page's Governing checks show its share and draw its utilisation bar like any other (a 30207 on 2 / 1 cSt oil against a 20 000 h minimum), though that share cannot be compared with another check's; the governing pick and the orders leave it unranked, and the typed results' report follow-up shows no bar for it. Show the calculation of a hybrid's life explains the modified life, said not determined, not the basic life its passing check goes by; - the Assistant's shortened record of an earlier solve lists a failing life check with its value (a hybrid QJ 207 in four-point contact: 94 476 h against a 1 000 000 h minimum) but without the check's outside-method flag or reason, though the same record says the bearing's life is not determined; the fail stands, and the full solve the Assistant reads carries the flag; - over a duty cycle whose cases stay within the rating method, the Assistant's solve data still gives the nominal summary's modified life where the loads as entered put κ below 0.1 (the Overhung pulley's BR2 on 10 / 2.5 cSt oil at 150 °C nominal, 35 °C in its one case: 107 191 h), with no note that it is outside the method; the duty's life, which the check and every display go by, is right (a nominal summary in four-point contact is marked); - Show the calculation of a check that fails with an unbounded share says it fails "whatever the value of the checked quantity"; for an axial preload that lifts off (a cross-located pair whose housing runs hotter than its shaft) the share is of the lifted-off

state only, and the same check passes once the temperatures no longer lift it; the check's message gives the cause; - "Make it pass" on a check not determined still steers its search by the check's share as modelled (a proposal is made only where the check is determined and passes after the change), and its "tried" list may give that share as the closest reached; - the load-spectrum card's chart still plots a bearing's damage shares where its table withholds them (a case's life outside the rating method with the life check failing); read the table; - the report's summary page labels a passing hybrid's life "modified reference rating life" before the parenthesis that says it is the basic life its check goes by; the bearing section is right; - the agent guide says a hybrid's check `value` is its basic life; where the check fails (`outsideMethod`) it is the modified life it fails on; - a variant comparison of a steel and a hybrid bearing that both pass their life check leaves that row out of the default CLI comparison and the report, though the two columns go by different lives (the row's `labels` say which); `--all` shows it; - the Overview's utilisation chart plots the share of a check failing outside its rating method beside the others, ordered first among the failures but drawn as if comparable; the fail itself is shown; - a hybrid roller bearing on estimated geometry passes on the lower of its two methods' basic lives (correct), but the report's summary page names that life with the other method's label, the Assistant's basis text points at `basicLifeHours` (the picked method's, possibly the higher) instead of the check's value, and a check made not determined by the other method's lower basic life does not quote it; the bearing finder's `basicLifeHours` and the bearing page's basic life rows show the picked method's too; the check's value and status are right; A later release types every result figure as determined or not determined, and every display, ranking and short message reads that type: this closes the ranking, display and short-message limits above for good.

- Slewing rings are out of scope (user test ut4): the four-point contact type rates the standard series (QJ) as a rolling bearing with rigid rings; a slewing ring's rings are bolted and flexible, and it is rated by its maker's curves. The app and the help say so, and a four-point contact bearing of a slewing ring's proportions (D_pw/D_w above 15) has its checks not determined (from 0.3.0). A thin-section four-point bearing of such proportions is not determined either (the safe side).
- An eccentric bolted joint whose interface is larger than VDI 2230's limiting size G in the bending plane (R0, Eq. 54, 55: d_W + h_min for a through bolt, 2·d_W tapped) is not rated: the checks that rest on the eccentric model are not determined, with the reason and what to do (define the interface by a recess, or check the joint numerically or by test). A model of the wider interface (its non-linear pressure) is not planned; a bolt pattern's per-bolt interface is the user's to choose (RR1 U-A12, decision 2). For a tapped thread G is taken at the upper end of VDI 2230's 1.5 … 2·d_W, so an interface between the two reads as within it; and where a working moment presses V closed in both load states while the preload alone lifts it, the opening check fails on a pressure at V below 0 that does not show the assembly state's tension (Opus review of RR1 branch 13, P3s).

Keys and splines

Parallel keys by DIN 6892 method B and DIN 6885-1, and involute splines by DIN 5480 and ISO 4156.

Methods and standards

Parallel keys after DIN 6892:2012 method B: DIN 6885-1:1968 key for the shaft diameter (6–500 mm, with the keyway depths t_1 and t2, their tolerance and the standard length range), load-bearing depths by eqs. (4) and (5), t1tr = t_1 − (r + s_1) − (d − √(d² − (b + 2·s_1)²))/2 in the shaft and t2tr = h − t_1 − (r + s_2) + (d − √(d² − (b + 2·s_2)²))/2 in the hub (r the key's chamfer, DIN 6885-1's r_1 max unless given; s_1, s_2 the chamfers on the shaft's and the hub's keyway edges, 0 unless given, as DIN 6885-1 gives none; method C's 0.45·h is within about 15 % of these for DIN 6885-1 keyways), bearing length l − b (form A) or l (form B) by eq. (6), load share 0.75 for two keys, load distribution factor K_λ unless given from 6.1.2.4: Kλe of Figures 4, 5 or 6 for the load entering the hub at the front, centre or rear (Figure 3; front unless given), at the hub's Q_A = d/D (a stepped hub's D by eq. (14)) and l_tr/d, read at 14 values of Q_A from 0.35 to the figure's end (0.875 or 0.89) and interpolated linearly in both, the edge values outside them, Kλe(2)·(l_tr/d)/2 beyond l_tr/d 2, and without the hub's diameter the least favourable Q_A up to 0.6 (6.2.2: thinner hubs need a strength proof of their own, not made here); 2·Kλe − 1 for two keys; flank pressure (eqs. (9.1) and (9.2) without the friction factor K_R of an interference fit) against f_S·Re for shaft and hub, each on its depth, and Re for the key at the higher pressure.

Splines (DIN 5480, ISO 4156): DIN 5480 geometry from the reference diameter d_B with the shaft profile shift x·m = (d_B − m·z − 1.1·m)/2, tip and root diameters (h_fP = 0.55·m) and form diameters after DIN 5480-1 Tables 3–4, dimensions over and between pins and across k teeth from the involute as tabulated in DIN 5480-2; DIN 5480 mean flank diameter (d_a1 + d_a2)/2 = d_B − 1.1·m and flank overlap 0.9·m; ISO 4156 flank overlap (D_ee max − D_ii min)/2 and mean flank diameter (D_ee max + D_ii min)/2 from ISO 4156-1:2021 Table 1 (H/h fit), 75 % of the teeth carrying, tooth shear at the pitch circle against Re/√3.

Static (peak torque) checks; fretting and fatigue under reversing torque are not assessed.

Validation

82 of 82 cases within their tolerance, in 5 groups.

Worked examples of standards and books

Parallel key by DIN 6892 method B: load-bearing depths, bearing length, stepped hub, load distribution and surface pressures

Kind of reference	Worked examples of standards and books
Implements	DIN 6892:2012 method B: eqs. (4) and (5) (load-bearing depths with the chamfers r, s_1, s_2), (6) (bearing length), (14) (stepped hub), 6.1.2.4 with Figures 5 and 6 (K_λe), (9.1), (9.2), (10.1), (10.2) (surface pressures)
Reference source	DIN 6892:2012, Annex D: D.1 (elastic pin coupling, key A32 × 18 × 125 on Ø120, 4 000 N·m with K_A 1.5, peak 15 000 N·m, stepped hub D1 200, D2 270, c 17 mm, load entry rear), D.2 (servo gearbox output shaft, Ø60, key 18 × 11) and D.3 (gear shaft, Ø210, key 50 × 28)
Cases	14 of 14 within tolerance

D.1's hub is entered with its stepped diameters, as the example takes them (the pitch circle of the coupling's bolt holes for D1). D.1 and D.3 print K_λe 1.07 at l_tr/d 0.78 where the figures' curves give 1.087 (Figure 6, Q_A 0.58) and 1.097 (Figure 5, Q_A 0.37): the engine reads the figures, 1.6 % and 2.5 % above the examples (on the safe side), so these two readings are not compared and D.1's pressures are compared at the printed K_λ 1.07, given. D.2's K_λe (Figure 5 at Q_A 0.5 and l_tr/d 1.51) is compared to the figure's reading, half its 0.02 division. Not in the engine and not compared: the friction factor K_R of an interference fit (D.2, D.3), the load direction change factor f_W and the peak load factor f_L, the hardness factor f_H, a brittle hub's tensile strength, and the bearing ratio 0.9 of two keys at the peak torque; the engine checks one torque, so D.1's peak pressures are its pressures at M_tmax with K_A 1.

Case	Reference	Axomek	Deviation	Tolerance	
D.1: bearing length l_tr, eq. (6)	93 mm	93 mm	0 mm	exact	**pass**
D.1: load-bearing depth t1tr, eq. (4)	8.13 mm	8.127329 mm	0.002670821 mm	0.005 mm	**pass**
D.1: load-bearing depth t2tr, eq. (5)	7.9 mm	7.9 mm	1.7764×10^{-15} mm	0.005 mm	**pass**
D.1: stepped hub's cylinder D, eq. (14)	207 mm	206.9528 mm	0.04724594 mm	0.5 mm	**pass**
D.1: hub diameter ratio Q_A = d/D	0.58	0.5798425	0.0001575111	0.005	**pass**
D.1: p1eq, eq. (9.1), at K_λ 1.07	141.5 MPa	141.5641 MPa	0.06405001 MPa	0.1370234 MPa	**pass**
D.1: p2eq, eq. (9.2), at K_λ 1.07	145.6 MPa	145.6377 MPa	0.03767524 MPa	0.05 MPa	**pass**
D.1: p1max, eq. (10.1), at K_λ 1.07	353.8 MPa	353.9101 MPa	0.110125 MPa	0.2675892 MPa	**pass**
D.1: p2max, eq. (10.2), at K_λ 1.07	364.1 MPa	364.0942 MPa	0.005811896 MPa	0.05 MPa	**pass**
D.2: load-bearing depth t1tr, eq. (4)	3.7 mm	3.684271 mm	0.01572875 mm	0.05 mm	**pass**
D.2: load-bearing depth t2tr, eq. (5)	4.1 mm	4.115729 mm	0.01572875 mm	0.05 mm	**pass**
D.2: K_λe, Figure 5 (centre), Q_A 0.5, l_tr/d 1.52	1.32	1.328333	0.008333333	0.01	**pass**
D.3: load-bearing depth t1tr, eq. (4)	12.9 mm	12.88039 mm	0.01960973 mm	0.05 mm	**pass**

Case	Reference	Axomek	Deviation	Tolerance	
D.3: load-bearing depth t2tr, eq. (5)	12.3 mm	12.31906 mm	0.01905876 mm	0.05 mm	pass

Parallel key: hub flank pressure by methods B and C

Kind of reference	Worked examples of standards and books
Implements	DIN 6892:2012, methods B and C: p = 2·T·K_A·K_λ/(d·h′·l_tr·i·φ); DIN 6885-1:1968 (key A10 × 8)
Reference source	Roloff/Matek Maschinenelemente Aufgabensammlung, 15th ed. (2010), problem 12.1 (p. 80, solution p. 293): a saw blade's hub of EN-GJL-200 on a Ø35 mm E295 shaft, key DIN 6885 A10 × 8 × 32 of C45+C, 4 kW at 2850 rpm (13.4 N·m), K_A 1.5
Cases	2 of 2 within tolerance

The book takes h′ = 0.45·h = 3.6 mm in both methods; the engine takes DIN 6892's load-bearing depths (3.13 mm in the hub here, so its own hub pressure is 17.5 MPa, on the safe side). The pressures are therefore compared at the book's height: the engine's hub pressure times its depth over 3.6 mm. Method C takes K_λ = 1, method B the book's 1.05, both given: the book reads its own simplified chart (TB 12-2c), where DIN 6892's Figure 6 (rear) gives the engine 1.06 at Q_A 0.5, so K_λ itself is not compared here (DIN 6892 Annex D.2 is). Problem 12.2 is not compared: its printed 73.3 MPa does not follow from its own inputs (450 N·m on Ø70 with h′ 0.45·11 and l_tr 62 mm gives 41.9 MPa; 73.3 MPa needs an application factor of 1.75, which the problem does not give), and its b) reads the chart at l′/d 1.03 where 62/70 is 0.89: slips of the book.

Case	Reference	Axomek	Deviation	Tolerance	
12.1 b): hub flank pressure by method B (K_λ 1.05) at h′ = 0.45·h	15.2 MPa	15.22727 MPa	0.02727273 MPa	0.05 MPa	pass
12.1 a): hub flank pressure by method C (K_λ 1) at h′ = 0.45·h	14.5 MPa	14.50217 MPa	0.002164502 MPa	0.05 MPa	pass

Standards’ tables and printed values

Splines and keys: DIN 5480 and DIN 6885-1 dimensions

Kind of reference	Standards’ tables and printed values
Implements	DIN 5480-2:2006; DIN 6885-1:1968
Reference source	DIN 5480-2:2006 and DIN 6885-1:1968 dimension tables
Cases	43 of 43 within tolerance

DIN 5480 splines W d_B × m × z computed from the reference profile of DIN 5480-1:2006 (Tables 3, 4) against the nominal and inspection dimensions tabulated in DIN 5480-2:2006 (Tables 2, 3, 11, 20, 21: profile shift, tip, root and form diameters, dimensions over and between pins with the tabulated pin sizes, teeth spanned and base tangent length); DIN 6885-1:1968 parallel key rows by shaft diameter.

Case	Reference	Axomek	Deviation	Tolerance	
W 6 × 0.5 × 10: M1 over pins	7.511 mm	7.511274 mm	0.0036 %	0.01 %	pass
W 6 × 0.5 × 10: M2 between pins	3.797 mm	3.796827 mm	0.0046 %	0.02 %	pass
W 6 × 0.5 × 10: base tangent length Wk	3.859 mm	3.858625 mm	0.0097 %	0.02 %	pass
W 40 × 1 × 38: M1 over pins	42.032 mm	42.03163 mm	8.8×10^{-6}	0.01 %	pass
W 40 × 1 × 38: M2 between pins	36.37 mm	36.37013 mm	3.7×10^{-6}	0.02 %	pass
W 40 × 1 × 38: base tangent length Wk	19.903 mm	19.90345 mm	0.0023 %	0.02 %	pass

Case	Reference	Axomek	Deviation	Tolerance	
W 9 × 1 × 7: M1 over pins	13.347 mm	13.34726 mm	0.0019 %	0.01 %	**pass**
W 9 × 1 × 7: M2 between pins	5.189 mm	5.189362 mm	0.007 %	0.02 %	**pass**
W 9 × 1 × 7: base tangent length Wk	4.857 mm	4.8569 mm	0.0021 %	0.02 %	**pass**
W 50 × 2.5 × 18: M1 over pins	56.1 mm	56.10023 mm	4.1×10^{-6}	0.01 %	**pass**
W 50 × 2.5 × 18: M2 between pins	40.552 mm	40.55166 mm	8.4×10^{-6}	0.02 %	**pass**
W 50 × 2.5 × 18: base tangent length Wk	27.026 mm	27.02587 mm	4.7×10^{-6}	0.02 %	**pass**
W 92 × 2.5 × 35: M1 over pins	97.024 mm	97.02394 mm	6.0×10^{-7}	0.01 %	**pass**
W 92 × 2.5 × 35: M2 between pins	82.461 mm	82.46056 mm	5.3×10^{-6}	0.02 %	**pass**
W 92 × 2.5 × 35: base tangent length Wk	49.159 mm	49.1595 mm	0.001 %	0.02 %	**pass**
W 50 × 2.5 × 18: profile shift x·m	1.125 mm	1.125 mm	0 %	10^{-9}	**pass**
W 50 × 2.5 × 18: tooth thickness s	5.226 mm	5.226029 mm	5.5×10^{-6}	0.01 %	**pass**
W 50 × 2.5 × 18: shaft tip d_a1	49.5 mm	49.5 mm	0 %	10^{-9}	**pass**
W 50 × 2.5 × 18: shaft root df1	44.5 mm	44.5 mm	0 %	10^{-9}	**pass**
W 50 × 2.5 × 18: shaft form dFf1 max	44.93 mm	44.93 mm	0 %	10^{-9}	**pass**
W 50 × 2.5 × 18: hub tip d_a2	45 mm	45 mm	0 %	10^{-9}	**pass**
W 50 × 2.5 × 18: hub root df2	50 mm	50 mm	0 %	10^{-9}	**pass**
W 50 × 2.5 × 18: hub form dFf2 min	49.57 mm	49.57 mm	0 %	10^{-9}	**pass**
Key for Ø7: width b	2 mm	2 mm	0 %	exact	**pass**
Key for Ø7: height h	2 mm	2 mm	0 %	exact	**pass**
Key for Ø7: shaft depth t_1	1.2 mm	1.2 mm	0 %	exact	**pass**
Key for Ø7: hub depth t2	1 mm	1 mm	0 %	exact	**pass**
Key for Ø40: width b	12 mm	12 mm	0 %	exact	**pass**
Key for Ø40: height h	8 mm	8 mm	0 %	exact	**pass**
Key for Ø40: shaft depth t_1	5 mm	5 mm	0 %	exact	**pass**
Key for Ø40: hub depth t2	3.3 mm	3.3 mm	0 %	exact	**pass**
Key for Ø100: width b	28 mm	28 mm	0 %	exact	**pass**
Key for Ø100: height h	16 mm	16 mm	0 %	exact	**pass**
Key for Ø100: shaft depth t_1	10 mm	10 mm	0 %	exact	**pass**
Key for Ø100: hub depth t2	6.4 mm	6.4 mm	0 %	exact	**pass**
Key for Ø250: width b	56 mm	56 mm	0 %	exact	**pass**
Key for Ø250: height h	32 mm	32 mm	0 %	exact	**pass**
Key for Ø250: shaft depth t_1	20 mm	20 mm	0 %	exact	**pass**
Key for Ø250: hub depth t2	12.4 mm	12.4 mm	0 %	exact	**pass**
Key for Ø480: width b	100 mm	100 mm	0 %	exact	**pass**
Key for Ø480: height h	50 mm	50 mm	0 %	exact	**pass**
Key for Ø480: shaft depth t_1	31 mm	31 mm	0 %	exact	**pass**
Key for Ø480: hub depth t2	19.5 mm	19.5 mm	0 %	exact	**pass**

ISO 4156 splines: flank overlap and mean diameter

Kind of reference	Standards’ tables and printed values
Implements	ISO 4156-1:2021, Table 1 (H/h fit): h_w = (D_ee max − D_ii min)/2, d_m = (D_ee max + D_ii min)/2
Reference source	ISO 4156-2:2005, tables of major and minor diameters D_ee max and D_ii min (printed to 0.01 mm)
Cases	18 of 18 within tolerance

The height h_w and mean diameter d_m compared are half the difference and the mean of the printed diameters.

Case	Reference	Axomek	Deviation	Tolerance	
30°, m 0.5, z 6: h_w	0.39 mm	0.3923303 mm	0.002330317 mm	0.005000001 mm	**pass**
30°, m 0.5, z 6: d_m	3.11 mm	3.10767 mm	0.002330317 mm	0.005000001 mm	**pass**
30°, m 0.5, z 8: h_w	0.42 mm	0.4223611 mm	0.002361117 mm	0.005000001 mm	**pass**
30°, m 0.5, z 8: d_m	4.08 mm	4.077639 mm	0.002361117 mm	0.005000001 mm	**pass**
30°, m 2, z 6: h_w	1.57 mm	1.569321 mm	0.0006787323 mm	0.005000001 mm	**pass**
30°, m 2, z 6: d_m	12.43 mm	12.43068 mm	0.0006787323 mm	0.005000001 mm	**pass**
30°, m 2, z 8: h_w	1.69 mm	1.689445 mm	0.0005555339 mm	0.005000001 mm	**pass**
30°, m 2, z 8: d_m	16.31 mm	16.31056 mm	0.0005555339 mm	0.005000001 mm	**pass**
30°, m 2, z 12: h_w	1.8 mm	1.801818 mm	0.001818332 mm	0.005000001 mm	**pass**
30°, m 2, z 12: d_m	24.2 mm	24.19818 mm	0.001818332 mm	0.005000001 mm	**pass**
37.5°, m 1, z 6: h_w	0.795 mm	0.7973024 mm	0.002302426 mm	0.005000001 mm	**pass**
37.5°, m 1, z 6: d_m	6.105 mm	6.102698 mm	0.002302426 mm	0.005000001 mm	**pass**
37.5°, m 1, z 8: h_w	0.825 mm	0.8263279 mm	0.001327924 mm	0.005000001 mm	**pass**
37.5°, m 1, z 8: d_m	8.075 mm	8.073672 mm	0.001327924 mm	0.005000001 mm	**pass**
45°, m 1, z 6: h_w	0.75 mm	0.7504902 mm	0.0004902432 mm	0.005000001 mm	**pass**
45°, m 1, z 6: d_m	6.05 mm	6.04951 mm	0.0004902432 mm	0.005000001 mm	**pass**
45°, m 1, z 8: h_w	0.765 mm	0.7644661 mm	0.0005339059 mm	0.005000001 mm	**pass**
45°, m 1, z 8: d_m	8.035 mm	8.035534 mm	0.0005339059 mm	0.005000001 mm	**pass**

The standards’ formulae worked by hand

Parallel key: load-bearing heights and flank pressures

Kind of reference	The standards’ formulae worked by hand
Implements	DIN 6892:2012 method B: eqs. (4) and (5) (load-bearing depths), (6) (bearing length), (9.1) and (10.1) (pressures), 6.1.2.4 with Figure 4 (K_λe, load entry front); DIN 6885-1:1968 (key and keyway)
Reference source	Worked by hand for a Ø40 mm shaft, key 12 × 8 form A 50 mm, 300 N·m, hub diameter not given: depths from the keyway depth t_1 5 mm less the key's chamfer and the circle's drop
Cases	5 of 5 within tolerance

Without the hub's diameter K_λe is Figure 4's at Q_A 0.6, the most unfavourable hub DIN 6892 6.2.2 needs no strength proof for; method C's 0.45·h would give 3.6 mm in the hub.

Case	Reference	Axomek	Deviation	Tolerance	
Load distribution factor K_λe, front, Q_A 0.6 (hub not given), l_tr/d 0.95	1.1975	1.1975	0	1×10^{-12}	**pass**
Load-bearing depth in the shaft t1tr, eq. (4)	3.478784 mm	3.478784 mm	4.4409×10^{-16} mm	1×10^{-12} mm	**pass**
Load-bearing depth in the hub t2tr, eq. (5)	3.321216 mm	3.321216 mm	0 mm	1×10^{-12} mm	**pass**
Hub flank pressure 2·T·K_λ/(d·h′·l_tr)	142.3266 MPa	142.3266 MPa	0 MPa	1×10^{-9} MPa	**pass**
Shaft flank pressure	135.8801 MPa	135.8801 MPa	2.8422×10^{-14} MPa	1×10^{-9} MPa	**pass**

Not validated against a published reference

- DIN 6892’s K_λe: compared with Annex D.2’s reading only; D.1 and D.3 print 1.07 where the figures give 1.087 and 1.097, so their pressures are compared at the printed value
- DIN 6892’s friction factor K_R, load direction change factor f_W and peak load factor f_L, which the engine does not have
- Spline load capacity (flank pressure and shear), and the splines’ estimated K_λ

Stated limits

Standards and scope **(help: modules/keys.md)**

- Parallel keys after DIN 6892:2012 method B (flank pressure on the load-bearing depths of eqs. (4) and (5), K_λ from Figures 4 to 6 of 6.1.2.4), key and keyway dimensions after DIN 6885-1:1968 (form A and B). The worked examples of DIN 6892's Annex D and problem 12.1 of the Roloff/Matek problems book are validation cases. Not included: the friction factor K_R of an interference fit (the key is rated as if the fit carried nothing), the load direction change factor f_W and the peak load factor f_L (6.1.3), and the hub's own strength for a thin hub (d/D from 0.6). f_W and f_L lower the permissible pressure for a drive that reverses or peaks often, so every key's and spline's result notes that reversals and the number of starts or peaks are not assessed: check such a drive by hand. A rated joint lists them under what is not checked (and a thin hub's own strength where d/D is 0.6 or more), so its passing checks are no plain pass.

- DIN 5480 geometry from the reference profile of DIN 5480-1:2006 (form diameters after Tables 3 and 4) and the inspection dimensions computed from the involute as tabulated in DIN 5480-2:2006. `axomek verify` checks these dimensions and the DIN 6885-1 key rows against the printed tables ("Splines and keys: DIN 5480 and DIN 6885-1 dimensions", 43 checks).
- ISO 4156 splines are rated for flank pressure and shear; the dimensions card is computed for DIN 5480 only.
- Static checks at the peak torque K_A·T only. Fretting, wear and fatigue under reversing torque are not assessed. DIN 6892 method A, the shaft strength proof of DIN 6892 method B (the keyway as a notch is in the shaft strength check), taper keys, Woodruff keys and polygon profiles are not included.

Known limits

- The lint's number-reader rule (`axomek/number-reader`, R-388; a development check, nothing users see) finds `parseFloat` anywhere and `Number`, `parseInt` or `+` of an input's or text area's `value` read where it is used (any type assignable to `HTMLInputElement` or `HTMLTextAreaElement`, generics included). It does not see the value passed on first (a variable, a destructured `{ value }`, a function's argument), a key computed at run time, `Number.call`/`apply`, an element typed `any`, or Angular template expressions. Reviews catch those; read typed numbers with `core/format.ts`.

- A result not determined is hidden display by display, not by one shared rule: in 0.3.0 the displays reviewed (the report, Show the calculation, the CLI's text, the comparisons, the Overview, the cards, the modules' tiles and the Assistant's digest) say "not determined". Those found last are refused instead: the Changes dialog's and Compare projects' check rows and the Overview's key results of keys and bolts say "not determined (see the check)"; the assembly view gives no share of the limit for a check not determined or not computed; and in the report an element's results, and a key's or a fit's results its checks go by (safeties, capacities, permissible values, pressures, shear, the fit's Turning row; with `axomek solve`'s key and fit lines), say "not determined (see the checks)" as soon as one of its checks is not determined, determined ones with them; a key's page refuses its flank pressures and tooth shear stress the same way (its permissible values and capacity, which do not depend on its rating, stay). A bolted joint with a check not determined (beyond VDI 2230's limit G, a duty not computed) has its whole Results block refused the same way in the report's section and in `axomek solve`'s line (its tightening torque, permissible assembly preload, resiliences and embedding with it, which the bolt page still shows); its page refuses its preload needed, load factor, clamp loads, stresses, opening and the joint diagram, but the notes under them, on its page and under the report's section, still quote the eccentric model's figures (M_Sb/W_b, F_Sm/F_0.2min, F_K against F_Kab). Make it pass and `axomek fix` give such a check no value either. A module's page, once one of an item's checks is not determined, shows no tile's caption (it may quote another check's figure) and refuses its tiles and Details tied to no check (a worm pair's load value C / Clim), and the Overview refuses the item's tiles tied to no check the same way, including ones that do not depend on the rating: a worm pair's method, centre distance and ratio, a motor's power and magnetic stiffness read "not determined" too; a later release shows those again. A design search meets no requirement or objective on a result of a design with any check not determined (a requirement's own check may be cleared, and the system life and power loss take every item's), its dimensions apart, nor on a fit's yield safety resting on an estimated yield strength, even where that figure itself would stand: the safe side; a later release decides each figure on its own. A bearing whose life the engine puts outside its rating method (κ below 0.1 where it turns, balls in four-point contact, a hybrid, whose a_ISO is defined for steel only) has all its lives, basic and modified, meet no search requirement or objective, as a minimum, a maximum or an objective, its minimum life cleared or not, and so has the system life with such a member; it says "not determined" without a life check too. The safe side: a hybrid's basic life, or one at a low κ, would stand; in a later release a search requirement on a figure whose own check is cleared is evaluated as that check, each life on its own. A design with a bearing the engine marks outside what it rates (a designation that contradicts it, a slewing ring's proportions) takes no calculated result in a search, its dimensions apart but that bearing's own (inferred from its geometry as modelled), as a design with a check not determined does, whatever its own checks say. A bearing whose designation contradicts it shows only its results' warnings it can vouch for: those at its own path, of the bearing as modelled, are left out (a validation warning at that same path too, such as that its geometry is estimated), its fields' warnings stay. A check not determined is said in a short form wherever its message would be rewritten: what it checks, named from its id, "not determined" and its kind's reason in fixed words, else "(see the check)", never a part of its message, which can hold a name the user wrote and the figures after it; a name that holds an engine mark's words can still pick another kind's reason (no figures). The full engine message, with the reason in context, comes back in a later release, when every display shares one rule. The bearing module's speed chart is not drawn while the bearing's operating nθ is not determined (a rotor's pull, a designation its geometry contradicts), saying why; `axomek speed-limits` then lists the curves' samples without an

operating point. A later release draws the curves apart from that load. A display not yet reviewed may still give such a number beside its check, which says "not determined". A later release routes every display of a check's value through one rule, with a test over every template. Known in 0.3.0, closed in a later release by that one shared display rule: - the Assistant's solve data still carries a check's value and utilisation where the check is not determined (a bearing's thermal speed whose nθ is withheld: its speed and share of the limit, from which the withheld nθ can be worked out); the check is flagged not determined there, the solve tool tells the Assistant never to quote a figure so flagged, and the check's row and every display say "not determined", never a pass; - a rack pinion shaft not rated at its mesh's peak because its only drive is solved for one power direction (the Dual-pinion EPS example's SH5, driven by worm pair WG1) is said, in the verdict's and the report's short reason, to be refused because its torque sources cannot be shared; its check's own note gives the right cause; - a design search takes no result from a design with any check not determined, so a project whose steering duty leaves a warning over its states may find no design meeting a shaft or bearing requirement (the safe side; a later release decides each figure on its own); - `Life0utside::against_bounds` keeps a maximum branch no caller uses (the search reads the hybrid flag directly); a later release removes it; - the Overview's governing check of a module, between two checks not determined, keeps the first in the list (the bearing listed first) where the engine picks by id, so the module tile and the verdict can name different checks, both said not determined; - a variant comparison can mark as best the life, and its check's margin, of a bearing the engine marks unrated (its designation contradicts its geometry), though the bearing's own results say its figures are of the bearing as modelled: compare such a design only after its designation is cleared or corrected; - a comparison of three or more designs can leave out a result row whose neighbouring columns each agree within the tolerance while the first and last do not (10 000, 10 000.8 and 10 001.6 h); `--all` shows it; - a comparison of designs whose life checks all pass on a hybrid's basic life (its modified life outside its rating method, so no best) leaves that check row out of the default CLI comparison and the report, though the minimum lives and the checks' shares differ (a hybrid 6309 at a 1 million h minimum against 4 million h); `--all` shows it. - a hybrid whose modified life fails its minimum keeps that failing check's number in its check row (the fail stands, on the safe side; the number rests on a_ISO, defined for steel bearings only), and so does any life check failing outside its rating method (a steel bearing turning below κ = 0.1, or with its balls in four-point contact): the report's Summary of checks and its Summary page's Governing checks show its share and draw its utilisation bar like any other (a 30207 on 2 / 1 cSt oil against a 20 000 h minimum), though that share cannot be compared with another check's; the governing pick and the orders leave it unranked, and the typed results' report follow-up shows no bar for it. Show the calculation of a hybrid's life explains the modified life, said not determined, not the basic life its passing check goes by; - the Assistant's shortened record of an earlier solve lists a failing life check with its value (a hybrid QJ 207 in four-point contact: 94 476 h against a 1 000 000 h minimum) but without the check's outside-method flag or reason, though the same record says the bearing's life is not determined; the fail stands, and the full solve the Assistant reads carries the flag; - over a duty cycle whose cases stay within the rating method, the Assistant's solve data still gives the nominal summary's modified life where the loads as entered put κ below 0.1 (the Overhung pulley's BR2 on 10 / 2.5 cSt oil at 150 °C nominal, 35 °C in its one case: 107 191 h), with no note that it is outside the method; the duty's life, which the check and every display go by, is right (a nominal summary in four-point contact is marked); - Show the calculation of a check that fails with an unbounded share says it fails "whatever the value of the checked quantity"; for an axial preload that lifts off (a cross-located pair whose housing runs hotter than its shaft) the share is of the lifted-off

state only, and the same check passes once the temperatures no longer lift it; the check's message gives the cause; - "Make it pass" on a check not determined still steers its search by the check's share as modelled (a proposal is made only where the check is determined and passes after the change), and its "tried" list may give that share as the closest reached; - the load-spectrum card's chart still plots a bearing's damage shares where its table withholds them (a case's life outside the rating method with the life check failing); read the table; - the report's summary page labels a passing hybrid's life "modified reference rating life" before the parenthesis that says it is the basic life its check goes by; the bearing section is right; - the agent guide says a hybrid's check `value` is its basic life; where the check fails (`outsideMethod`) it is the modified life it fails on; - a variant comparison of a steel and a hybrid bearing that both pass their life check leaves that row out of the default CLI comparison and the report, though the two columns go by different lives (the row's `labels` say which); `--all` shows it; - the Overview's utilisation chart plots the share of a check failing outside its rating method beside the others, ordered first among the failures but drawn as if comparable; the fail itself is shown; - a hybrid roller bearing on estimated geometry passes on the lower of its two methods' basic lives (correct), but the report's summary page names that life with the other method's label, the Assistant's basis text points at `basicLifeHours` (the picked method's, possibly the higher) instead of the check's value, and a check made not determined by the other method's lower basic life does not quote it; the bearing finder's `basicLifeHours` and the bearing page's basic life rows show the picked method's too; the check's value and status are right; A later release types every result figure as determined or not determined, and every display, ranking and short message reads that type: this closes the ranking, display and short-message limits above for good.

- `axomek check --strict` and `axomek solve --strict` exit 3 for every file with something its method does not assess, which no input clears yet (4 where a check fails and the file was solved completely; a file that was not, an unknown field under `--strict` among it, exits 3 first): a rated key or spline (reversals and load peaks), a rated fit (its shaft seat), a belt or chain drive (its capacity), a fatigue-rated weld (its toes), a yoke on its stop, a support with a designation only and a planetary set without a planet bearing. Check those by hand and use `axomek check` without `--strict` for such files (it still exits 4 on a failing check and 3 on one not determined); a way to mark an item as checked by hand is for a later release.

- A test over every example and five more cases (a hybrid, a four-point contact bearing, a designation its geometry contradicts, the electric motor pulled off centre, a life below κ 0.1 failing beside a determined fail) replaces every figure the engine withholds and checks that no display shows it and no verdict or ranking moves with it (the "poison" harness). What it still finds is listed, each with its reason, in `crates/axm-cli/tests/poison/expected-fail.json`. Beside the limits above it found: the report's appendix of calculations gives a figure not determined as a number (a contradicted bearing's life, the pulled motor's critical speed); `axomek diff --json` and Compare projects give the check and result rows of such a life their numbers without an `outsideMethod` flag; the design search's `--json` gives a candidate's `objective` with no flag of its own where it is such a figure or one not determined (its `undetermined` names the metric; the search does not choose by it); `axomek find-bearing --json` lists a candidate's checks in an order its outside-method life's share moves (the candidates' order does not); and `axomek solve --json` gives `shaft.governingCriticalSpeed` with no flag where the check `minCriticalSpeed` is not determined (read that check's `undetermined`). A bearing's contact stress shows as a number where the static safety that goes by it is not determined: in the Overview's key results, `axomek solve`'s `p max` and the report's Max contact stress (read the static safety's "not determined" beside it). The app's checks panel picks µm or mm for the limit of a length check below 1 mm by the value as modelled, also where the check says "not determined" (no fixture has such a check yet). The harness itself does not tell a leak that moves to the same words on another line, a further difference inside a line whose wording already differs, or a leak moving to another item at the same place of a list from the one its table lists; nor, in the report's decisions, items whose ids are numbers alone ("1", "2").
- Keyboard and screen readers, for a later release (user tests ut3 and RR1): the app's screen reader checks read Windows' UI Automation from a real Edge window (`npm run smoke:screen-reader`, on demand, not in CI) and the browser's accessibility tree; no run with NVDA or JAWS themselves has been made, and a check by a person with one is still advised. A check's rule written as a formula in its name (a bearing's minimum load, "k_r 0.025·(ν·n/1000)^(2/3)·(d_m/100)²") is read out symbol by symbol. A press on a fix button while a list shows an option not yet applied applies the list first, and the click is lost: press again once the suggestions are solved. A chart's "Show the values as a table" button shows only with the keyboard focus on it or the pointer over the chart (always shown only where the main pointer is touch: on a touchscreen laptop whose main pointer is a trackpad it starts hidden), so voice control that numbers what is shown may skip it on a computer with a mouse. The edges between the columns have no single-pointer way to resize them other than dragging (the keyboard has its own; WCAG 2.2 2.5.7 asks for one), and their 24 px targets reach from the border into the drawing's and the results' left side, so a control within 24 px of that border is pressed by its edge instead. On a window too narrow for both chosen widths, a column's edge does not go wider than it is shown (narrow the other column first). With a part shown in the Shaft view's results column, the inputs' edge reckons with the results width chosen rather than the part's, so it may stop short of the room there is, and a drag on a window too narrow for both may change the results width kept. A refusal already there as the Shaft page opens (a shaft without supports solved on the Assembly page first) is in the page's status region from the start, so a screen reader need not say it on its own: it is read with the results.

- The keyboard-access check (`apps/desktop/scripts/keyboard-access.mjs`, quick CI) is a CI aid, not a proof: it reads what the templates and the components' metadata say, not what runs. It does not read metadata written in shorthand: a host's role and tabindex given as `host: { role, tabindex }` (the host then counts as no control, so Escape alone passes its click), or `{ templateUrl }` (that template is skipped). It looks a template constant up by its name in the file, so a constant of the same name in a narrower scope can be read instead. And it counts what an `<ng-template>` holds as shown with what is around it, so a list's keyboard owner there passes the list's items even when no outlet renders it. The app has none of these patterns today; the browser tests check real keys on the main paths.
- A placed key's keyway (see "From 0.3.0") is listed in the shaft's strength results at every point it is assessed, each mesh station along the key and the middle between two, which makes the list long for a shaft with long keys (the Drive shaft example's two keys give some 120 points); its lowest safety is the one the checks take. Between two sampled points the shaft's own weight may lift the moment by up to $q \cdot h^2/32$ (q its weight per length, h an element's length): a strength check with a placed keyway point whose safety that bound could take below its limit (through the lower of the bending amplitude strength and fatigue limit, the yield strength or, for a hard layer, its cracking criterion) is not determined (a warning, never a pass), saying so; on a shaft that does not rotate, where the missed stress is a mean stress whose effect is not bounded, a fatigue check with a placed keyway under the shaft's weight is never determined (take the key off the shaft and enter its keyway as a notch); refine the mesh (`settings.maxElementLength`) or enter a keyway notch where the moment along the key is largest. The exact search between the points comes in a later release.
- Bug reports: the inbox's source filter finds only reports that name a source, so those that arrived before 0.3.0 show under every source only. The reproduce check opens and solves the attached project only, and decides only from an error code (the report's) or an engine crash: it says "fixed" only when the code is gone, the report's command (or its last step) is a plain solve or validate of one named file (no pattern, no check, no variant, case or other option but the output's), and the run opened the project without warnings (none of newerFile, ids renamed), the reported code is the engine's (not a file's, the account's or the app's engine host's, such as io, signInRequired or native) and it computed everything (no solve error, nothing incomplete, no item's error); a wrong result reported in words alone, or another operation (an export, an edit), is "could not run", with the engine's output beside the report's expected and actual for the administrator to judge; it never says fixed on a guess. After a crash or freeze in one version and an update to another, no report is offered (it would go with the new version's number), nor when the log cannot tell which session ended badly (the run's mark names no session the log has, or more than one, as when two tabs or desktop windows ended at once; or a freeze kept while the page closed names none). On the web a run counts as ended badly only once its mark has had no beat for a minute, so a tab that crashed and was reloaded at once is offered its report at a later start. A freeze report leaves out the engine crashes the desktop app logged (they are the process's, not one window's); a freeze the browser or the system caused (a hidden or frozen tab, the computer asleep, a clock change) is not offered, so a real freeze that begins while the page is hidden, or overlaps sleep, is not offered either; and a web page restored from the browser's back-forward cache no longer marks its run, so a browser crash after that goes unnoticed. Both for a later release. App Check is not on yet (it needs a reCAPTCHA key and a privacy-text change; see App Check on the web above): until it is, the daily limits per install and per network address are the only protection of submitFeedback against abuse. The on-device log of the web app is one per browser profile: two tabs writing at once may drop the other's latest entries.

- A bearing whose eC·Cu/P stays within about 0.01 of 5 (where aISO leaves its cap of 50) along a whole stroke keeps the duty's search for that regime change refining: one such test case needs some 168 s natively per derivation, so it stops at the work limit and the bearing's checks over the strokes are undecided rather than split at the change. The app computes the duty on request, in a worker of its own with progress and Cancel (0.3.0). The work limits stay as they were: they are counted, not timed, so the app's duty equals `axomek solve`'s, and in the browser's engine (some 1.2 to 1.5 times the native time measured) the tooth rating's limit already takes about a minute and the derivation's some 3 to 4 min, past the minute the on-request duty aims for. Keying the FE root assessment by a rounded lead is not done: it would change the results of duties that finish today, and no bound puts the rounded lead's FE/ISO factor on the safe side. Such a bearing is not decided by a direct bracket either: along a part its eC·Cu/P follows the force one way and the speed (through the viscosity ratio) the other, so it can dip below 5 and back between two samples, which only a search for its least value finds, and that value is not provably single-troughed. Both stay for a later release.
- The app's computed steering duty belongs to the project object it was computed for: showing another shaft (which orders the project anew without editing it) makes it out of date, and a run under way is cancelled (0.3.0). On the safe side: Recompute duty, or go back to the shaft shown. Keying it by the project's content comes in a later release.

- A steering duty over a steering system (0.3.0) rates the rack meshes' teeth, and with strokes the system's shafts, the bearings linked on them that take the load cases, its worm pairs and its ball nut over its states (see Steering system). Not rated over the duty (a later release): a mounted gear pair (rated at the system's `state`, its checks not rated over the duty, never a pass: its rating takes its solved shafts' KHβ, which the states do not give it yet), and likewise fits, keys and housings. A rotor's air gap is not rated over the states (its band between them is not drawn; never a pass: rate it at the duty's heaviest state). A bearing's warning over the states (a minimum load undecided at its valley) is never satisfied, and its gyroscopic slip (the high-speed model's) is the system's state's, a pass not rated over the duty. Beside such a duty a design search's metrics are refused but the shaft's mass (their values over the states have no bound between them, and some items' results are the system's state's): search with requireChecks, whose checks take the bands. The ball nut's life is the states' Miner sum (as a bearing's), its static safety the worse of the states' and the system's own state's. Left at the system's `state`: the couplings, a bearing with `useLoadCases` false, and everything at a misuse event (the events rate the pinions only; the elements keep their own `peakFactor` for a static peak, which over the duty multiplies every state's loads, as over load cases). A worm pair and a drive take the worst of their states, each rated as if it lasted their life: on the safe side, as DIN 3996 holds no load spectrum here. A shaft's states are each rated with their stresses fully reversed (as a mounted ball screw's states are), on the safe side where a stroke does not turn round. The band of a bearing's checks is drawn from its life, static safety, contact stress and thermal speed in the states, relative to each check's utilisation. Two supports in seats of one housing (whose seats push each other with the bearings' loads) leave the elements unrated over the duty (a warning at `steering.duty`, their checks never a pass): the states do not couple the seats again (a later release). The project's own load cases beside such a duty rate the shafts and bearings instead (a warning at `steering.duty`), and their checks are then not rated over the duty (never a pass): remove the load cases to rate them over it. The elements' states are some four times the duty points (284 for the Dual-pinion EPS's duty), each a state solve and a solve of every shaft: at most 1200 (`system_cases::CASE_LIMIT`) and at most 12,000 solves (`WORK_LIMIT`: each state one, and one for each shaft and each bearing on a shaft; some 25 s natively and 40 s in the browser), past which the elements are not rated over the duty (a warning at `steering.duty`; their checks never pass) while the teeth are. Measured under load on the build machine (one process, natively): the Dual-pinion EPS's duty 9.3 s for the teeth and 19 s in all; the Belt-driven ball-nut and Column EPS examples given its duty 2.4 s and 2.8 s for the teeth, 7.0 s and 5.2 s in all (168 and 88 states); the browser's engine took about the same under that load (17.9 s, 7.6 s and 6.3 s). It refuses a stroke's `torque` table (the system splits the rack force, not a pinion's torque); a rack mesh off the system's rack is not rated over the duty (a warning; it keeps its rating at its own load). Each duty point is a state solve (some 15 to 30 ms natively), at most 4000 per duty. Rating the teeth of two pinions is the slow part: the Dual-pinion EPS template's duty (a parking, a manoeuvring and an on-centre stroke, both ways) takes some 4.7 s natively and 7.3 s in the browser's engine on the build machine (some 20 s natively on a laptop, 4 s of it the state solves); the app computes it on request, with progress and Cancel (0.3.0), and solves each edit without it. Positions past the rack's travel are refused (each point is a state of the system, within its stops). The lift is rated within 1e-4 mm of the states' between points (the contact ratios within some 4e-5), not raised; the forces are raised to bound theirs. Misuse events are rated at standstill (vehicle speed 0, forward efficiencies, no friction).
- A parallel key's keyway edge chamfers s1 and s2 (DIN 6892 eqs. (4) and (5)) are 0 unless entered: DIN 6885-1 gives none, the drawing does, and each lowers the depth the key bears by its size. Enter them from the drawing.

- A parallel key's K_λ is read from DIN 6892's Figures 4 to 6 at 14 values of Q_A and six of l_tr/d (within about 0.005 of the figures), not by the more accurate method the standard cites; its Annex D.1 and D.3 print 1.07 where the figures give 1.087 and 1.097, so the app's pressures there come out 1.6 % and 2.5 % above the examples'. A thin hub (d/D from 0.6), whose own strength DIN 6892 6.2.2 then asks to be proven, is noted, not assessed. The friction factor K_R of an interference fit is not in the app (the key is rated as if the fit carried nothing, on the safe side). Nor are the load direction change factor f_W and the peak load factor f_L (6.1.3), which lower the permissible pressure for a drive that reverses or peaks often: not on the safe side, so every key's and spline's result says that reversals and the number of starts or peaks are not assessed (from 0.3.0; user test RR1). Inputs for them are planned for a later release.
- A press fit's yield strengths are sized for steels only (DIN 743-2's K1 at the hub's outside diameter and the shaft's seat diameter): the cast irons and aluminium of the list keep their listed values, and a yield strength entered by hand is taken as the part's. A shaft larger elsewhere than at its seat is weaker than its seat diameter gives (DIN 743-2 takes its largest diameter), which only the shaft strength check takes. The size classes the app holds end at 100 mm for C45 +QT, 160 mm for the alloy steels and 250 mm for S235 and S355 (EN 10025-2:2019, Table 6, p. 27; its class over 250 up to 400 mm is for flat products only, Clause 1, and is not taken for a fit's round parts), and it holds none for E335 and the case-hardening steels' cores: beyond them a passing yield check is not determined until the guaranteed yield strength at that size is entered. The class is taken at the part's finished diameter, not at the size of the bar or forging it is turned from, whose class the steel's guarantee follows and which can be lower (an S355 hub of Ø40 from Ø45 bar: 335 MPa, where Ø40 takes 345; review of 0.3.2): for a part near its yield limit, enter the yield strength of the stock's size. E335's classes (Table 7) wait for an input of the stock's size for the same reason. A fit is not linked to a shaft, so its torque is not compared with the torque its seat passes (a placed key's is): a later release.
- The shaft strength check (DIN 743) takes DIN 743-2's K1 alone, as the standard prescribes where the strength at the part's size is not known (6.2): for an alloy steel inside a size class it can lie above the class's guaranteed minimum (42CrMo4 +QT at Ø50: 748.6 MPa against EN 10083-3's 650 MPa), as in any calculation after DIN 743. Enter the guaranteed strengths ("strengths as given for the size") for a shaft near its limit. Press fits hold it to the class (see "From 0.3.0"). Keys of form AB, C to J and lift-out bevels are not in the app, nor a keyway shorter than its key: the key's length stands for both, where DIN 6892 takes the shorter for l_tr and K_λ. A spline's K_λ is still the estimate of 1 up to a length of one diameter, + 0.4 per further diameter: DIN 6892 is for parallel keys only, and DIN 5466-1's factors are not taken.
- A key's and a spline's yield strength (shaft, hub and key) is the material's value at 16 mm, whatever the part's size, as in 0.2.7. A larger part's can be lower (the release-candidate test's keyhub: an S355 hub of Ø90, 355 → about 314 MPa by DIN 743-2's K1), and the safeties shown are then optimistic. For a part over 16 mm, enter the supplier's guaranteed yield strength at that size. The app's Materials lists hold the steels' 16 mm values and have no field for it, so set it in the file or with `echo '{"op":"updateKey","id":"K1","patch":{"hubMaterial":{"name":"S355 at 90 mm","yieldStrength":314}}}' | axomek apply <file> -` (piped, as PowerShell 5.1 drops the quotes of an argument; `shaftMaterial`, `keyMaterial` likewise), under a name of its own: a material that keeps a listed name is shown in the app with the list's Re, though the file's value is the one rated. Taking it at size, as press fits do, is for a later release.

- A key or spline added after another in 0.3.1 or earlier took that joint's hub outside diameter (and step), which a saved file cannot tell from one entered: check `hubDiameter` of keys added that way. From 0.3.2 a new joint starts without it (DIN 6892's least favourable hub until given); a seat reference that takes it from the shaft is for 0.3.7.
- Messages and refusals, for a later release (RR1 branch 22, what it left): - Validation errors from an element's own engine are placed at the field their message starts with, but about 75 of the project's validation errors still give no hint beyond their message ("must be positive"), and a JSON value of the wrong type in `axomek apply` is refused without its field's name (RR1 TA-F53, agent F5). - A power typed into a torque field is converted at the speed field's value at that moment, said only until it is applied; the torque then stays when the speed changes (RR1 U-A28). A key placed on a shaft keeps its own diameter and torque, and a keyed section can't be swept, optimised or proposed by `fix` (U-A27). "What matters most" can rate a screw's root diameter as not moving its speed check (U-A24, not traced). The planetary refusals and fixes U-C3 lists (a nearby set the search finds but the card does not, a ring shift offered before a planet change, the default planet count beside a typed ring), the tilt note shown above a failing minimum load, a warning's reason shown only on the Overview, "Driven by" offering stages that close a loop, and a single tapered roller bearing's raw "could not reach equilibrium" in the bearing finder. - The Assistant's results give a life in whole hours, as the app shows it, so a life just under its minimum (19 999.6 h against 20 000 h) reads as the minimum there; its check, failing, keeps the exact value. Where one case of a duty cycle puts a bearing outside its rating method (κ below 0.1, balls in four-point contact), the Assistant is told to quote none of its lives, a hybrid's basic life included, not even the modified life at the loads as entered where that one is within its method (the safe side). The room the app makes to keep a field in place is kept through a mouse click on a button in the panel, not yet through a tap on a touch screen (Opus review of RR1 branch 22). - A support's own Designation is shown as its linked bearing's and can't be typed over while a bearing is linked; the value saved in the file stays and comes back when the bearing is unlinked.
- US customary units change the display and the entry only: the bearings are metric (no inch-bore bearings, adapter sleeves, pillow blocks or insert bearings in the presets or the bearing finder), the steels go by their EN names, the keys by DIN 6885-1, and the shaft drawing is metric on an ISO sheet, which its dialog says with US units shown (its fits table gives inches beside the mm). The help (Units, Bearings) says so. Inch drawings, ANSI sheets and US parts are not in 0.3.0 (RR1 decision 5).

Helical springs: EN 13906

Cold coiled helical compression, extension and torsion springs of round wire by EN 13906-1, -2 and -3, with the largest wire diameter d_max of EN 10270-1 Table 3 and EN 10270-3 Table 5.

Methods and standards

Helical springs of round wire, cold coiled: compression springs after EN 13906-1, rate R = G·d⁴/(8·D³·n), shear stress τ = 8·D·F/(π·d³) uncorrected for static loads, against τzul = 0.5·R_m (a usual design value; EN 13906-1 limits it by the relaxation allowed) and at the solid length against τc,zul = 0.56·R_m (10.1.1), corrected τk = k·τ with Bergsträsser's k = (w + 0.5)/(w − 0.75) for fatigue; solid length n_t·d_max (ground ends, 9.10 eq. 15) or (n_t + 1.5)·d_max (eq. 16) with the largest wire d_max = d + its upper deviation of EN 10270-1:2024 Table 3 (coiled wire), for 1.4310 EN 10270-3:2011 Table 5 class T12, the widest for coiled wire (the standard names no default class), least gaps S_a = (0.0015·D²/d + 0.1·d)·n (1.5 times under cyclic load, cold coiled); buckling deflection s_K with the seating coefficient ν; natural frequency 3 560·d/(n·D²)·√(G/ρ); outside diameter growth 0.1·(m² − 0.8·m·d − 0.2·d²)/D with m = (s_c + n·d)/n (ground) or (s_c + (n + 1.5)·d)/n; E, G and ρ of EN 13906-1 Tables 3 and 4 (1.4310 heat treated) at room temperature.

Fatigue on a Goodman-type line from the fatigue strength τ_kO0 at τkU = 0 (estimated 0.23·R_m, 0.30·R_m shot peened, for 10^7 strokes, unless given) to 0.5·R_m, in place of the standard's diagrams.

The forces the spring must give are not an input and not checked: compare F_1 and F_2 (M_1 and M_2) with them; Make it pass and the design search may change them.

Extension springs after EN 13906-2 with the initial tension, τzul = 0.45·R_m; the initial stress τ_0 = 8·D·F_0/(π·d³) within the range that can be coiled, 231·e^(−0.105·w) ± 6.9·(4 − (w − 3)/6.5) MPa (the spring makers' preferred range, in place of the standard's diagram; up to w 20, not checked beyond); the hooks at F_2: bending in the loop about the spring axis σ_b = q_1·32·max(r_1, D/2)·F/(π·d³) + 4F/(π·d²) against 0.7·R_m and torsion in the bend into it τ = q_2·8·D·F/(π·d³) against 0.45·R_m, with Wahl's curved-bar factors q_1 = (4·c_1² − c_1 − 1)/(4·c_1(c_1 − 1)), q_2 = (4·c_2 − 1)/(4·c_2 − 4), c = 2r/d from the loop's and the bend's mean radii (German hook r_2 = d, English hook r_2 = max(D/4, d) unless given).

Torsion springs after EN 13906-3, R = E·d⁴/(3667·D·n_k) N·mm/° with the legs in n_k, σq = q·32·M/(π·d³) against 0.7·R_m, close-wound body (n_t + 1.5)·d_max unloaded and (n_t + 1.5 + α_2/360°)·d_max at α_2 (9.2.8 eqs. 18 and 19); cyclic torsion springs on a Goodman-type line from σ_qO0 at σqU = 0 (estimated 0.40·R_m, 0.50·R_m shot peened, unless given) to 0.7·R_m with the corrected stresses at α_1 and α_2.

Tensile strengths: for SL, SM, SH, DM and DH the minimum of EN 10270-1:2024 Table 3 in the diameter row lo < d ≤ hi that holds d, stepwise (DM as SM, DH as SH; straightened and cut lengths may be up to 10 % lower, its footnote c); for 1.4310 typical minimum values after EN 10270-3, interpolated by wire diameter and not checked against the standard.

Validation

59 of 59 cases within their tolerance, in 5 groups.

Worked examples of standards and books

Compression springs: rate, least gaps, stresses, correction factor and natural frequency

Kind of reference	Worked examples of standards and books
Implements	EN 13906-1:2013, eqs. (6) (rate), (7) (stress), (1) and (9) (correction k and corrected stress), 9.9 eq. (14a) (least gaps S_a, 1.5 times under cyclic load), 9.10 eq. (15) (solid length with d_max, EN 10270-1:2024 Table 3), 9.12 eq. (20) (natural frequency)
Reference source	Roloff/Matek Maschinenelemente Aufgabensammlung, 15th ed. (2010), problems 10.23 to 10.28 (results and solution pp. 281 to 283); Roloff/Matek Maschinenelemente, 21st ed. (2013), example 10.4 (pp. 344 to 346)
Cases	26 of 26 within tolerance

Each spring as the book lays it out (d, D, n, n_t, L_0, L_1, L_2; G 81 500 MPa); the tolerance is the book's printed precision. The book's S_a of example 10.4 is the static sum (its S′_a = 1.5·S_a is printed rounded up to 16.5 mm), so that case runs static. Solid lengths L_c = n_t·d_max (eq. 15) are compared where the book's d_max is EN 10270-1:2024 Table 3's (10.24, 10.27, example 10.4); 10.21, 10.23 and 10.26 take the problems book's older wire tolerances.

Case	Reference	Axomek	Deviation	Tolerance	
10.23: rate	20.3 N/mm	20.33528 N/mm	0.03528265 N/mm	0.05 N/mm	**pass**
10.24: rate	40 N/mm	39.92749 N/mm	0.07250637 N/mm	0.5 N/mm	**pass**
10.24: sum of least gaps S_a	7.8 mm	7.7525 mm	0.0475 mm	0.05 mm	**pass**
10.24: solid length L_c	55.3 mm	55.275 mm	0.025 mm	0.05 mm	**pass**
10.25: rate	177 N/mm	176.8663 N/mm	0.1336806 N/mm	0.5 N/mm	**pass**
10.25: sum of least gaps S_a	2.8 mm	2.79 mm	0.01 mm	0.05 mm	**pass**
10.25, alternative: rate	174 N/mm	174.063 N/mm	0.06298828 N/mm	0.5 N/mm	**pass**
10.26: rate	28.1 N/mm	28.13139 N/mm	0.03139205 N/mm	0.05 N/mm	**pass**
10.26: sum of least gaps S_a	4.1 mm	4.125 mm	0.025 mm	0.05 mm	**pass**
10.27: rate	24 N/mm	23.96608 N/mm	0.03391739 N/mm	0.5 N/mm	**pass**
10.27: least gaps under cyclic load S′_a	10.2 mm	10.17036 mm	0.02964375 mm	0.05 mm	**pass**
10.27: largest wire d_max	5.035 mm	5.035 mm	0 mm	0.0005 mm	**pass**
10.27: solid length L_c	52.9 mm	52.8675 mm	0.0325 mm	0.05 mm	**pass**
10.27: stress τ_2	417 MPa	416.7801 MPa	0.2198775 MPa	0.5 MPa	**pass**
10.27: correction factor k	1.225	1.225225	0.0002252252	0.0005	**pass**
10.27: corrected stress τk2	511 MPa	510.6495 MPa	0.3504806 MPa	0.5 MPa	**pass**
10.28: rate	19.6 N/mm	19.63671 N/mm	0.03670588 N/mm	0.05 N/mm	**pass**
10.28: least gaps under cyclic load S′_a	8.1 mm	8.088281 mm	0.01171875 mm	0.05 mm	**pass**
10.28: correction factor k	1.23	1.227273	0.002727273	0.005	**pass**
10.28: corrected stress τk2	808 MPa	807.8668 MPa	0.1331856 MPa	0.5 MPa	**pass**
10.28: natural frequency f_e	273 1/s	273.0299 1/s	0.02986734 1/s	0.5 1/s	**pass**
Example 10.4: rate	16.2 N/mm	16.20368 N/mm	0.003681818 N/mm	0.05 N/mm	**pass**

Case	Reference	Axomek	Deviation	Tolerance	
Example 10.4: sum of least gaps S_a	10.86 mm	10.8625 mm	0.0025 mm	0.005 mm	**pass**
Example 10.4: correction factor k	1.26	1.260116	0.0001156069	0.005	**pass**
Example 10.4: largest wire d_max	4.535 mm	4.535 mm	0 mm	0.0005 mm	**pass**
Example 10.4: solid length L_c	84 mm	83.8975 mm	0.1025 mm	0.5 mm	**pass**

Extension springs: rate with initial tension, forces and stresses

Kind of reference: Worked examples of standards and books

Implements: EN 13906-2, the rate $R = G \cdot d^4/(8 \cdot D^3 \cdot n)$, $F = F_0 + R \cdot s$ and $\tau = 8 \cdot D \cdot F/(\pi \cdot d^3)$

Reference source: Roloff/Matek Maschinenelemente Aufgabensammlung, 15th ed. (2010), problems 10.29 to 10.31 (p. 67, results p. 283); Roloff/Matek Maschinenelemente, 21st ed. (2013), example 10.5 (pp. 346 to 347)

Cases: 6 of 6 within tolerance

10.30 is a spring measured at two lengths (250 N at 180 mm, 550 N at 233 mm), from which the book derives $F_0 = 80$ N (printed to whole newtons): the engine's forces from F_0 80 N and its rate are compared with the measured ones within 1 N. In example 10.5 the book gives $F_2 \approx 785$ N with the largest permissible F_0 255 N instead of the spring's own 248.5 N (with which F_2 is 778.7 N): a slip of the book, not compared.

Case	Reference	Axomek	Deviation	Tolerance	
10.29: rate	20.7 N/mm	20.66514 N/mm	0.03485884 N/mm	0.05 N/mm	**pass**
10.30: force at 180 mm (measured)	250 N	249.9834 N	0.01661301 N	1 N	**pass**
10.30: force at 233 mm (measured)	550 N	550.2874 N	0.2873707 N	1 N	**pass**
10.30: stress τ_2	492 MPa	492.0872 MPa	0.08723231 MPa	0.5 MPa	**pass**
10.31: initial stress τ_0	108 MPa	107.5633 MPa	0.4367233 MPa	0.5 MPa	**pass**
Example 10.5: force F_1 at s_1 20 mm	400 N	399.9954 N	0.004645161 N	0.05 N	**pass**

Torsion springs: rate, moment, stress correction q, bending stress and body length

Kind of reference	Worked examples of standards and books
Implements	EN 13906-3:2014, eqs. (21) (active coils from the rate, 3667 = 64·180/π), q = (w + 0.07)/(w − 0.75), σq = q·32·M/(π·d³) and 9.2.8 eq. (18) (body length (n + 1.5)·d_max, close wound)
Reference source	Roloff/Matek Maschinenelemente Aufgabensammlung, 15th ed. (2010), problems 10.6 to 10.8 (solution pp. 277 to 278); Roloff/Matek Maschinenelemente, 21st ed. (2013), example 10.1 (pp. 341 to 342)
Cases	10 of 10 within tolerance

Legs left out, as the book does. The moment is checked with the book's own unrounded number of coils (24.1 and 18.83), which it works out for its moment at its angle; the tolerance is that number's printed precision. The book reads q to two decimals from its table TB 10-4, so its σq is uncertain by ±0.005 times the uncorrected stress, which the σq cases' tolerance adds to the printed precision. The body length L_K0 (eq. 18): the problems book takes the nominal d where EN 13906-3 takes d_max, so 10.7's tolerance adds (n + 1.5) times d's deviation (0.030 mm) to the printed whole millimetre; in 10.8 the deviation (0.010 mm) stays within the printed precision.

Case	Reference	Axomek	Deviation	Tolerance	
Example 10.1: moment at 120° with n 24.1	24000 N·mm	23986.28 N·mm	0.057 %	0.21 %	**pass**
Example 10.1: correction factor q	1.25	1.252308	0.002307692	0.005	**pass**
Example 10.1: bending stress σq	890 MPa	892.0311 MPa	2.031067 MPa	8.6 MPa	**pass**
10.7: moment at 180° with n 18.83	4000 N·mm	3999.108 N·mm	0.022 %	0.027 %	**pass**
10.7: correction factor q	1.12	1.123897	0.00389662	0.005	**pass**
10.7: bending stress σq	832 MPa	834.3319 MPa	2.331874 MPa	4.2 MPa	**pass**
10.7: body length L_K0 (n 18.75)	77 mm	77.5575 mm	0.5575 mm	1.1075 mm	**pass**
10.8: body length L_K0 (n 3)	3.2 mm	3.195 mm	0.005 mm	0.05 mm	**pass**
10.6: correction factor q	1.15	1.149091	0.0009090909	0.005	**pass**
10.8: correction factor q	1.08	1.076789	0.003210702	0.005	**pass**

The standards’ formulae worked by hand

Compression spring: rate, stresses, solid length and least working length

Kind of reference	The standards’ formulae worked by hand
Implements	EN 13906-1:2013, clause 9 (9.9 for the sum of least gaps); EN 10270-1:2024, Table 3
Reference source	The clause 9 formulae worked by hand: d 2.5 mm, D 20 mm, n 8.5 (10.5 total), ground ends, DH wire, L_0 60, L1 50, L2 35 mm, cyclic loading
Cases	10 of 10 within tolerance

Case	Reference	Axomek	Deviation	Tolerance	
Rate R = G·d⁴/(8·D³·n)	5.852194 N/mm	5.852194 N/mm	6.7×10^{-8}	10^{-6}	**pass**
Force F2 at L2	146.3049 N	146.3049 N	1.1×10^{-9}	10^{-6}	**pass**
Shear stress τ2 = 8·D·F2/(π·d³)	476.879 MPa	476.8797 MPa	1.5×10^{-6}	0.001 %	**pass**
Stress correction factor k	1.172414	1.172414	0 %	10^{-12}	**pass**
Corrected stress τk2	559.1 MPa	559.1003 MPa	6.1×10^{-7}	0.001 %	**pass**

Case	Reference	Axomek	Deviation	Tolerance	
Largest wire d_max = d + upper deviation (EN 10270-1 Table 3)	2.525 mm	2.525 mm	0 %	10^{-12}	**pass**
Solid length Lc = n_t·dmax	26.5125 mm	26.5125 mm	0 %	10^{-12}	**pass**
Sum of least gaps Sa (9.9, cyclic)	6.2475 mm	6.2475 mm	1.4×10^{-16}	10^{-12}	**pass**
Least working length Ln = Lc + Sa	32.76 mm	32.76 mm	0 %	10^{-12}	**pass**
R_m of DH wire at d 2.5 mm (EN 10270-1 Table 3)	1900 MPa	1900 MPa	0 %	exact	**pass**

Extension spring hooks and torsion spring rate and stress

Kind of reference	The standards’ formulae worked by hand
Implements	EN 13906-2:2001 (hooks, curved-bar factors); EN 13906-3:2014 (torsion springs)
Reference source	The formulae worked by hand on the compression spring's wire and coils: extension 60, 70, 90 mm with F_0 10 N and German hooks; torsion 30° and 90° with 20 mm legs
Cases	7 of 7 within tolerance

The hook stresses use Wahl's curved-bar factors q1 and q2 at the loop and the bend.

Case	Reference	Axomek	Deviation	Tolerance	
Torsion spring rate RM = E·d⁴/(3667·D·nk)	12.59384 N·mm/°	12.59408 N·mm/°	0.0019 %	0.005 %	**pass**
Torsion spring body LK = (n + 1.5)·dmax (EN 13906-3 eq. 18)	25.25 mm	25.25 mm	0 %	10^{-12}	**pass**
Torsion spring body at α_2 = (n + 1.5 + α_2/360)·dmax (eq. 19)	25.88125 mm	25.88125 mm	0 %	10^{-12}	**pass**
German hook: bending factor q1	1.102679	1.102679	0 %	10^{-12}	**pass**
German hook: torsion factor q2	1.75	1.75	0 %	10^{-12}	**pass**
German hook: bending stress σb at F2	1371.7 MPa	1371.715 MPa	0.0011 %	0.01 %	**pass**
German hook: torsion stress τh at F2	1058.5 MPa	1058.489 MPa	0.0011 %	0.1 %	**pass**

Not validated against a published reference

- The stainless wire’s solid length: no worked example of stainless wire is compared (its d_max takes EN 10270-3 Table 5’s class T12)
- Permissible stresses, buckling, the growth of the outside diameter, the initial tension that can be coiled and extension spring hooks: no worked example is compared
- Fatigue (a Goodman-type line in place of the standard’s diagrams)

Stated limits

Standards and scope **(help: modules/springs.md)**

- Compression springs after EN 13906-1:2013, the edition consulted for this page. Extension and torsion springs follow EN 13906-2 and EN 13906-3 as described in the method notes.

- Cold coiled springs of round wire only. Hot coiled, conical, barrel-shaped and disc springs, and wire of other cross-sections, are not covered.
- Static limits are the usual design values above. EN 13906-1 limits static stress by the relaxation allowed (10.2); relaxation is not computed or checked, and the spring's operating temperature is not an input. A spring held compressed, warm or near its static limit loses force over time: check it with the wire maker's relaxation data.
- Material values are at room temperature. The temperature dependence of G and E (8.2), transverse loading (9.13) and impact loading (9.15) are not applied.
- Fatigue uses a Goodman-type line instead of the standard's fatigue diagrams.
- The fatigue of extension springs is not assessed, because it depends on the hooks.
- The forces the spring must give are not an input and not checked: compare F_1 and F_2 (M_1 and M_2 for torsion springs) with them yourself. **Make it pass** lists how each proposal moves them under **Also**.
- The initial tension range is the spring makers' preferred range, in place of the standard's diagram. Wire strengths are EN 10270-1:2024's minima by diameter row (the grades SL to DH) and typical minimum values after EN 10270-3 (1.4310, interpolated by wire diameter and not checked against the standard).

Known limits

- An electric motor's critical speed and magnetic stability with a static eccentricity on coupled bearings (review of ut4-motor-critical-speed; see "From 0.3.0"). In 0.3.0: - A pull not along a bearing's other loads (more than 1° apart: a belt or a gear's force across the weight) leaves both not rated: the bearing's load turns as the eccentricity grows, and its stiffness along y or z can be least in between, which neither the states taken nor a bound with each bearing equally stiff every way at its least load bounded (a reviewer's case read 9 988 rpm where a state inside the bound gives 9 604 rpm). This blocks a belt- or gear-driven motor with an eccentricity from the coupled critical speed, and 0.3.0 names no way around it: no stiffness entered on the supports (with coupling off) is known to be safe, as the least one lies somewhere between centred and pulled (keep the eccentricity: its pull counts in the bearing loads and the air gap; 0 would drop it from them). - A state the coupling cannot settle exactly (bearings loose in their clearance, e.g. the Electric motor example without its wave spring, centred) leaves a passing result not decided (a warning). A motor whose bearings carry no load with the rotor centred (the weight off, a vertical motor without preload) has both not rated, with a hint (a horizontal motor: turn the weight on; a vertical one, or one held only by a preload, 0.3.0 cannot rate). The hints of the not decided and not determined results name no stiffness to enter: none is known to be safe there (review: a recipe of the bearings' stiffness without the pull could pass a motor that fails). - A coupled bearing under an axial load (a spring preload, as in the Electric motor example, or the shaft's own axial loads) or preloaded from within (an X or O pair's preload, a negative operating clearance) leaves a passing result not decided (a warning): its radial stiffness need not grow with its radial load, so the states taken bound nothing between them (a reviewer's case: e = 0.2 passed 15 000 rpm at 15 908 rpm where e = 0.15 gives 13 878 rpm). - A spring preload is refused on any shaft joined to another by a bearing between them (shafts solved together do not carry it), also where the spring would act on its own housing. - The unbalance response is the given eccentricity's, with a warning: at a smaller one it can be larger. - Up to four pulling rotors on a shaft; more have both not rated. The axial modes of a state keep the locating bearings' axial stiffness and the housings' seat motions of the pulled state. A critical speed or magnetic stability that is not rated fails, shown as "not rated" in place of its placeholder value (0 rpm, a share of 100 %), and What matters most leaves it out, saying why. Planned for a later release: the rotor pulled within its bearings' clearance to where they hold it (it never sits centred there), in place of the centred state; and a sweep of the eccentricity with a proven bound between its steps, for a pull across the bearings' loads, for bearings under an axial load (a spring preload) and for the unbalance response.
- One-way bearings, for a later release (Opus review of ut4-bearings): a single angular contact ball bearing free to tilt (on a shaft) whose axial load along its facing is below $F_r \cdot \tan \alpha$ (as a floating bearing, or locating with next to no axial load) is rated with its ring tilted to balance it, as the solver finds it, and passes; a real one needs a second bearing or a preload to keep its rings together. Held square, or a tapered roller bearing, it is refused. A thrust ball or roller bearing whose axial load reverses between load cases is rated each way (no field says whether it is single- or double-direction). The axial direction check fails only a load against the facing. On a floating support the bearing finder keeps a one-way candidate's preset facing and does not turn it to face the support's spring preload (`springPreload`): facing against the spring by more than the check's allowance (5 % of $F_r \cdot \tan \alpha$), the candidate fails its axial direction check and is not offered as passing, though turned round it could pass, on the safe side (Opus review of the ut4-motor-critical-speed merge). Turn such a bearing round by hand (`facing`).

- A steering block (strokes or misuse events) beside a rack member or a rack mesh mounted on one, without a steering system, is refused at `steering` (0.3.0): without a system the duty rates standalone rack meshes. Its derived load cases would load the rack member, whose yokes change regime and whose spring braces hold or give way with the force along the strokes, and the cases do not bound those states between them; and it would rate a mounted mesh's teeth without the rack's lift at the contact. The `steering-rack` template with a stroke now gives that error and derives no load cases. Give the project a steering system (the duty is then rated over its balance), or rate the duty in a copy whose rack meshes stand alone (the `steering-duty` template).
- A spring of stainless wire (1.4310) takes d_max with EN 10270-3:2011 Table 5's class T12, the widest for coiled wire, as the standard names no class to assume: a wire ordered to a tighter class has a slightly shorter solid length. Its notes say so.
- A spring's relaxation (its loss of force over time, warm or near its static limit; EN 13906-1, 10.2) is not computed or checked, and its operating temperature is not an input: the materials are at room temperature. The springs help says so; for a later release (RR1 D, user tests springs and mechanism).

Welds: EN 1993-1-8 and EN 1993-1-9

Fillet and butt welds by EN 1993-1-8 (static) and EN 1993-1-9:2005 (fatigue).

Methods and standards

Weld groups after EN 1993-1-8: straight fillet, full-penetration or partial-penetration butt weld lines with their throat turned into the face plane (area, centroid, I_x, I_y, I_xy, polar moment I_p of the lines, own terms $l^3/12$); forces at the load point taken to the centroid; in the face plane $\tau = F/A$ plus $M_z \cdot r/I_p$, across it $\sigma = F_z/A$ plus bending about the principal axes; per line $\tau_\parallel$ along it and, for fillet welds, the stresses at right angles split on the 45° throat as $\sigma_\perp = (f_z \pm f_n)/\sqrt{2}$, $\tau_\perp = (f_z \mp f_n)/\sqrt{2}$ on the worse side (the side the attached part stands on is not modelled); directional method $\sqrt{(\sigma_\perp^2 + 3(\tau_\perp^2 + \tau_\parallel^2))} \le f_u/(\beta_w \cdot \gamma_{M2})$ and $\sigma_\perp \le 0.9 \cdot f_u/\gamma_{M2}$, simplified method $a \cdot |\text{stress}| \le a \cdot f_u/(\sqrt{3} \cdot \beta_w \cdot \gamma_{M2})$; full-penetration butt welds as the weaker part, $\sqrt{(\sigma_\perp^2 + 3(\tau_\perp^2 + \tau_\parallel^2))} \le f_y/\gamma_{M0}$; partial-penetration butt welds (4.7.2) with the effective throat the depth of penetration less 2 mm (unless proven by procedure tests), their throat in the attached plate ($\sigma_\perp$ normal to the face, $\tau_\perp$ across the line) and designed as fillet welds with deep penetration by the directional or simplified method. The group's fillet and partial-penetration welds are taken as one lap (the joint type is not known), its length L_j the largest distance between two of their ends, gaps included; beyond $150 \cdot a$ each weld's design resistance is times $\beta_{Lw.1} = 1.2 - 0.2 \cdot L_j/(150 \cdot a)$, beyond 1.7 m times $\beta_{Lw.2} = 1.1 - L_w/17$ (L_w in m, 0.6 to 1) if smaller (4.11).

Fatigue after EN 1993-1-9:2005 (EN 1993-1-9:2025 lowers the curves of detail categories below 71, the default 36 among them, beyond $5 \cdot 10^6$ cycles; not followed): stress ranges as multiples of the loads, $\Delta\sigma_w = \sqrt{(\Delta\sigma_\perp^2 + \Delta\tau_\perp^2)}$ and $\Delta\tau_w = \Delta\tau_\parallel$ on the throat of fillet and partial-penetration butt welds (Table 8.5 details 3 and 8: root failure 36*, shear 80, unless a line gives its own detail categories), S–N curves through $\Delta\sigma_C$ ($m = 3$ to $5 \cdot 10^6$, $m = 5$ to the cut-off at 10^8) and $\Delta\tau_C$ ($m = 5$, cut-off at 10^8), γ_{Ff} and γ_{Mf}, damage $D_\sigma + D_\tau$ by Palmgren–Miner at each of eleven points per line; the largest range of the blocks with cycles, without γ_{Ff}, against $1.5 \cdot f_y$ (normal: $\Delta\sigma_w$ on a fillet or partial-penetration throat, $\sigma_\perp$ and $\tau_\perp$ together as Table 8.5 detail 3 rates them) and $1.5 \cdot f_y/\sqrt{3}$ (shear), 8(1). Only the weld throat (or a full-penetration butt line's own category) is rated: toe failure of the connected plates (Table 8.5 details 1 and 2) is not checked, as the plate thicknesses are not inputs; check it by hand.

Steel strengths of EN 1993-1-1 Table 3.1 for $t \le 40$ mm.

Validation

61 of 61 cases within their tolerance, in 6 groups.

Worked examples of standards and books

Fillet welds of an eccentrically loaded gusset plate: directional and simplified method

Kind of reference	Worked examples of standards and books
Implements	EN 1993-1-8:2005, 4.5.3.2 (directional method, eqs. (4.1) and (4.2)) and 4.5.3.3 (simplified method, eq. (4.4)); Table 4.1 (β_w)
Reference source	Roloff/Matek Maschinenelemente, 21st ed. (2013), example 6.2 (pp. 183 to 184): a 10 mm gusset plate on a column flange, two fillet welds a 4 × 220 mm, S235, F = γ_Q·Q_K = 1.5·125 kN at 35° and e = 60 mm
Cases	11 of 11 within tolerance

The book concentrates each weld in its root line, as the engine does (two lines 10 mm apart). It rounds σ⊥ and τ∥ to whole MPa before it combines them into √(σ⊥² + 3·(τ⊥² + τ∥²)) = 325 MPa; carried through, that rounding is worth ±1.3 MPa, which that case's tolerance adds to the printed precision.

Case	Reference	Axomek	Deviation	Tolerance	
A_w = Σa·l_eff	1760 mm²	1760 mm²	0 mm²	0.5 mm²	**pass**
I_w = 2·a·l³/12	7.099×10^6 mm⁴	7.0987×10^6 mm⁴	333.3333 mm⁴	500 mm⁴	**pass**
M_Ed = V_Ed·e	9.215×10^6 N·mm	9.2155×10^6 N·mm	460.4983 N·mm	500 N·mm	**pass**
σ_w = N/A_w + M·y/I_w at the weld's end	204 MPa	203.907 MPa	0.09299848 MPa	0.5 MPa	**pass**
τ∥ = V/A_w	87 MPa	87.26762 MPa	0.2676184 MPa	0.5 MPa	**pass**
σ⊥ = τ⊥ = σ_w/√2	144 MPa	144.184 MPa	0.1840235 MPa	0.5 MPa	**pass**
√(σ⊥² + 3·(τ⊥² + τ∥²))	325 MPa	325.5811 MPa	0.5810839 MPa	1.8 MPa	**pass**
f_u/(β_w·γ_M2)	360 MPa	360 MPa	0 MPa	0.5 MPa	**pass**
0.9·f_u/γ_M2	259 MPa	259.2 MPa	0.2 MPa	0.5 MPa	**pass**
Simplified: f_vw,d = f_u/(√3·β_w·γ_M2)	208 MPa	207.8461 MPa	0.1539031 MPa	0.5 MPa	**pass**
Simplified: resultant stress σ_wv = √(σ_w² + τ∥²)	222 MPa	221.7965 MPa	0.203466 MPa	0.5 MPa	**pass**

Standards’ tables and printed values

Fatigue strength curves: constant amplitude fatigue limit and cut-off limit

Kind of reference	Standards’ tables and printed values
Implements	EN 1993-1-9:2005, 7.1 and Figures 7.1 and 7.2 (N_D $5 \cdot 10^6$ with m 3 then 5 to N_L 10^8 for normal stress, m 5 to N_L 10^8 for shear)
Reference source	EN 1993-1-9:2025, Table 8.1 (curves of Figure 8.2a, detail categories 71 to 160) and Table 8.2 (Figure 8.4), pp. 42 to 43
Cases	18 of 18 within tolerance

The 2025 edition keeps the 2005 curves the engine implements for welded details of category 71 and above and for shear; it gives details below 71 (Figure 8.2b) a new curve (N_D 10^7, Δσ_D $0.585 \cdot \Delta\sigma_C$, Δσ_L $0.369 \cdot \Delta\sigma_C$), which the engine, after the 2005 edition, does not follow, so those rows are not compared. The engine's Δσ_D is where its curve gives $5 \cdot 10^6$ cycles and its Δσ_L the lowest range with a finite life, both found by bisection on its curve. The table works its values with the factors 0.737, 0.405 and 0.457 printed to three decimals and rounds them to three figures: the tolerance is these two roundings together.

Case	Reference	Axomek	Deviation	Tolerance	
Δσ_C 160: Δσ_D	118 MPa	117.889 MPa	0.094 %	0.49 %	**pass**
Δσ_C 160: Δσ_L	64.8 MPa	64.75411 MPa	0.071 %	0.2 %	**pass**
Δσ_C 140: Δσ_D	103 MPa	103.1529 MPa	0.15 %	0.55 %	**pass**
Δσ_C 140: Δσ_L	56.7 MPa	56.65984 MPa	0.071 %	0.21 %	**pass**
Δσ_C 125: Δσ_D	92.1 MPa	92.10079 MPa	8.6×10^{-6}	0.12 %	**pass**
Δσ_C 125: Δσ_L	50.6 MPa	50.58915 MPa	0.021 %	0.22 %	**pass**
Δσ_C 112: Δσ_D	82.5 MPa	82.52231 MPa	0.027 %	0.13 %	**pass**
Δσ_C 112: Δσ_L	45.4 MPa	45.32787 MPa	0.16 %	0.23 %	**pass**
Δσ_C 100: Δσ_D	73.7 MPa	73.68063 MPa	0.026 %	0.14 %	**pass**
Δσ_C 100: Δσ_L	40.5 MPa	40.47132 MPa	0.071 %	0.25 %	**pass**
Δσ_C 90: Δσ_D	66.3 MPa	66.31257 MPa	0.019 %	0.14 %	**pass**
Δσ_C 90: Δσ_L	36.5 MPa	36.42419 MPa	0.21 %	0.26 %	**pass**
Δσ_C 80: Δσ_D	59 MPa	58.9445 MPa	0.094 %	0.15 %	**pass**
Δσ_C 80: Δσ_L	32.4 MPa	32.37705 MPa	0.071 %	0.28 %	**pass**
Δσ_C 71: Δσ_D	52.3 MPa	52.31325 MPa	0.025 %	0.16 %	**pass**
Δσ_C 71: Δσ_L	28.8 MPa	28.73464 MPa	0.23 %	0.3 %	**pass**
Δτ_C 100: Δτ_L	45.7 MPa	45.73051 MPa	0.067 %	0.22 %	**pass**
Δτ_C 80: Δτ_L	36.6 MPa	36.5844 MPa	0.043 %	0.25 %	**pass**

The standards’ formulae worked by hand

Long fillet welds: the reduction factors β_Lw.1 and β_Lw.2

Kind of reference	The standards’ formulae worked by hand
Implements	EN 1993-1-8:2005, 4.11, eqs. (4.9) and (4.10)
Reference source	The clause's formulas worked by hand for a strap lapped on a plate by two longitudinal fillet welds a 4 mm, S355, 1000 mm (β_Lw.1) and 100 mm (none) long, and for runs over 1.7 m (β_Lw.2)
Cases	8 of 8 within tolerance

The group does not say which kind of joint a weld is in, so every fillet or partial-penetration run longer than 150·a (or 1.7 m) takes the smaller factor that applies, on the safe side where 4.11(2) would spare it; the run is the line with the collinear lines joined to it end to end.

Case	Reference	Axomek	Deviation	Tolerance	
β_Lw.1 for L_j 1000 mm, a 4 mm	0.866667	0.8666667	3.8×10^{-7}	10^{-6}	**pass**
Lap 1000 mm > 150·a: utilisation over β_Lw.1	1.039073	1.039073	0 %	10^{-12}	**pass**
Simplified method: F_w,Rd times β_Lw.1	1.039073	1.039073	0 %	10^{-12}	**pass**
Intermittent lap 2 × (400 + 400) mm over 1000 mm: utilisation over β_Lw.1	1.039073	1.039073	2.1×10^{-16}	10^{-12}	**pass**
Transverse end welds 1000 mm apart: utilisation over β_Lw.1	1.036853	1.036853	0 %	10^{-12}	**pass**
Lap 100 mm < 150·a: full resistance	9.002525	9.002525	0 %	10^{-12}	**pass**
β_Lw.2 for L_w 3.4 m	0.9	0.9	1.2×10^{-16}	10^{-12}	**pass**
β_Lw.2 not below 0.6	0.6	0.6	0 %	10^{-12}	**pass**

Fillet weld groups: stresses in the throat and the directional method

Kind of reference	The standards’ formulae worked by hand
Implements	EN 1993-1-8:2005, 4.5.3.2 (directional method) and 4.5.3.3 (simplified method), Table 4.1; EN 1993-1-1:2005, Table 3.1
Reference source	Weld lines as lines of their throat, worked by hand: a single 100 mm line (a 5 mm) loaded along and across it, a 100 × 60 mm rectangle (a 4 mm) in torsion and two 80 mm flange welds (a 5 mm) in bending; S355, γ_M2 1.25
Cases	12 of 12 within tolerance

Case	Reference	Axomek	Deviation	Tolerance	
Longitudinal weld: τ∥ = F/(a·l)	20 MPa	20 MPa	0 %	10^{-12}	**pass**
Longitudinal weld: equivalent stress √3·τ∥	34.64102 MPa	34.64102 MPa	2.1×10^{-16}	10^{-12}	**pass**
S355: limit f_u/(β_w·γ_M2)	453.3333 MPa	453.3333 MPa	0 %	10^{-12}	**pass**
Transverse weld: σ⊥ = F/(√2·a·l)	14.14214 MPa	14.14214 MPa	1.3×10^{-16}	10^{-12}	**pass**
Transverse weld: τ⊥ = F/(√2·a·l)	14.14214 MPa	14.14214 MPa	1.3×10^{-16}	10^{-12}	**pass**
Transverse weld: equivalent stress √2·F/(a·l)	28.28427 MPa	28.28427 MPa	0 %	10^{-12}	**pass**
Transverse weld: σ⊥ = F/(√2·a·l)	14.14214 MPa	14.14214 MPa	1.3×10^{-16}	10^{-12}	**pass**
Transverse weld: τ⊥ = F/(√2·a·l)	14.14214 MPa	14.14214 MPa	1.3×10^{-16}	10^{-12}	**pass**

Case	Reference	Axomek	Deviation	Tolerance	
Transverse weld: equivalent stress √2·F/(a·l)	28.28427 MPa	28.28427 MPa	0 %	10^{-12}	pass
Flange welds in bending: Ix = 2·a·l·y²	2×10^{6} mm⁴	2×10^{6} mm⁴	0 %	10^{-12}	pass
Rectangle in torsion: polar moment a·(b + h)³/6	2.7307×10^{6} mm⁴	2.7307×10^{6} mm⁴	0 %	10^{-12}	pass
Rectangle in torsion: corner stress M·r/Ip	21.35358 MPa	21.35358 MPa	0 %	10^{-12}	pass

Weld fatigue: S–N curves and damage

Kind of reference: The standards’ formulae worked by hand

Implements: EN 1993-1-9:2005, 7.1, Figures 7.1 and 7.2 (m = 3 and 5, knee at 5·10⁶, cut-off at 10⁸); Palmgren–Miner damage sum

Reference source: The S–N curves worked by hand for detail categories 71 and 36 (normal) and 80 (shear)

Cases: 7 of 7 within tolerance

Case	Reference	Axomek	Deviation	Tolerance	
Category 36: damage of 2·10⁶ cycles at Δσc	1	1	0 %	10^{-12}	pass
Category 36: damage of 10⁶ cycles at 0.6·Δσc (m = 5)	0.07161757	0.07161757	1.2×10^{-15}	10^{-12}	pass
Category 71: cycles at Δσc	2×10^{6}	2×10^{6}	0 %	10^{-12}	pass
Category 71: cycles at the knee Δσc·0.4^(1/3)	5×10^{6}	5×10^{6}	1.9×10^{-16}	10^{-12}	pass
Category 71: cycles at the cut-off	1×10^{8}	1×10^{8}	6.0×10^{-16}	10^{-12}	pass
Category 71: cycles at 2·Δσc (m = 3)	250000	250000	0 %	10^{-12}	pass
Shear category 80: cycles at Δτc	2×10^{6}	2×10^{6}	0 %	10^{-12}	pass

Fatigue: the stress range limit 1.5·f_y and the cut-off margin

Kind of reference: The standards’ formulae worked by hand

Implements: EN 1993-1-9:2005, 8(1), eq. (8.1); 7.1, Figure 7.1 (Δσ_L)

Reference source: The clause's limits worked by hand for two full-penetration butt welds 10 × 100 mm in S355 and a longitudinal fillet weld; the cut-off margin of a 220 mm double fillet weld a 12 mm under bending, as a user test worked it by hand

Cases: 5 of 5 within tolerance

The ranges against 1.5·f_y are those of the blocks that count cycles, without γ_Ff, as (8.1) states them.

Case	Reference	Axomek	Deviation	Tolerance	
Δσ limit 1.5·f_y (S355)	532.5 MPa	532.5 MPa	0 %	10^{-12}	pass
Δτ limit 1.5·f_y/√3 (S355)	307.439 MPa	307.439 MPa	0 %	10^{-12}	pass
Δσ 600 MPa over 532.5 MPa	1.126761	1.126761	0 %	10^{-12}	pass
Δτ 300 MPa over 1.5·f_y/√3	0.9758033	0.9758033	0 %	10^{-12}	pass
Double fillet a 12: Δσ_w/(Δσ_L/γ_Mf)	0.9465209	0.9465209	0 %	10^{-9}	pass

Not validated against a published reference

- Butt and partial-penetration butt welds, and weld groups in torsion: no worked example is compared (the cases are the formulae worked by hand)
- Fatigue damage of a weld group: only the fatigue strength curves are compared, and not those of details below category 71, which EN 1993-1-9:2025 changed

Stated limits

Standards and scope **(help: modules/welds.md)**

- EN 1993-1-8:2005 4.5.3.2 (directional method), 4.5.3.3 (simplified method), 4.7.2 (partial-penetration butt welds) and 4.11 (long joints); full-penetration butt welds as the parent metal.
- A fillet weld the standard does not let carry load is refused where it is, by either method: a throat below 3 mm (4.5.2(2)), or an effective length below 30 mm or 6 times the throat (4.5.1(2)). Fillet lines joined end to end count as one weld for its length, so a curved weld drawn as short straight lines, or a weld run round a corner, is judged by its whole length.
- EN 1993-1-9:2005 nominal stress method with the detail categories of its Tables 8.3 and 8.5 (Table 8.5 detail 3, root failure 36*, and detail 8, shear 80, by default) or per line. The table numbers are the 2005 edition's, which stays valid until March 2028; EN 1993-1-9:2025 renumbers the detail tables.
- Steel strengths of EN 1993-1-1 Table 3.1 for t ≤ 40 mm.
- Only the weld throat is rated in fatigue (and a full-penetration butt line's toe when given its own category). **Toe failure of the connected plates** (EN 1993-1-9:2005 Table 8.5 details 1 and 2) is not checked, as the plate thicknesses are not inputs; the results and the report say so. Check the plates' nominal stress range at the weld toes by hand: on a thin plate with large fillets it governs.
- Not covered: weld lines out of one plane, curved or circular welds (approximate them with short straight lines), the side on which the attached part stands (the worse 45° throat is taken), hot-spot and effective-notch fatigue methods, stainless and aluminium welds, and lamellar tearing. There is no weld-specific case in `axomek verify` yet.

Known limits

- `axomek check --strict` and `axomek solve --strict` exit 3 for every file with something its method does not assess, which no input clears yet (4 where a check fails and the file was solved completely; a file that was not, an unknown field under `--strict` among it, exits 3 first): a rated key or spline (reversals and load peaks), a rated fit (its shaft seat), a belt or chain drive (its capacity), a fatigue-rated weld (its toes), a yoke on its stop, a support with a designation only and a planetary set without a planet bearing. Check those by hand and use `axomek check` without `--strict` for such files (it still exits 4 on a failing check and 3 on one not determined); a way to mark an item as checked by hand is for a later release.
- Weld fatigue follows EN 1993-1-9:2005, and says so on its method line, in the report and in Show calculation. EN 1993-1-9:2025 (Figure 8.2b) gives the welded details below category 71, the default fillet weld root category 36 among them, a lower curve: the constant amplitude fatigue limit at 10^7 cycles and $0.585 \cdot \Delta\sigma_C$ instead of $5 \cdot 10^6$ and $0.737 \cdot \Delta\sigma_C$, and a cut-off limit of $0.369 \cdot \Delta\sigma_C$ instead of $0.405 \cdot \Delta\sigma_C$, so ranges beyond $5 \cdot 10^6$ cycles do more damage than the app counts. Until the 2025 edition comes as an option (planned for a later release), check a weld of a category below 71 that sees more than $5 \cdot 10^6$ cycles against that edition by hand.

- Weld fatigue rates the weld throat (and a full-penetration butt line's toe when the line has its own category), not the connected plates at the weld toes (EN 1993-1-9:2005 Table 8.5 details 1 and 2): the plate thicknesses are not inputs. The results and the report say so; check the plates' nominal stress range at the toes by hand, which governs a thin plate with large fillets. A toe check with the plates' thicknesses is planned for a later release. Without them, a throat larger than its plate, or a full-penetration weld drawn as two lines at the plate's faces (the plate counted twice), is not caught; the help says how to draw it.
- A weld group does not say which kind of joint a weld is in, so EN 1993-1-8 4.11's long-joint reduction takes the group's fillet and partial-penetration welds as one lap, its length L_j the largest distance between two of their ends (gaps included, in whatever direction the force passes), wherever that is longer than 150·a or 1.7 m, also where the clause spares a weld (no lap, a girder's flange-to-web weld, 4.11(2)) or two welds stand far apart in one group, and the smaller of β_Lw.1 and β_Lw.2: on the safe side. From 900·a, where β_Lw.1 reaches 0, such a group is refused, also a weld the clause would spare (a = 4 mm over 3.6 m): give a larger throat, or design separate shorter joints. A weld group stands for one joint with all the welds of its lap: a lap drawn as several groups escapes the reduction, as no group sees the lap's whole length. A switch for a weld that is no lap is planned for a later release.
- Weld fatigue counts damage on the slope-5 branch for every range down to the cut-off, also in a spectrum of one block, where EN 1993-1-9 lets a constant-amplitude range below Δσ_D do none: a truly constant load is rated on the safe side.

Ball screws: ISO 3408

Ball screws by ISO 3408-4 and -5: load ratings, life, rigidity, buckling and critical speed, and the nut ball by ball.

Methods and standards

Ball screws after ISO 3408-5: equivalent speed and cubic mean load per load direction weighted by revolutions (eqs. 23–27); a preloaded double nut shares each step between its nuts with the operational preload 0.6·F_pr, Fa = fop·F_pr·(1 + F/(2^1.5·fop·F_pr))^1.5 and Fa − F on the other nut up to the lift-off load 2^1.5·fop·F_pr, where eq. 29 reaches F (eq. 28 prints 2^1.5·F_pr, which would leave the loaded nut below F and jump at the limit), above it F on the loaded nut alone (eqs. 28–34); L = (Cam/Fm)³ 10⁶ rev per direction or nut, resulting life (L1^−10/9 + L2^−10/9)^−9/10 (eq. 39), times the reliability factor of Table 3; modified ratings C_am = C_a·f_h·f_ac·f_m and C_0am = C_0a·f_h0·f_ac (hardness, accuracy grade, steel melting; eqs. 19–22, Tables 1–2); static safety C_0am over the largest nut load with the full preload; with the internal geometry the basic ratings C_0a and C_a of the nut from eqs. 1–18 and the static axial rigidity after ISO 3408-4 (screw shaft eq. 3 or 6 with the core diameter d_0 − D_w·cos α and E 210 000 MPa, nut body eq. 9/29, ball contacts eqs. 13–27/35, accuracy correction Table 1, overall eq. 2): a nut without preload at its largest axial load (eq. 27), a preloaded nut at its preload (eq. 35) up to its lift-off force 2^1.5·F_pr, beyond which the loaded half alone, without preload, at that load (eqs. 27 and 9 for one half); critical speed of the shaft as a beam of the root diameter (less a bore) n_cr = 60/2π·λ²/l²·√(EI/ρA) with λ 4.730, 3.927, π, 1.875 and Euler buckling π²EI/(K·l)² with K 0.5, 0.699, 1, 2 (fixed–fixed, fixed–supported, supported–supported, fixed–free; E 206 000 MPa, ρ 7.85 t/m³), allowed 80 % of n_cr and 50 % of F_k against the largest compressive force of the steps in the compression direction (catalogue factors f·d_2/l²·10⁷ and m·d_2⁴/l²·10⁴ are these allowed values already); efficiency tan φ/tan(φ+ρ); preload torque 0.05·(tan φ)^−½·F_pr·P_h/2π.

The nut is an end-cap nut: each thread start is one circuit running the whole nut (each half of a preloaded double or lead-offset nut), so its loaded turns i are the starts times the loaded turns per start. A single nut preloaded by a lead offset (its two halves in two-point contact, which ISO 3408-4 clauses 5.1 b and 5.5.2.3 rate with the double nut) shares each step as the double nut does, and so does a single nut preloaded by oversize balls in four-point contact on a Gothic arch, its two contact pairs taken as the two nuts, each with all the nut's turns (ISO 3408-4 clause 5.5.2.4 rates it like a double nut; ISO 3408-5 has no clause of its own for either single nut); the entered C_a is per half, or of the whole nut for the four-point nut, and the ratings and rigidity from the geometry take one half's turns (eq. 36) or, for the four-point nut, all its turns (eq. 37), the nut body twice (eq. 29) for every preloaded nut.

A preload on a nut with backlash, none on a preloaded nut, and four-point contact on a nut with backlash or a double nut are refused.

Thread starts n put the grooves P_h/n apart: the ratings, balls per turn and lead angle follow from the lead P_h, the nut body's loaded length in eq. 9 is i·P_h/n, and grooves no wider apart than the balls are refused.

Four-point contact rubs more than two-point, which no standard puts a number on: the torques take the friction coefficient as entered, to be taken from the maker with the preload torque.

With the internal geometry, ball by ball: each ball between the screw and nut grooves (centres (fs + fn − 1)·D_w apart at the contact angle), its load from the Hertz approach of both contacts (exact elliptic integrals) at the contact angle the axial shift gives; the shift along the nut changed by the strain of the screw core and nut body under the force each carries (load taken out at the nut's far or flange end); a preload as two halves pressed against each other (a double nut, or a lead-offset nut's two halves), or for a four-point nut both contact pairs of each ball pressed by the oversize balls and solved together, the core and the body strained by the net force; each thread start's circuit on its own helix, P_h/n along; life from the ball loads after Lundberg–Palmgren per turn and groove (ISO

281 point-contact capacities, no b_m) by Palmgren–Miner over the spectrum (a ball carrying less than 10^{-9} of the largest ball load counts as unloaded), also scaled to the catalogue C_a by $(C_a/C_{geometry})^3$; static safety $(4200\ MPa/p_{max})^3$; the contact ellipse's reach against the groove edge.

Validation

29 of 29 cases within their tolerance, in 3 groups.

Worked examples of standards and books

Ball screws: ISO 3408-4 rigidity and ISO 3408-5 life

Kind of reference	Worked examples of standards and books
Implements	ISO 3408-4:2006; ISO 3408-5:2006 eqs. 28–32, 39, Table 3
Reference source	ISO 3408-4:2006, Annex A
Cases	20 of 20 within tolerance

The static axial rigidity of the preloaded double nut worked in ISO 3408-4:2006 Annex A (Ø63.5 × 5, D_w 3.5, i = 5, z_2 = 3, f 0.55, α 45°, D_1 75, F_pr 4 kN, ls 1 m fixed at both ends, grade 3; the annex rounds its intermediate values to four digits); ISO 3408-5:2006 preloaded nut loads (eqs. 28–32 with fop 0.6, lift-off where eq. 29 reaches the force), the resulting life of two lives (eq. 39) and the reliability factors of Table 3; the screw shaft's first bending critical speed and Euler buckling load as a beam.

Case	Reference	Axomek	Deviation	Tolerance	
Balls per turn z_1 (eq. 21)	54	54	0 %	exact	**pass**
Screw shaft Rs2,min (eq. 6)	2457 N/µm	2456.895 N/µm	0.0043 %	0.1 %	**pass**
Nut body Rn/s,pr (eqs. 9, 29)	7458 N/µm	7461.251 N/µm	0.044 %	0.1 %	**pass**
Rigidity characteristic k (eq. 23)	105.07 N/µm^1.5	105.0584 N/µm^1.5	0.011 %	0.1 %	**pass**
Ball contacts Rb/t (eq. 35)	2923 N/µm	2923.023 N/µm	7.9×10^{-6}	0.1 %	**pass**
Nut unit Rnu,ar (eqs. 36, 38)	1155 N/µm	1155.129 N/µm	0.011 %	0.1 %	**pass**
Ball screw Rbs (eq. 2)	785.7 N/µm	785.7175 N/µm	0.0022 %	0.1 %	**pass**
Four-point nut, 1 × 5 turns: Rbs (eqs. 37, 2)	785.7 N/µm	785.7175 N/µm	0.0022 %	0.1 %	**pass**
Lead-offset nut, 2 × 5 turns: Rbs (eqs. 36, 2)	785.7 N/µm	785.7175 N/µm	0.0022 %	0.1 %	**pass**
Loaded nut at F = F_pr (eq. 29)	1202.104 N	1202.104 N	0 %	10^{-12}	**pass**
Relieved nut at F = F_pr (eq. 30)	202.1036 N	202.1036 N	0 %	10^{-12}	**pass**
Nut loads at no force: fop·F_pr	600 N	600 N	0 %	10^{-12}	**pass**
Relieved nut above Flim (eq. 32)	0 N	0 N	0 N	1×10^{-12} N	**pass**
Loaded nut at lift-off 2^1.5·fop·F_pr (eqs. 29, 31)	1697.056 N	1697.056 N	0 %	10^{-12}	**pass**
Resulting life both ways (eq. 39)	1.0718×10^{9} rev	1.0718×10^{9} rev	1.6×10^{-15}	10^{-12}	**pass**
Critical speed, supported–supported	3621.022 rpm	3621.022 rpm	0 %	10^{-12}	**pass**

Case	Reference	Axomek	Deviation	Tolerance	
Euler buckling load, supported–supported	80839.18 N	80839.18 N	0 %	10^{-12}	**pass**
Reliability factor far at 95 %	0.62	0.62	0 %	10^{-12}	**pass**
Reliability factor far at 97 %	0.44	0.44	0 %	10^{-12}	**pass**
Reliability factor far at 99 %	0.21	0.21	0 %	10^{-12}	**pass**

Closed-form solutions

Ball screws: ball by ball

Kind of reference	Closed-form solutions
Implements	Hertz contact; equilibrium of the balls
Reference source	Closed-form solutions
Cases	8 of 8 within tolerance

Hertz contact of a ball on a flat against the closed form; a Ø32 × 10 screw with 6.35 mm balls and rigid screw core and nut body: the balls' axial loads add up to the force, a preloaded double nut's second half lifts off at 2^(3/2) = 2.83 × the preload (Hertz, constant contact angle), and the life falls with the cube of the load; C_a rates one nut of a double nut, so far above lift-off the loaded half lives as a single nut of its turns.

Case	Reference	Axomek	Deviation	Tolerance	
Ball on flat: contact radius	0.3191252 mm	0.3191252 mm	1.7×10^{-16}	10^{-9}	**pass**
Ball on flat: peak pressure	4688.341 MPa	4688.341 MPa	5.8×10^{-16}	10^{-9}	**pass**
Sum of the balls' axial loads at 5 kN	5000 N	5000 N	2.6×10^{-14}	10^{-6}	**pass**
Largest over mean ball load, rigid	1	1	8.9×10^{-16}	10^{-9}	**pass**
Double nut lift-off over preload	2.828427	2.844468	0.57 %	5 %	**pass**
Life at F over life at 2F	8	7.890882	1.4 %	5 %	**pass**
Lifted-off double nut over a single nut of its half: rating	1	1	0 %	10^{-6}	**pass**
Lifted-off double nut over a single nut of its half: life at C_a	1	1	0 %	10^{-6}	**pass**

Mounted ball screw: critical speed over its bearing span

Kind of reference	Closed-form solutions
Implements	—
Reference source	Closed form: a beam's first bending critical speed n = 60/2π·λ²/L²·(d_2/4)·√(E/ρ), λ = 3.9266 fixed–supported, over the span between the screw shaft's bearings
Cases	1 of 1 within tolerance

The allowed speed is 80 % of it, as the screw's check takes it.

Case	Reference	Axomek	Deviation	Tolerance	
ball-screw-bearings BS1: allowed speed over the 1125.5 mm span	5239.57 rpm	5239.564 rpm	1.2×10^{-6}	0.001 %	**pass**

Stated limits

Standards and scope (help: modules/ball-screws.md)

- Life after ISO 3408-5:2006 and rigidity after ISO 3408-4:2006. The 1989 drafts DIN 69051-4 and DIN 69051-6 differ in several constants; the program follows the ISO parts.
- The lift-off of the preloaded double nut is taken where eq. 29 reaches the force (2^1.5 · fop · Fpr), not at the 2^1.5 · Fpr that eq. 28 prints, so the loaded nut never carries less than the force.
- `axomek verify` checks the rigidity against the worked example of ISO 3408-4 Annex A, the preloaded nut loads, eq. 39 and Table 3.
- Not covered: thermal growth of the screw, lubrication and wear, the choice of motor and gearbox (the profile gives the torques and speeds at the screw), and fatigue of the nut body. The support bearings and the screw shaft's strength are rated when the screw is on its shaft (see above), with the nut at the one position given; the end fixity is checked against the bearings' places and axial roles, not their tilt stiffness, and the unsupported length is not compared with the bearing span.

Known limits

- On the steering's ball nut shaft, the friction of the nut's linked bearings shows in the power flow as power leaving that shaft (19 W in the Ball-nut EPS example): the nut's stage passes on what reaches its thread. The totals and efficiencies hold it once, as the report's power loss does.
- Steering duty, for a later release (user test ut4): - A worm pair or a gear pair over a steering duty keeps its own required life (a worm pair rated at its worst state as if that lasted its life, a gear pair over the cases at its own life's cycles); the duty's length checks the ball nut's, the ball screws' and the bearings' lives. - Editing a mesh's Peak rack force for static rating in the duty's refusal card makes the duty not computed again, so the card (and the field in it) goes until Compute duty runs; with two refused meshes, the second is edited after recomputing, or on its mesh's own panel. - The drivetrain's natural frequencies (system modes) are dropped as out of date when another shaft is shown, though they do not depend on it; Compute them again. - While the duty computes, a check it decides reads "computing…" in every list of checks the app shows, also another report's (a version compared) of the same kind; the project's verdict line keeps the engine's count (a failure at the state counts) and says the duty is being computed. - A design search's work budget counts a remembered result of the steering duty at every request within one solve, so a design that runs on and on is stopped later than it could be (its time limit and Cancel still stop it); the engine's other remembered results (a gear tooth root's or a housing's finite elements) are not counted, which the budget's floor of 5000 steps covers.
- Steering misuse events (user test ut4): the misuse events rate the pinions' tooth roots (their contact stress is reported, not checked) and the pinions' shafts statically; the ball nut and its bearing are rated in the steering state and over the duty's strokes, not at a misuse event's peak, and the elements keep their own peak factor for a static peak. A worm pair that drives an assist pinion (a plastic wheel too), the other shafts and the bearings are not rated at a misuse event either, and only the report's method notes say so: the results do not list them as not checked for it. A mesh's peak rack force for static rating is a case of its own, the whole force through that pinion with the assist failed, rated for root and contact, so the same kerb strike can pass as a misuse event and fail there; the checks do not name it as the assist-failed case. Both err on the safe side or are disclosed; listing them on screen is for a later release (RR1 U-A25).

- A refusal's fix buttons, all for a later release: for an item whose id does not start with a capital letter, a suggestion is found only when it changes that item itself; in a locked project the buttons are disabled without saying why; the item solves a suggestion takes while something is refused are not counted in the solve-work table; the ball screw's lead table follows ISO 3408-1's nominal leads, not yet checked against the standard. A planet bearing whose outside diameter reaches the planet's root diameter is refused and its field marked, with no button (no number to search).
- A ball screw on its shaft (`screws[].mounting`) loads it with the nut at one place (`mounting.position`), not along its stroke; its end fixity is checked against the bearings' places and axial roles, not their tilt stiffness, and its unsupported length is not compared with the bearing span. On the safe side, as the screw's states rate the shaft: a load step is taken as driving (the larger torque), and the screw shaft's fatigue (and that of every shaft joined to it) takes its whole torque and axial force as fully reversed over two or more cases even where they never turn round. Refused rather than combined: a mounted screw with a motion profile (its ramps' bearing damage needs the life integrated over the speed), one with steps of its own in a project with load cases or steering strokes, and a second screw with steps of its own beside one whose steps are the cases (a later release).
- A ball screw whose nut turns (`turns: "nut"`) is mounted by its nut's shaft; the screw, which travels, is not one of the project's members, so what it pushes takes its thrust only in the user's own loads. A step the load drives back on a mounted screw whose shaft is joined to another (coupling, drive, gear pair) is refused: a load case cannot turn the train's power round (a later release).
- A mounted ball screw that does not derive the load cases (one step, or linked to them) is the shaft and turns at its speed: entered at another speed (0 too) it is refused, not rated at its own, whose critical speed, d·n and life would be at a speed it does not turn at; rating it at the shaft's speed is for a later release.
- Ball screw nuts are end-cap nuts, each thread start one circuit running the nut (each half of a split nut). A nut with tube or internal-deflector returns (several circuits a start, each over a few turns) is not modelled as such, for a later release: enter its total loaded turns per start. Its ISO 3408 ratings and rigidity are then right (they take the loaded turns i); the ball model's load per ball and life differ by a few per cent from those of separate circuits (see "From 0.3.0").

- A steering's ball nut (`steering.system.assist.ballNut`) is no radial support of the rack itself (0.3.0): the rack needs a support there (a bushing with the nut bearing's stiffness, as the "Belt-driven ball-nut EPS" template has). The torque the nut passes into the rack's thread is not put on the rack (it is held against turning by the pinion's teeth and the yoke, whose ratings leave it out), and the nut's travel is not checked against the threaded length: keep the thread long enough for the nut over the rack's travel. On-centre, the feel's friction torque takes the preload's and the nut bearing's drags but not the motor's, which only the return force counts; it takes the nut's drag with the motor idle (the bearing's friction under that drag at the creep speed), so where the assist curve gives torque within the friction band the band comes out a little wide (a 1000 N preload on the template's nut, before its bearing's drag was counted: 0.601 N·m against 0.591 N·m solved at the band's edge). The steering's balance counts the friction of the bearings linked on the nut's shaft only (as a drag at the nut, see "From 0.3.0"): a bearing linked on another shaft of the steering's drive (the column, the pinion, the motor) is in the power loss but its friction is not in the driver's and the motor's torques, which come out low by it (no example links one). The nut bearing's friction in the balance is its rating's as the solve rates it (its housing's seats moved as the solve moves them); should the two differ, the solve warns at `steering.system.assist.ballNut`. A bearing linked on the nut's shaft whose outer ring sits in another shaft (a bearing between shafts) refuses the steering's state: its friction turns with the two shafts' relative speed and may drive the nut, which the balance does not hold; seat it in the housing, or unlink it to take `bearingDrag`. For a later release. Its thread is taken right-handed (a left-hand one only turns the nut the other way). Without a duty its screw is rated in the steering state alone, so its life is that state's; a steering duty with strokes rates it over the duty's states (0.3.0). The rack's misuse events leave the nut's force out: the meshes hold the event alone, on the safe side for their teeth and the bar between them and the joints, while the nut and its bearing are rated in the state only.
- Where a rack mesh is given its pinion torque and no efficiency (its own rating then takes 0.92), the steering duty rates its force tables at the default for a known rack force, 0.85: a stroke the duty card starts at the mesh's rated rack force is rated 0.92/0.85, 8 %, above the mesh's own driving load, and linked items whose loads follow the rack's actual force (a ball screw) are scaled 8 % above it too. On the safe side; exact where the mesh's efficiency is entered.
- A steering duty whose strokes reverse the rack force (a return pass against a resisting load, a signed load below zero) is refused at `steering.duty` beside anything the load cases reach that a case cannot reverse (0.3.0): a mounted gear or worm pair or rack mesh, a drive, a coupling, a rotor, a bearing with loads of its own, a gear pair rated over the cases. The derived cases reverse only the entered shaft loads and ball screw forces, so the others would be rated at the forward loads on the reversed passes (a rotor's torque, unreversed, cancelled the reversed input in one review's case: 18.444302 where 9.222151 is right). In gearbox-single-stage with a rack mesh and resisting strokes with return passes, BR1 passed 5.5 at 5.778100 where the reversed pass gives 5.260948. One-way strokes are rated; reversing stages over the duty come in a later release.

- A steering duty over a steering system (0.3.0) rates the rack meshes' teeth, and with strokes the system's shafts, the bearings linked on them that take the load cases, its worm pairs and its ball nut over its states (see Steering system). Not rated over the duty (a later release): a mounted gear pair (rated at the system's `state`, its checks not rated over the duty, never a pass: its rating takes its solved shafts' KHβ, which the states do not give it yet), and likewise fits, keys and housings. A rotor's air gap is not rated over the states (its band between them is not drawn; never a pass: rate it at the duty's heaviest state). A bearing's warning over the states (a minimum load undecided at its valley) is never satisfied, and its gyroscopic slip (the high-speed model's) is the system's state's, a pass not rated over the duty. Beside such a duty a design search's metrics are refused but the shaft's mass (their values over the states have no bound between them, and some items' results are the system's state's): search with requireChecks, whose checks take the bands. The ball nut's life is the states' Miner sum (as a bearing's), its static safety the worse of the states' and the system's own state's. Left at the system's `state`: the couplings, a bearing with `useLoadCases` false, and everything at a misuse event (the events rate the pinions only; the elements keep their own `peakFactor` for a static peak, which over the duty multiplies every state's loads, as over load cases). A worm pair and a drive take the worst of their states, each rated as if it lasted their life: on the safe side, as DIN 3996 holds no load spectrum here. A shaft's states are each rated with their stresses fully reversed (as a mounted ball screw's states are), on the safe side where a stroke does not turn round. The band of a bearing's checks is drawn from its life, static safety, contact stress and thermal speed in the states, relative to each check's utilisation. Two supports in seats of one housing (whose seats push each other with the bearings' loads) leave the elements unrated over the duty (a warning at `steering.duty`, their checks never a pass): the states do not couple the seats again (a later release). The project's own load cases beside such a duty rate the shafts and bearings instead (a warning at `steering.duty`), and their checks are then not rated over the duty (never a pass): remove the load cases to rate them over it. The elements' states are some four times the duty points (284 for the Dual-pinion EPS's duty), each a state solve and a solve of every shaft: at most 1200 (`system_cases::CASE_LIMIT`) and at most 12,000 solves (`WORK_LIMIT`: each state one, and one for each shaft and each bearing on a shaft; some 25 s natively and 40 s in the browser), past which the elements are not rated over the duty (a warning at `steering.duty`; their checks never pass) while the teeth are. Measured under load on the build machine (one process, natively): the Dual-pinion EPS's duty 9.3 s for the teeth and 19 s in all; the Belt-driven ball-nut and Column EPS examples given its duty 2.4 s and 2.8 s for the teeth, 7.0 s and 5.2 s in all (168 and 88 states); the browser's engine took about the same under that load (17.9 s, 7.6 s and 6.3 s). It refuses a stroke's `torque` table (the system splits the rack force, not a pinion's torque); a rack mesh off the system's rack is not rated over the duty (a warning; it keeps its rating at its own load). Each duty point is a state solve (some 15 to 30 ms natively), at most 4000 per duty. Rating the teeth of two pinions is the slow part: the Dual-pinion EPS template's duty (a parking, a manoeuvring and an on-centre stroke, both ways) takes some 4.7 s natively and 7.3 s in the browser's engine on the build machine (some 20 s natively on a laptop, 4 s of it the state solves); the app computes it on request, with progress and Cancel (0.3.0), and solves each edit without it. Positions past the rack's travel are refused (each point is a state of the system, within its stops). The lift is rated within 1e-4 mm of the states' between points (the contact ratios within some 4e-5), not raised; the forces are raised to bound theirs. Misuse events are rated at standstill (vehicle speed 0, forward efficiencies, no friction).

- Over a steering system's duty states (0.3.0), all on the side of saying more or of reporting less, for a later release: a rack buckling span that exists in some of a span's states and not in others (the bar moves under the supports) draws no marginal band over that change; a shaft standing still at the system's `state` takes FKM's checks over a number of cycles at the worst state for all its cycles instead of their Miner sum; a state that gives the ball nut no thrust can leave its rating over the duty refused; the system's own state, kept among the worst-case checks, is rated with its mean and amplitude stresses where the duty's states take theirs fully reversed (a state heavier than every duty point is rated more leniently than they are); a refused duty marks every drive's checks, also those of a drive off the steering's train; a bearing's band for its checks other than life and static safety is drawn from those quantities' bands rather than its own; `atState` does not list bearings off the shafts or planet bearings (rated at their own loads), nor the couplings.

Linear guides: ISO 14728

Profiled rail guides by ISO 14728: carriage loads, ratings and life.

Methods and standards

Linear guides after ISO 14728-1/-2: carriage loads on a rigid table shared linearly over the carriage pattern, $F_{z,i} = F_z/n + M_x \cdot y_i/\Sigma y^2 + M_y \cdot x_i/\Sigma x^2$ with roll $M_x = F_z \cdot y + F_y \cdot z$ and pitch $M_y = F_z \cdot x + F_x \cdot z$ (F_z pressing, z up: a horizontal force above the carriages presses the side it points to); moments the pattern cannot react are carried by each carriage and counted as $C_0 \cdot M/M_0$; equivalent load $P = |F_y| + |F_z|$ + moment terms; life $L = (f_H \cdot f_T \cdot f_C/f_W \cdot C/P)^p \cdot 50$ or 100 km (p = 3 balls, 10/3 rollers); system life with Weibull slope 10/9; static safety C_0/P.

Validation

6 of 6 cases within their tolerance, in 1 groups.

Closed-form solutions

Linear guide: carriage loads by rigid sharing, and the rating life

Kind of reference	Closed-form solutions
Implements	ISO 14728-1 (rating life $L = (C/P)^3 \cdot 100$ km for balls)
Reference source	Statics of a rigid table on 2 × 2 carriages (rails 300 mm, carriages 200 mm apart), C 30 kN, worked by hand
Cases	6 of 6 within tolerance

Case	Reference	Axomek	Deviation	Tolerance	
F_z 8 kN central: each carriage's load	2000 N	2000 N	0 N	1×10^{-9} N	**pass**
F_z 8 kN central: rating life $(C/P)^3 \cdot 100$ km	337500 km	337500 km	0 km	1×10^{-6} km	**pass**
Life in hours at 10 double strokes of 0.5 m a minute	562500 h	562500 h	0 h	0.001 h	**pass**
System life of four equal carriages over one's: 4^(−0.9)	0.2871746	0.2871746	1.1102×10^{-16}	1×10^{-12}	**pass**
F_z 8 kN at (100, 150) mm: nearest carriage	6000 N	6000 N	0 N	1×10^{-9} N	**pass**
F_z 8 kN at (100, 150) mm: opposite carriage	-2000 N	-2000 N	0 N	1×10^{-9} N	**pass**

Not validated against a published reference

- No worked example of ISO 14728 is compared: the cases are the statics of a rigid table and the life formula

Stated limits

Standards and scope (help: modules/linear-guides.md)

- Rating life after ISO 14728-1 and static safety after ISO 14728-2, with the life factors f_H, f_T, f_C and f_W as catalogues state them.
- The table is rigid and all carriages equally stiff. Elastic tables, carriage preload, rail mounting errors and statically indeterminate stiffness sharing are not modelled.

- One load case per guide, or a motion profile: a profile varies the force along the rails and optionally adds the constant normal weight m·g·cos θ to F_z (off by default). F_y and entered F_z stay constant. For loads across the rails that change over the cycle, use the largest or a weighted mean.
- Lubrication, seals, the drive force capacity and the rail's own strength are not assessed.

Belt and chain drives

Belt and chain drives: strand forces, shaft loads, slip and the drives’ ratings.

Methods and standards

Belt and chain drives: effective force Ft = 2T/d_1, ratio d_2/d_1 without slip (z_2/z_1 with teeth), output torque T·i·η.

Strand tensions after Euler–Eytelwein (new drives): wrap angle β = π − 2·asin((d_2 − d_1)/2a) on the smaller pulley, (F_1 − F_c)/(F_2 − F_c) ≤ e^(μ′β) with μ′ = μ (flat belt) or μ/sin(φ/2) (V-belt, groove angle φ), centrifugal tension F_c = q·v²; with fixed centres F_1 = F_0 + Ft/2 and F_2 = F_0 − Ft/2 from the pretension F_0 (the least that does not slip, F_c + (Ft/2)·(e^(μ′β) + 1)/(e^(μ′β) − 1), unless given; a timing belt's least keeps its slack strand taut, F_c + Ft/2; a chain's slack strand carries F_c only); the shaft load is the vector sum of F_1 − F_c and F_2 − F_c along the strands, off the centre line towards the tight strand; slip check F_0 against the least.

Sprocket pitch diameters p/sin(180°/z), the chain's speed varying by 1 − cos(180°/z_1) (polygon effect).

Drives saved without a tension model keep the shaft load k·Ft along the centre line (k typically 2 for V-belts, 2.5 flat belts, 1.3 timing belts, 1.1 chains).

Validation

6 of 6 cases within their tolerance, in 1 groups.

Closed-form solutions

Belt and chain drives: strand forces, wrap and pitch polygons

Kind of reference	Closed-form solutions
Implements	—
Reference source	Closed forms: Euler–Eytelwein's capstan ratio e^(μβ), the open belt's wrap angle 180° − 2·asin((d_2 − d_1)/(2a)), a V-belt's wedge friction μ/sin(φ/2), a sprocket's pitch diameter p/sin(180°/z) and its polygon effect 1 − cos(180°/z)
Cases	6 of 6 within tolerance

Mechanics of the strands only; belt and chain ratings to their makers' tables are not validated here.

Case	Reference	Axomek	Deviation	Tolerance	
Chain p 12.7 mm: pitch diameter of a 17-tooth sprocket	69.11583 mm	69.11583 mm	0 %	10^{-15}	**pass**
Chain, 17 teeth: polygon effect 1 − cos(180°/z)	0.0170269	0.0170269	0 %	10^{-15}	**pass**
Flat belt, μ 0.3, 180°: friction ratio e^(μβ)	2.566332	2.566332	0 %	10^{-12}	**pass**
Flat belt, F_t 1000 N: least pretension F_t/2·(m + 1)/(m − 1)	1138.434 N	1138.434 N	0 %	10^{-12}	**pass**
V-belt, μ 0.25, 34° groove: wedge friction μ/sin 17°	0.8550759	0.8550759	0 %	10^{-12}	**pass**
Ø100 to Ø300 at 400 mm: wrap angle of the small pulley	151.045 °	151.045 °	2.5×10^{-9}	10^{-8}	**pass**

Not validated against a published reference

- No standard’s or maker’s rating of belts and chains: only the mechanics of the strands are validated

Stated limits

Standards and scope (help: modules/drives.md)

- The strand model is the classical Euler–Eytelwein capstan relation with centrifugal tension. No belt or chain standard is applied.
- Not assessed: the belt section and number of belts, belt power ratings, belt or chain life, chain wear, tooth jump of timing belts and sprocket strength. Use the belt or chain maker's data for these.
- The shaft load is the running one: each strand pulls with its tension less F_c. At standstill both strands pull with the full pretension F_0 (with the strand model), so a belt with a large centrifugal tension pulls the shafts about 2·F_c harder there. The shafts' and bearings' static checks take the running load: check them at the standstill pull yourself where F_c is large.
- The centre distance is fixed; tensioner pulleys and idlers are not modelled.
- The efficiency is a constant per kind, not computed from the load.

Known limits

- `axomek check --strict` and `axomek solve --strict` exit 3 for every file with something its method does not assess, which no input clears yet (4 where a check fails and the file was solved completely; a file that was not, an unknown field under `--strict` among it, exits 3 first): a rated key or spline (reversals and load peaks), a rated fit (its shaft seat), a belt or chain drive (its capacity), a fatigue-rated weld (its toes), a yoke on its stop, a support with a designation only and a planetary set without a planet bearing. Check those by hand and use `axomek check` without `--strict` for such files (it still exits 4 on a failing check and 3 on one not determined); a way to mark an item as checked by hand is for a later release.

- An electric motor's critical speed and magnetic stability with a static eccentricity on coupled bearings (review of ut4-motor-critical-speed; see "From 0.3.0"). In 0.3.0: - A pull not along a bearing's other loads (more than 1° apart: a belt or a gear's force across the weight) leaves both not rated: the bearing's load turns as the eccentricity grows, and its stiffness along y or z can be least in between, which neither the states taken nor a bound with each bearing equally stiff every way at its least load bounded (a reviewer's case read 9 988 rpm where a state inside the bound gives 9 604 rpm). This blocks a belt- or gear-driven motor with an eccentricity from the coupled critical speed, and 0.3.0 names no way around it: no stiffness entered on the supports (with coupling off) is known to be safe, as the least one lies somewhere between centred and pulled (keep the eccentricity: its pull counts in the bearing loads and the air gap; 0 would drop it from them). - A state the coupling cannot settle exactly (bearings loose in their clearance, e.g. the Electric motor example without its wave spring, centred) leaves a passing result not decided (a warning). A motor whose bearings carry no load with the rotor centred (the weight off, a vertical motor without preload) has both not rated, with a hint (a horizontal motor: turn the weight on; a vertical one, or one held only by a preload, 0.3.0 cannot rate). The hints of the not decided and not determined results name no stiffness to enter: none is known to be safe there (review: a recipe of the bearings' stiffness without the pull could pass a motor that fails). - A coupled bearing under an axial load (a spring preload, as in the Electric motor example, or the shaft's own axial loads) or preloaded from within (an X or O pair's preload, a negative operating clearance) leaves a passing result not decided (a warning): its radial stiffness need not grow with its radial load, so the states taken bound nothing between them (a reviewer's case: e = 0.2 passed 15 000 rpm at 15 908 rpm where e = 0.15 gives 13 878 rpm). - A spring preload is refused on any shaft joined to another by a bearing between them (shafts solved together do not carry it), also where the spring would act on its own housing. - The unbalance response is the given eccentricity's, with a warning: at a smaller one it can be larger. - Up to four pulling rotors on a shaft; more have both not rated. The axial modes of a state keep the locating bearings' axial stiffness and the housings' seat motions of the pulled state. A critical speed or magnetic stability that is not rated fails, shown as "not rated" in place of its placeholder value (0 rpm, a share of 100 %), and What matters most leaves it out, saying why. Planned for a later release: the rotor pulled within its bearings' clearance to where they hold it (it never sits centred there), in place of the centred state; and a sweep of the eccentricity with a proven bound between its steps, for a pull across the bearings' loads, for bearings under an axial load (a spring preload) and for the unbalance response.
- Belt and chain capacity (user test ut4): a belt drive checks only its pretension (against slip, or a slack strand), never what it can transmit: its power rating, tooth shear or tensile strength; a chain is not rated for wear and fatigue. The unchecked list names each drive's capacity as not rated. Take it from the maker's data.
- A belt's shaft load is the running one, its strands less their centrifugal tension F_c; at standstill a belt (friction or timing) pulls with its whole pretension, about 2·F_c more (a fast V-belt in user test RR1: 1.11 kN running, 1.40 kN at standstill). The shafts' and bearings' static checks take the running load times its peak factor, so the standstill pull is not checked; the drives help says so. Static checks at the larger of the two are for a later release (RR1 U-A21).

- A steering's ball nut (`steering.system.assist.ballNut`) is no radial support of the rack itself (0.3.0): the rack needs a support there (a bushing with the nut bearing's stiffness, as the "Belt-driven ball-nut EPS" template has). The torque the nut passes into the rack's thread is not put on the rack (it is held against turning by the pinion's teeth and the yoke, whose ratings leave it out), and the nut's travel is not checked against the threaded length: keep the thread long enough for the nut over the rack's travel. On-centre, the feel's friction torque takes the preload's and the nut bearing's drags but not the motor's, which only the return force counts; it takes the nut's drag with the motor idle (the bearing's friction under that drag at the creep speed), so where the assist curve gives torque within the friction band the band comes out a little wide (a 1000 N preload on the template's nut, before its bearing's drag was counted: 0.601 N·m against 0.591 N·m solved at the band's edge). The steering's balance counts the friction of the bearings linked on the nut's shaft only (as a drag at the nut, see "From 0.3.0"): a bearing linked on another shaft of the steering's drive (the column, the pinion, the motor) is in the power loss but its friction is not in the driver's and the motor's torques, which come out low by it (no example links one). The nut bearing's friction in the balance is its rating's as the solve rates it (its housing's seats moved as the solve moves them); should the two differ, the solve warns at `steering.system.assist.ballNut`. A bearing linked on the nut's shaft whose outer ring sits in another shaft (a bearing between shafts) refuses the steering's state: its friction turns with the two shafts' relative speed and may drive the nut, which the balance does not hold; seat it in the housing, or unlink it to take `bearingDrag`. For a later release. Its thread is taken right-handed (a left-hand one only turns the nut the other way). Without a duty its screw is rated in the steering state alone, so its life is that state's; a steering duty with strokes rates it over the duty's states (0.3.0). The rack's misuse events leave the nut's force out: the meshes hold the event alone, on the safe side for their teeth and the bar between them and the joints, while the nut and its bearing are rated in the state only.
- Over steering-derived load cases a shaft's FKM damage check (with a number of cycles) is undecided where it passes when the shaft's loads follow the speed (0.3.0): every shaft of a mounted stage (a gear or worm pair, a rack mesh, a drive), whose mesh efficiency is rated at each case's speed and enters a member's torque, and whose belt's pull rises with the square of the speed, and every shaft joined to one by a coupling or a bearing between shafts. The cases' chord split bounds that damage only where the stresses follow the force alone. In belt-drive with SH1's FKM over 800000 cycles and one-way strokes over a 126–130 mm/rev lead, the cases passed 6.681 at 6.681179 where the strokes give 6.680947.

- A steering duty over a steering system (0.3.0) rates the rack meshes' teeth, and with strokes the system's shafts, the bearings linked on them that take the load cases, its worm pairs and its ball nut over its states (see Steering system). Not rated over the duty (a later release): a mounted gear pair (rated at the system's `state`, its checks not rated over the duty, never a pass: its rating takes its solved shafts' KHβ, which the states do not give it yet), and likewise fits, keys and housings. A rotor's air gap is not rated over the states (its band between them is not drawn; never a pass: rate it at the duty's heaviest state). A bearing's warning over the states (a minimum load undecided at its valley) is never satisfied, and its gyroscopic slip (the high-speed model's) is the system's state's, a pass not rated over the duty. Beside such a duty a design search's metrics are refused but the shaft's mass (their values over the states have no bound between them, and some items' results are the system's state's): search with requireChecks, whose checks take the bands. The ball nut's life is the states' Miner sum (as a bearing's), its static safety the worse of the states' and the system's own state's. Left at the system's `state`: the couplings, a bearing with `useLoadCases` false, and everything at a misuse event (the events rate the pinions only; the elements keep their own `peakFactor` for a static peak, which over the duty multiplies every state's loads, as over load cases). A worm pair and a drive take the worst of their states, each rated as if it lasted their life: on the safe side, as DIN 3996 holds no load spectrum here. A shaft's states are each rated with their stresses fully reversed (as a mounted ball screw's states are), on the safe side where a stroke does not turn round. The band of a bearing's checks is drawn from its life, static safety, contact stress and thermal speed in the states, relative to each check's utilisation. Two supports in seats of one housing (whose seats push each other with the bearings' loads) leave the elements unrated over the duty (a warning at `steering.duty`, their checks never a pass): the states do not couple the seats again (a later release). The project's own load cases beside such a duty rate the shafts and bearings instead (a warning at `steering.duty`), and their checks are then not rated over the duty (never a pass): remove the load cases to rate them over it. The elements' states are some four times the duty points (284 for the Dual-pinion EPS's duty), each a state solve and a solve of every shaft: at most 1200 (`system_cases::CASE_LIMIT`) and at most 12,000 solves (`WORK_LIMIT`: each state one, and one for each shaft and each bearing on a shaft; some 25 s natively and 40 s in the browser), past which the elements are not rated over the duty (a warning at `steering.duty`; their checks never pass) while the teeth are. Measured under load on the build machine (one process, natively): the Dual-pinion EPS's duty 9.3 s for the teeth and 19 s in all; the Belt-driven ball-nut and Column EPS examples given its duty 2.4 s and 2.8 s for the teeth, 7.0 s and 5.2 s in all (168 and 88 states); the browser's engine took about the same under that load (17.9 s, 7.6 s and 6.3 s). It refuses a stroke's `torque` table (the system splits the rack force, not a pinion's torque); a rack mesh off the system's rack is not rated over the duty (a warning; it keeps its rating at its own load). Each duty point is a state solve (some 15 to 30 ms natively), at most 4000 per duty. Rating the teeth of two pinions is the slow part: the Dual-pinion EPS template's duty (a parking, a manoeuvring and an on-centre stroke, both ways) takes some 4.7 s natively and 7.3 s in the browser's engine on the build machine (some 20 s natively on a laptop, 4 s of it the state solves); the app computes it on request, with progress and Cancel (0.3.0), and solves each edit without it. Positions past the rack's travel are refused (each point is a state of the system, within its stops). The lift is rated within 1e-4 mm of the states' between points (the contact ratios within some 4e-5), not raised; the forces are raised to bound theirs. Misuse events are rated at standstill (vehicle speed 0, forward efficiencies, no friction).

- The steering duty rates only the bearings linked to the system's shafts' supports. A support that names a bearing by its designation only links none: nothing rates it, at the system's `state` or over the duty; the duty lists it (`elements.unlinked`) and the summary names it as not assessed. The steering examples draw their bearings so: the Dual-pinion EPS's 6001 and HK2816 on the steering pinion and 6001 and 6203 on the assist pinion's shaft, the Column EPS's and the Steering rack's likewise, every other support of theirs (the column, the intermediate and input shafts, the motors) plain. Only the Belt-driven ball-nut EPS links a bearing, its nut bearing BR1 (see "From 0.3.0"). So the Dual-pinion EPS's duty rates no bearing. The designations are drawn, not sized: linked as drawn (6001 preset, default grease), the Dual-pinion EPS's assist-pinion 6001 at S51 would not carry its parking strokes (static safety some 0.6). Sizing and linking the examples' bearings is for a later release; until then link bearings at those supports to rate them over the duty (a needle bearing such as the HK2816 as a bearing of your own).

Housings

Housing stiffness from finite element models, built in or imported.

Methods and standards

Housings: finite elements on a mesh that follows the bore (rings of elements round it, spokes out to the plate's edges, a ring on the hub's rim, ribs with straight sides), of a wall with the bearing bore, held at the chosen edges (or on bolted joints across them); plane-stress quadrilaterals for the plate model, eight-node hexahedra with incompatible modes (Wilson–Taylor) for the 3D solid with its hub and ribs; solved by sparse Cholesky factorisation.

The bearing load is a cosine pressure on the loaded half of the seat on the bore surface; the seat stiffness is the load over the seat's mean displacement: the wall's vertical one acts in the shaft's xy plane and its horizontal one in the xz plane, each in series with the support's stiffness and its bearing; in 3D a pressure growing linearly across the seat gives the tilt stiffness (moment over the slope of the seat's radial displacement), likewise per plane, in series with the support's own.

A wall with more bores is split between them by straight cuts and meshed round each; it is solved for a load on every seat, giving the condensed compliance between the seats (symmetric, the seat motion being measured as the work-conjugate of the load) and the coupled stiffness matrix; the loads on the other seats move a seat by that compliance, the shafts being solved again until those motions settle.

A housing imported from another FE program as a stiffness matrix reduced to the seat nodes (Nastran DMIG, Abaqus, Ansys) gives the seats' compliance as the matrix's inverse at the seat nodes (Cholesky factorisation; the other degrees of freedom unloaded), in the model's axes turned to the shaft's.

A housing imported as a Nastran solid mesh (linear and quadratic tetrahedra, wedges and hexahedra; the eight-node brick with incompatible modes) is solved by the same sparse Cholesky factorisation (nested-dissection or minimum-degree order, whichever fills less), its bores found as the seats by fitting circles to the outer faces facing the shaft's axis, the seat loads integrated over the faces with their own shape functions; held faces, face loads and seats may also be chosen by patch: outer faces joined across edges where their normals differ by under 30°, each recognised as a plane (normals within 2°) or a cylinder (normals across the one direction they leave out, a circle fitted to its corner nodes), a seat on a cylindrical patch taking its centre, diameter and length.

The housing's mass enters the critical speeds by Guyan reduction to the seats ($\Phi^T \cdot M \cdot \Phi$ over the static shapes of unit seat motions, consistent element mass): each housed support becomes the bearing's spring to a housing node held by the seat's stiffness and carrying its share of the mass, per bending plane.

A housing's temperature rise (uniform, $\int \nabla N \cdot E/(1-2\nu) \cdot \alpha \Delta T \, dV$, plane stress for the plate), its weight and point loads move its seats by the first harmonic of the bores' radial displacement, which the shaft solve takes as its bearings' ground moving.

A turning housing or imported part (speed n about the shaft's axis through seat 1) carries the centrifugal body load as consistent nodal loads $\int N \cdot \rho \cdot \omega^2 \cdot r \, dV$ (r from the axis at each integration point; density from the housing, MAT1 RHO for Nastran meshes); each seat's growth is the mean radial displacement over it (checked against Lamé's rotating annular disc in plane stress, $u = \rho\omega^2 r/(8E) \cdot [(3+\nu)(1-\nu)(a^2+b^2) + (3+\nu)(1+\nu)a^2b^2/r^2 - (1-\nu^2)r^2]$), and a bearing's housing fit there loses 2·(seat growth − the outer ring's free growth as a rotating annulus) of its interference.

Validation

8 of 8 cases within their tolerance, in 2 groups.

Closed-form solutions

Imported solid mesh: growth of a turning disc

Kind of reference	Closed-form solutions
Implements	—
Reference source	Lamé's plane-stress rotating disc: Ø80/Ø200 × 10 mm steel at 10 000 rpm, meshed with 20-node bricks and 4-node tetrahedra (6 × 48 × 2 cells)
Cases	4 of 4 within tolerance

The tolerance is the mesh's discretisation error: 1 % for 20-node bricks, 5 % for the stiffer 4-node tetrahedra.

Case	Reference	Axomek	Deviation	Tolerance	
20-node bricks: radial growth at r = 40 mm	0.01398674 mm	0.01392629 mm	0.43 %	1 %	**pass**
20-node bricks: radial growth at r = 100 mm	0.01258479 mm	0.01253072 mm	0.43 %	1 %	**pass**
4-node tetrahedra: radial growth at r = 40 mm	0.01398674 mm	0.01363602 mm	2.5 %	5 %	**pass**
4-node tetrahedra: radial growth at r = 100 mm	0.01258479 mm	0.01241462 mm	1.4 %	5 %	**pass**

Housing seat tilt in series with a coupled bearing

Kind of reference	Closed-form solutions
Implements	—
Reference source	Closed form: two rotational springs in series, the shaft's slope at the support θ = M·(1/k_b + 1/k_s), M the bearing's moment, k_b its secant tilt stiffness at the coupled state, k_s the seat's tilt stiffness from the wall's FE model
Cases	4 of 4 within tolerance

The housing-seats example (the overhung pulley shaft, each 6207 bearing in a grey-iron end shield of the wall's own 3D model) with bearing coupling on, the supports given no tilt stiffness of their own. The walls at the material's Young's modulus and at 1/100 of it, where the seat is softer in tilt than the bearing.

Case	Reference	Axomek	Deviation	Tolerance	
wall at E, slope at B1	1.617305 arcmin	1.617305 arcmin	2.7×10^{-16}	2 %	**pass**
wall at E, slope at B2	0.7868658 arcmin	0.7868658 arcmin	8.5×10^{-16}	2 %	**pass**
wall at E/100, slope at B1	3.224429 arcmin	3.224429 arcmin	9.6×10^{-16}	2 %	**pass**
wall at E/100, slope at B2	2.394366 arcmin	2.394366 arcmin	9.3×10^{-16}	2 %	**pass**

Not validated against a published reference

- Built-in housing walls and the planet carrier’s finite elements: compared in the tests with plate theory, mesh refinement and imported meshes, not with an external benchmark

Stated limits

Scope and limits (help: modules/housings.md)

- The model is linearly elastic. Held edges are rigid, or springs when bolted joints are given. No gaps, contact or slip are modelled.
- The bearing load is a cosine pressure on the loaded half of the seat. Seat stiffness is the load over the seat's mean displacement.
- Seat motion from loads on the housing is the first harmonic of the bore's radial displacement.
- The coupling between seats is left out of the critical speed modes, which take each seat's own stiffness and mass.
- A reduced matrix gets no centrifugal growth (a warning says so) and cannot carry an elastic outer ring.
- STEP import runs in the desktop app on Windows only.

Known limits

- A motor bearing's minimum load and gyroscopic slip over its rotor's static eccentricity (user test rr1-motor; see "From 0.3.0"). While any rotor in the project pulls, their pass (or a warning for a bearing its axial load holds) counts only in the case rated in full, where all of these hold: - an uncoupled shaft alone: Couple linked bearings off, in no group (a rigid coupling, a bearing between shafts) and in no housing whose seats move each other; - no load cases; - the minimum load checked on the radial load F_r alone (not the equivalent load P of angular contact, tapered roller and thrust bearings), one bearing on its support; - no axial load on the shaft (an axial load shares out anew and can hold or let go of a bearing between the states); - the least loaded state (the least radial load, linear in the pulls without coupling) solved exactly, and the bearing loaded at the eccentricity. Every other such pass or held warning is not decided, saying why: "enter the eccentricity 0 to see it" (the Electric motor example, coupled, with an eccentricity entered, is one; it ships centred). Every bearing's gyroscopic slip margin is not decided in every case while any rotor pulls: the least radial load does not bound its balls' gyroscopic margin (a later release bounds the margin itself). A later release: a bound on the coupled shaft's least bearing load between centred and the eccentricity (the bearings' load–deflection laws bound how far load moves between them), solved with the groups' and housings' own equilibrium in each state and per load case; P, the axial holding and the balls' loads minimised themselves.

- A result not determined is hidden display by display, not by one shared rule: in 0.3.0 the displays reviewed (the report, Show the calculation, the CLI's text, the comparisons, the Overview, the cards, the modules' tiles and the Assistant's digest) say "not determined". Those found last are refused instead: the Changes dialog's and Compare projects' check rows and the Overview's key results of keys and bolts say "not determined (see the check)"; the assembly view gives no share of the limit for a check not determined or not computed; and in the report an element's results, and a key's or a fit's results its checks go by (safeties, capacities, permissible values, pressures, shear, the fit's Turning row; with `axomek solve`'s key and fit lines), say "not determined (see the checks)" as soon as one of its checks is not determined, determined ones with them; a key's page refuses its flank pressures and tooth shear stress the same way (its permissible values and capacity, which do not depend on its rating, stay). A bolted joint with a check not determined (beyond VDI 2230's limit G, a duty not computed) has its whole Results block refused the same way in the report's section and in `axomek solve`'s line (its tightening torque, permissible assembly preload, resiliences and embedding with it, which the bolt page still shows); its page refuses its preload needed, load factor, clamp loads, stresses, opening and the joint diagram, but the notes under them, on its page and under the report's section, still quote the eccentric model's figures (M_Sb/W_b, F_Sm/F_0.2min, F_K against F_Kab). Make it pass and `axomek fix` give such a check no value either. A module's page, once one of an item's checks is not determined, shows no tile's caption (it may quote another check's figure) and refuses its tiles and Details tied to no check (a worm pair's load value C / Clim), and the Overview refuses the item's tiles tied to no check the same way, including ones that do not depend on the rating: a worm pair's method, centre distance and ratio, a motor's power and magnetic stiffness read "not determined" too; a later release shows those again. A design search meets no requirement or objective on a result of a design with any check not determined (a requirement's own check may be cleared, and the system life and power loss take every item's), its dimensions apart, nor on a fit's yield safety resting on an estimated yield strength, even where that figure itself would stand: the safe side; a later release decides each figure on its own. A bearing whose life the engine puts outside its rating method (κ below 0.1 where it turns, balls in four-point contact, a hybrid, whose a_ISO is defined for steel only) has all its lives, basic and modified, meet no search requirement or objective, as a minimum, a maximum or an objective, its minimum life cleared or not, and so has the system life with such a member; it says "not determined" without a life check too. The safe side: a hybrid's basic life, or one at a low κ, would stand; in a later release a search requirement on a figure whose own check is cleared is evaluated as that check, each life on its own. A design with a bearing the engine marks outside what it rates (a designation that contradicts it, a slewing ring's proportions) takes no calculated result in a search, its dimensions apart but that bearing's own (inferred from its geometry as modelled), as a design with a check not determined does, whatever its own checks say. A bearing whose designation contradicts it shows only its results' warnings it can vouch for: those at its own path, of the bearing as modelled, are left out (a validation warning at that same path too, such as that its geometry is estimated), its fields' warnings stay. A check not determined is said in a short form wherever its message would be rewritten: what it checks, named from its id, "not determined" and its kind's reason in fixed words, else "(see the check)", never a part of its message, which can hold a name the user wrote and the figures after it; a name that holds an engine mark's words can still pick another kind's reason (no figures). The full engine message, with the reason in context, comes back in a later release, when every display shares one rule. The bearing module's speed chart is not drawn while the bearing's operating nθ is not determined (a rotor's pull, a designation its geometry contradicts), saying why; `axomek speed-limits` then lists the curves' samples without an

operating point. A later release draws the curves apart from that load. A display not yet reviewed may still give such a number beside its check, which says "not determined". A later release routes every display of a check's value through one rule, with a test over every template. Known in 0.3.0, closed in a later release by that one shared display rule: - the Assistant's solve data still carries a check's value and utilisation where the check is not determined (a bearing's thermal speed whose nθ is withheld: its speed and share of the limit, from which the withheld nθ can be worked out); the check is flagged not determined there, the solve tool tells the Assistant never to quote a figure so flagged, and the check's row and every display say "not determined", never a pass; - a rack pinion shaft not rated at its mesh's peak because its only drive is solved for one power direction (the Dual-pinion EPS example's SH5, driven by worm pair WG1) is said, in the verdict's and the report's short reason, to be refused because its torque sources cannot be shared; its check's own note gives the right cause; - a design search takes no result from a design with any check not determined, so a project whose steering duty leaves a warning over its states may find no design meeting a shaft or bearing requirement (the safe side; a later release decides each figure on its own); - `Life0utside::against_bounds` keeps a maximum branch no caller uses (the search reads the hybrid flag directly); a later release removes it; - the Overview's governing check of a module, between two checks not determined, keeps the first in the list (the bearing listed first) where the engine picks by id, so the module tile and the verdict can name different checks, both said not determined; - a variant comparison can mark as best the life, and its check's margin, of a bearing the engine marks unrated (its designation contradicts its geometry), though the bearing's own results say its figures are of the bearing as modelled: compare such a design only after its designation is cleared or corrected; - a comparison of three or more designs can leave out a result row whose neighbouring columns each agree within the tolerance while the first and last do not (10 000, 10 000.8 and 10 001.6 h); `--all` shows it; - a comparison of designs whose life checks all pass on a hybrid's basic life (its modified life outside its rating method, so no best) leaves that check row out of the default CLI comparison and the report, though the minimum lives and the checks' shares differ (a hybrid 6309 at a 1 million h minimum against 4 million h); `--all` shows it. - a hybrid whose modified life fails its minimum keeps that failing check's number in its check row (the fail stands, on the safe side; the number rests on a_ISO, defined for steel bearings only), and so does any life check failing outside its rating method (a steel bearing turning below κ = 0.1, or with its balls in four-point contact): the report's Summary of checks and its Summary page's Governing checks show its share and draw its utilisation bar like any other (a 30207 on 2 / 1 cSt oil against a 20 000 h minimum), though that share cannot be compared with another check's; the governing pick and the orders leave it unranked, and the typed results' report follow-up shows no bar for it. Show the calculation of a hybrid's life explains the modified life, said not determined, not the basic life its passing check goes by; - the Assistant's shortened record of an earlier solve lists a failing life check with its value (a hybrid QJ 207 in four-point contact: 94 476 h against a 1 000 000 h minimum) but without the check's outside-method flag or reason, though the same record says the bearing's life is not determined; the fail stands, and the full solve the Assistant reads carries the flag; - over a duty cycle whose cases stay within the rating method, the Assistant's solve data still gives the nominal summary's modified life where the loads as entered put κ below 0.1 (the Overhung pulley's BR2 on 10 / 2.5 cSt oil at 150 °C nominal, 35 °C in its one case: 107 191 h), with no note that it is outside the method; the duty's life, which the check and every display go by, is right (a nominal summary in four-point contact is marked); - Show the calculation of a check that fails with an unbounded share says it fails "whatever the value of the checked quantity"; for an axial preload that lifts off (a cross-located pair whose housing runs hotter than its shaft) the share is of the lifted-off

state only, and the same check passes once the temperatures no longer lift it; the check's message gives the cause; - "Make it pass" on a check not determined still steers its search by the check's share as modelled (a proposal is made only where the check is determined and passes after the change), and its "tried" list may give that share as the closest reached; - the load-spectrum card's chart still plots a bearing's damage shares where its table withholds them (a case's life outside the rating method with the life check failing); read the table; - the report's summary page labels a passing hybrid's life "modified reference rating life" before the parenthesis that says it is the basic life its check goes by; the bearing section is right; - the agent guide says a hybrid's check `value` is its basic life; where the check fails (`outsideMethod`) it is the modified life it fails on; - a variant comparison of a steel and a hybrid bearing that both pass their life check leaves that row out of the default CLI comparison and the report, though the two columns go by different lives (the row's `labels` say which); `--all` shows it; - the Overview's utilisation chart plots the share of a check failing outside its rating method beside the others, ordered first among the failures but drawn as if comparable; the fail itself is shown; - a hybrid roller bearing on estimated geometry passes on the lower of its two methods' basic lives (correct), but the report's summary page names that life with the other method's label, the Assistant's basis text points at `basicLifeHours` (the picked method's, possibly the higher) instead of the check's value, and a check made not determined by the other method's lower basic life does not quote it; the bearing finder's `basicLifeHours` and the bearing page's basic life rows show the picked method's too; the check's value and status are right; A later release types every result figure as determined or not determined, and every display, ranking and short message reads that type: this closes the ranking, display and short-message limits above for good.

- An electric motor's critical speed and magnetic stability with a static eccentricity on coupled bearings (review of ut4-motor-critical-speed; see "From 0.3.0"). In 0.3.0: - A pull not along a bearing's other loads (more than 1° apart: a belt or a gear's force across the weight) leaves both not rated: the bearing's load turns as the eccentricity grows, and its stiffness along y or z can be least in between, which neither the states taken nor a bound with each bearing equally stiff every way at its least load bounded (a reviewer's case read 9 988 rpm where a state inside the bound gives 9 604 rpm). This blocks a belt- or gear-driven motor with an eccentricity from the coupled critical speed, and 0.3.0 names no way around it: no stiffness entered on the supports (with coupling off) is known to be safe, as the least one lies somewhere between centred and pulled (keep the eccentricity: its pull counts in the bearing loads and the air gap; 0 would drop it from them). - A state the coupling cannot settle exactly (bearings loose in their clearance, e.g. the Electric motor example without its wave spring, centred) leaves a passing result not decided (a warning). A motor whose bearings carry no load with the rotor centred (the weight off, a vertical motor without preload) has both not rated, with a hint (a horizontal motor: turn the weight on; a vertical one, or one held only by a preload, 0.3.0 cannot rate). The hints of the not decided and not determined results name no stiffness to enter: none is known to be safe there (review: a recipe of the bearings' stiffness without the pull could pass a motor that fails). - A coupled bearing under an axial load (a spring preload, as in the Electric motor example, or the shaft's own axial loads) or preloaded from within (an X or O pair's preload, a negative operating clearance) leaves a passing result not decided (a warning): its radial stiffness need not grow with its radial load, so the states taken bound nothing between them (a reviewer's case: e = 0.2 passed 15 000 rpm at 15 908 rpm where e = 0.15 gives 13 878 rpm). - A spring preload is refused on any shaft joined to another by a bearing between them (shafts solved together do not carry it), also where the spring would act on its own housing. - The unbalance response is the given eccentricity's, with a warning: at a smaller one it can be larger. - Up to four pulling rotors on a shaft; more have both not rated. The axial modes of a state keep the locating bearings' axial stiffness and the housings' seat motions of the pulled state. A critical speed or magnetic stability that is not rated fails, shown as "not rated" in place of its placeholder value (0 rpm, a share of 100 %), and What matters most leaves it out, saying why. Planned for a later release: the rotor pulled within its bearings' clearance to where they hold it (it never sits centred there), in place of the centred state; and a sweep of the eccentricity with a proven bound between its steps, for a pull across the bearings' loads, for bearings under an axial load (a spring preload) and for the unbalance response.

- Steering duty, for a later release (user test ut4): - A worm pair or a gear pair over a steering duty keeps its own required life (a worm pair rated at its worst state as if that lasted its life, a gear pair over the cases at its own life's cycles); the duty's length checks the ball nut's, the ball screws' and the bearings' lives. - Editing a mesh's Peak rack force for static rating in the duty's refusal card makes the duty not computed again, so the card (and the field in it) goes until Compute duty runs; with two refused meshes, the second is edited after recomputing, or on its mesh's own panel. - The drivetrain's natural frequencies (system modes) are dropped as out of date when another shaft is shown, though they do not depend on it; Compute them again. - While the duty computes, a check it decides reads "computing…" in every list of checks the app shows, also another report's (a version compared) of the same kind; the project's verdict line keeps the engine's count (a failure at the state counts) and says the duty is being computed. - A design search's work budget counts a remembered result of the steering duty at every request within one solve, so a design that runs on and on is stopped later than it could be (its time limit and Cancel still stop it); the engine's other remembered results (a gear tooth root's or a housing's finite elements) are not counted, which the budget's floor of 5000 steps covers.
- Cancel stops a housing FE solve between its steps, not inside the factorisation (the matrix library cannot be interrupted): on the largest model measured that is about 2 s on 4 threads (5 s on one), on a model near the 4 GB limit far longer (a later release).
- A steering's ball nut (`steering.system.assist.ballNut`) is no radial support of the rack itself (0.3.0): the rack needs a support there (a bushing with the nut bearing's stiffness, as the "Belt-driven ball-nut EPS" template has). The torque the nut passes into the rack's thread is not put on the rack (it is held against turning by the pinion's teeth and the yoke, whose ratings leave it out), and the nut's travel is not checked against the threaded length: keep the thread long enough for the nut over the rack's travel. On-centre, the feel's friction torque takes the preload's and the nut bearing's drags but not the motor's, which only the return force counts; it takes the nut's drag with the motor idle (the bearing's friction under that drag at the creep speed), so where the assist curve gives torque within the friction band the band comes out a little wide (a 1000 N preload on the template's nut, before its bearing's drag was counted: 0.601 N·m against 0.591 N·m solved at the band's edge). The steering's balance counts the friction of the bearings linked on the nut's shaft only (as a drag at the nut, see "From 0.3.0"): a bearing linked on another shaft of the steering's drive (the column, the pinion, the motor) is in the power loss but its friction is not in the driver's and the motor's torques, which come out low by it (no example links one). The nut bearing's friction in the balance is its rating's as the solve rates it (its housing's seats moved as the solve moves them); should the two differ, the solve warns at `steering.system.assist.ballNut`. A bearing linked on the nut's shaft whose outer ring sits in another shaft (a bearing between shafts) refuses the steering's state: its friction turns with the two shafts' relative speed and may drive the nut, which the balance does not hold; seat it in the housing, or unlink it to take `bearingDrag`. For a later release. Its thread is taken right-handed (a left-hand one only turns the nut the other way). Without a duty its screw is rated in the steering state alone, so its life is that state's; a steering duty with strokes rates it over the duty's states (0.3.0). The rack's misuse events leave the nut's force out: the meshes hold the event alone, on the safe side for their teeth and the bar between them and the joints, while the nut and its bearing are rated in the state only.

- A steering duty over a steering system (0.3.0) rates the rack meshes' teeth, and with strokes the system's shafts, the bearings linked on them that take the load cases, its worm pairs and its ball nut over its states (see Steering system). Not rated over the duty (a later release): a mounted gear pair (rated at the system's `state`, its checks not rated over the duty, never a pass: its rating takes its solved shafts' KHβ, which the states do not give it yet), and likewise fits, keys and housings. A rotor's air gap is not rated over the states (its band between them is not drawn; never a pass: rate it at the duty's heaviest state). A bearing's warning over the states (a minimum load undecided at its valley) is never satisfied, and its gyroscopic slip (the high-speed model's) is the system's state's, a pass not rated over the duty. Beside such a duty a design search's metrics are refused but the shaft's mass (their values over the states have no bound between them, and some items' results are the system's state's): search with requireChecks, whose checks take the bands. The ball nut's life is the states' Miner sum (as a bearing's), its static safety the worse of the states' and the system's own state's. Left at the system's `state`: the couplings, a bearing with `useLoadCases` false, and everything at a misuse event (the events rate the pinions only; the elements keep their own `peakFactor` for a static peak, which over the duty multiplies every state's loads, as over load cases). A worm pair and a drive take the worst of their states, each rated as if it lasted their life: on the safe side, as DIN 3996 holds no load spectrum here. A shaft's states are each rated with their stresses fully reversed (as a mounted ball screw's states are), on the safe side where a stroke does not turn round. The band of a bearing's checks is drawn from its life, static safety, contact stress and thermal speed in the states, relative to each check's utilisation. Two supports in seats of one housing (whose seats push each other with the bearings' loads) leave the elements unrated over the duty (a warning at `steering.duty`, their checks never a pass): the states do not couple the seats again (a later release). The project's own load cases beside such a duty rate the shafts and bearings instead (a warning at `steering.duty`), and their checks are then not rated over the duty (never a pass): remove the load cases to rate them over it. The elements' states are some four times the duty points (284 for the Dual-pinion EPS's duty), each a state solve and a solve of every shaft: at most 1200 (`system_cases::CASE_LIMIT`) and at most 12,000 solves (`WORK_LIMIT`: each state one, and one for each shaft and each bearing on a shaft; some 25 s natively and 40 s in the browser), past which the elements are not rated over the duty (a warning at `steering.duty`; their checks never pass) while the teeth are. Measured under load on the build machine (one process, natively): the Dual-pinion EPS's duty 9.3 s for the teeth and 19 s in all; the Belt-driven ball-nut and Column EPS examples given its duty 2.4 s and 2.8 s for the teeth, 7.0 s and 5.2 s in all (168 and 88 states); the browser's engine took about the same under that load (17.9 s, 7.6 s and 6.3 s). It refuses a stroke's `torque` table (the system splits the rack force, not a pinion's torque); a rack mesh off the system's rack is not rated over the duty (a warning; it keeps its rating at its own load). Each duty point is a state solve (some 15 to 30 ms natively), at most 4000 per duty. Rating the teeth of two pinions is the slow part: the Dual-pinion EPS template's duty (a parking, a manoeuvring and an on-centre stroke, both ways) takes some 4.7 s natively and 7.3 s in the browser's engine on the build machine (some 20 s natively on a laptop, 4 s of it the state solves); the app computes it on request, with progress and Cancel (0.3.0), and solves each edit without it. Positions past the rack's travel are refused (each point is a state of the system, within its stops). The lift is rated within 1e-4 mm of the states' between points (the contact ratios within some 4e-5), not raised; the forces are raised to bound theirs. Misuse events are rated at standstill (vehicle speed 0, forward efficiencies, no friction).

- A Nastran mesh is refused for a MAT1 of a density outside 500 to 25 000 kg/m³, or two MAT1 of different densities or stiffnesses, and for a GRID with its displacements in another coordinate system (CD), also when no solid element uses that MAT1 or GRID (a shell's material, a rigid link's node). On the safe side: the message names the card or says there are several materials; remove the unused cards, or export the solid part on its own. So is an SPC that moves its node (D not 0, an enforced displacement), and a file with several SPC sets and no case control choosing one: hold the node with D 0, or export the case control (SPC = n) with the mesh.
- A solid wall's hub whose rim lies past 80 % of the way from its bore to the plate's nearest edge (or to half way to the next bore's hub) is refused, with the largest diameter that fits: the mesh's ring between hub and wall lies there at most. On the safe side; a hub nearly as wide as the wall round it comes in a later release.
- Compare variants shows no housing results (a housing wall's stiffness, stresses or modes) for a later release (user test ut4): a variant that changes a housing is compared by its shafts', bearings' and checks' results only.
- A reduced housing matrix from a Nastran DMIG punch, an Abaqus matrix or a CSV states no count of its entries, so a file cut off between two entries cannot be told and reads with the missing ones as 0 (a stiffer housing); check that its last entry is there. Harwell–Boeing and Matrix Market files are held to their headers (see "From 0.3.0"). A reduced matrix is read up to 600 degrees of freedom: a Craig–Bampton reduction keeping more modes is refused (RR1 audit O-12).
- Housings: a seat a support, fit or rib sits in can't be moved by an edit, and putting a seat found on a bore onto that bore's own patch (**Seat it here**) counts as a move, so it is refused while something sits there; unlink it first, or seat it on the patch before linking (housing-032 review, O3).
- Housings, for a later release (user test RR1, gap round): a housing FE solve runs on the app's own thread, so the window stops answering for the 20–30 s of a large mesh's solve and Windows may say the app stopped responding (it finishes; wait). An imported mesh without a density (a STEP part, or a Nastran deck without RHO) is taken as steel, 7 850 kg/m³, beside the grey iron's E and ν the housing keeps, while the density's tooltip names 7 200. The app does not mesh an existing housing's STEP part again at another element size, and the size is not kept with the mesh.

Steering systems

Rack and pinion steering: the system’s forces and feel, the assist, and its duty from load cycles.

Methods and standards

The steering state is solved for the steering-wheel torque from the quasi-static balance on the rack: each source's torque times its gain (rack force per unit torque along the connection train, with each stage's efficiency for the power's direction) plus the assist motor's torque from its characteristic equals the rack force plus the rack's friction in the direction of motion. The characteristic is piecewise linear and never falls, so the balance is solved exactly segment by segment, the motor limit being one more segment.

Forward efficiencies apply when power flows to the rack, back-driving efficiencies (a worm's η') when the road drives the rack; at standstill the gear is solved creeping in the force's direction, without friction. The rack friction (yokes, bushings, seals) comes from the rack solved at the torques found, repeated to a fixed point with the efficiencies.

The torque sensor carries the driver's torque less the losses of the couplings before it; the column bearings' friction is not taken off. The column's universal joints pass the mean torque at a 1:1 mean speed; their speed fluctuation $1/\cos\beta - \cos\beta$ is reported, their secondary moments are not computed.

The feel numbers are quasi-static and linear about the rack's centre with $F_R \to 0$. Compliances are referred to the steering wheel by the square of the lossless speed ratio: shaft torsion $\Sigma\, L/(G \cdot Jp)$ between where the torque enters and leaves (round sections), the torsion bar $G \cdot \pi \cdot d^4/(32 \cdot L)$ or a coupling's given stiffness (couplings without one are rigid), each rack mesh $1/(c_{\gamma\alpha} \cdot b \cdot \cos^2\alpha_t)$ along the pinion's tangent with $c_{\gamma\alpha}$ of ISO 6336-1:2019 at $z_{n2} = \infty$, the pinion shaft's bending at the mesh under a unit force along the rack with its supports as modelled, and the rack bar $\Sigma\, L/(E \cdot A)$ from the driver's mesh to the held tie-rod ends, the tie rods and the housing in series.

With the assist, the compliances after the assist enters (C_{down}) carry its force too, G_a being the assist's rack force over the driver's per unit driver torque at the sensor torque's origin (0 inside a deadband): $K = 1/(C_{up} + C_{down} + G_a \cdot C_{cross})$, where C_{cross} is what they let the driver's side move per unit of the assist: C_{down} where the assist enters at the driver's mesh or above it; for an assist pinion elsewhere on the rack the bar's and tie rods' part is their cross-flexibility between the two meshes, $C_{L(lower)} \cdot C_{R(upper)}/C_{total}$ with both ends held. The feel numbers compare designs; vehicle on-centre metrics (torque against lateral acceleration) need a vehicle model and are not computed.

Each tooth has a contact window of width $\varepsilon_\gamma \cdot c/z_1$ centred on its displacement.

Every pass adds one cycle per loaded flank at its peak root and contact stresses after applying local lead and geometry; this is conservative.

Each member's root and contact peaks are selected separately, including load and lead corners, centre crossings and numerically refined interior extrema; the window is first cut where the dynamic factor changes regime (the resonance ratio crossing the main resonance range's lower edge, 1.15 or 1.5, or a line load of 100 N/mm, found exactly), so each refinement stays within one regime and a narrow resonance range is not missed.

Tooth zero is centred at displacement zero; variable lead is integrated to set tooth phase, and shifting the phase by half a pitch may change the count by one tooth.

Pinion tooth k collects rack teeth j with $j \equiv k \pmod{z_1}$.

Damage uses the linear Miner sum of ISO 6336-6 with $K_A = 1$, because the stroke tables already give the loads.

Each flank takes one cycle per pass: the raw root and contact stress peaks are reported, while damage and safety use the lowest local ratio of permissible stress to stress at each life knee, interpolated and extrapolated on the safe side; a restoring pass through the centre can load both flanks once.

The strokes are rated at the driving load, as the rack mesh's own rating takes its efficiency: the pinion drives the rack against the road, so a force table (the rack's force F) is rated at F/η, η the mesh's efficiency (0.85 by default), and a torque table (the pinion's torque, the driving load) as entered; a misuse event is the road driving the rack, at its rack force F. The derived cases scale the loads by the same driving load over the reference mesh's (its torque's rack force, or its rack force over η).

ISO 6336-3:2019 Annex B gives Y_M per tooth from the flanks' cubic-mean line loads; the printed 1 > R > 0 in B.1 is read as negative R.

misuse events are checked for root strength; their contact stress is reported, not checked: ISO 6336's static contact strength is a pitting limit, which a single event does not reach.

Duty contact safeties use only the bins with cycles; misuse adds no fatigue cycles.

With strokes, the duty replaces the mesh's own-load fatigue checks; the nominal rating remains a reference, and peak-load static and contact-ratio checks remain.

Contact-ratio requirements cover the nominal position, every stroke's endpoints and crossed lead corners (including the centre), and misuse positions, even at zero load.

Outside ISO's geometry range, duty and misuse root assessments use the larger of ISO and FE stress for each member; a missing FE assessment withholds the duty rating.

The project's load cases are parts of each stroke direction, cut at its ends, every corner of its load and lead tables, the zero crossings of a signed table and the centre under a restoring law, each span between in equal parts (about nine per direction, fewer when the cases would exceed 50).

Both ends of every part are cases, so every extreme of the steering force is one, whatever other loads a linked module adds.

Each end runs at its own pinion speed, so the speed extremes are cases too.

The cases of a part keep its exact time and exact pinion revolutions and exactly the chord bound's revolutions at each end's force, $(\bar{F} - F_L)/(F_H - F_L)$ of them at the larger force, $\bar{F}$ the part's mean rack force over its revolutions; where the two ends alone cannot, a third case carries one end's force at the other end's speed.

By the chord bound this is exact for a damage per revolution linear in the steering force and overestimates any convex one, such as a power of a resultant load affine in it, also one that falls as the steering force grows (gravity against it).

The cases also bound the ratings of every linked module: each part is sampled on an even grid, toward its ends and on both sides of every point where a linked gear pair's dynamic factor changes regime (the resonance ratio N crossing the main resonance range's lower edge N_S or 1.15, ISO 6336-1, found exactly from N and N_S, which are monotone in the load and the speed), and a part where a bearing's life, static safety or contact stress, a gear pair's contact or root stress, a ball screw's life or force or a shaft's fatigue or static safety is worse inside than at the part's own cases (by more than 1e-9, or 1e-8 for the iterated bearing and coupled shaft solutions) is split at the worst point and both halves get their own exact cases.

Their damage is bounded too: where a bearing's or ball screw's life or a gear pair's damage per hour summed over a part's cases falls below its integral over the part (five-point Gauss in each piece between the regime changes), as a damage concave in the speed does at the part's ends, a share of

the part's revolutions moves to its mean lead with the chord's force split, which keeps the time, the revolutions and the force distribution exact and the speed extremes as cases, and the share is the least that brings every such sum up to its integral; where no share does, the part is halved.

A gear pair's fatigue safety is the factor on its stresses at which its damage reaches 1, so its damage is bounded at that factor too: parts are also cut where a curve's scaled stress crosses a knee of its S–N curve, and where the cases' safety is above the factor at which the duty's own damage, integrated over the strokes between those cuts, reaches 1, the cases are bounded again at that factor (and refused if their safety is still above it).

Every linked module declares where its rating changes regime through one interface, as residuals that change sign there, each monotone in the load and the speed so that the search between a part's ends finds every crossing; the damage integrals split there, and each piece is refined adaptively against an error estimate (five-point Gauss on each interval against its halves, to 1e-6 of the integral, at most twelve halvings), which also rates each half just inside its ends, where neither rule has a point. The rule: every jump is declared, or, where it cannot be placed, removed by an upper envelope that is continuous and never below the rating, so that the cases can only count more damage; a bend (continuous, its slope changing) is left to the peak search, which converges onto it as onto any peak, and to the integrals' refinement, whose error estimate holds across it. A jump left to the error estimate is not enough: where every point of both rules falls on one side of it, they agree and the integral is wrong. Changes that depend on the load in ways the boundaries do not give make their module's checks marginal where its cases come near one, and a change the boundary search cannot locate (a residual close to zero over a stretch) leaves its module's checks undecided, marginal where they pass. The regime changes of the linked ratings (branches, limits, least or largest of several, table steps that depend on the load or the speed; the position enters only through them) are these.

Gear pair (ISO 6336-1, -2, -3 and -6 as implemented). Declared: the dynamic factor's regimes (the resonance ratio N against the main resonance range's lower edge N_S, a jump; 1.15, a jump with the 2019 edition's limit of 2 outside the main range, and 1.5, a bend; a line load of 100 N/mm, where N_S, c′ and cγ bend); the running-in allowances' limits at 5 and 10 m/s, where yα and yβ, so K_v, K_Hβ and K_Hα and the stresses, jump (for a gear that is not surface hardened; a hardened gear's limit is the same at every speed); and the S–N knees of each scaled stress (a bend, and a jump to no damage at a flat end); and where a pitting curve's slope above its static knee changes segment with the speed (that slope is its first falling segment's, and a segment stops falling where the running minimum makes it flat, its knee as high as an earlier one: the damage of a stress above the static knee grows without bound as the segment flattens, then takes the next segment's slope), as ln(σk/σj) for each pair of knees, monotone in the speed (each knee is σ_Hlim·Z_NT·(Z_L·ZV·Z_R·Z_W)^(1 − s)·Z_W,stat^s with s fixed by its cycles). Bends, which need no boundary: K_v held to 2 and to at least 1 (2019) and Bk = |1 − c′·C_a/(F_t/b)|; K_Hβ's Fβx (the favourable contact pattern's |1.33·B1·f_sh − fHβ5|, the largest of three, yβ's limit; its two branches meet with their slopes); K_Hα and K_Fα held between 1 and their limits; ZV's floor at 0.1 m/s; Z_W's roughness held to 3…16 and its reference to at least 1; the S–N curve's running minimum. Without load or speed dependence: Z_L, Z_R, Y_X, YR, Yδ, the life factors and the static share (taken at the knees' cycles), the factors cv1…cv7, the tip relief taken, K_Hβ without a shaft layout (an input).

Rack mesh (its own duty). The dynamic factor's regimes and the running-in allowances' limits are declared in its tooth-window search, and each window's peaks are found by golden section within one regime. With a variable lead the rack's local geometry moves with the position, and with it the contact ratios: ε_γ = 2, where K_v's coefficients cv1 to cv4 and cv6 jump (ISO 6336-1, 6.3.2), and ε_α = 1, where the root's load point and Y_ε change, are declared too, each lead segment split where the contact ratio peaks (it is unimodal there, so each piece's ratios are monotone in the position); the geometry's other branches are bends (cv7's ranges, K_Hα's at ε_γ = 2, Z_ε's and the single contact factor's at ε_β = 1), and its other branches are the gear pair's.

Bearing (ISO 281, ISO 16281:2025 as implemented). Every check its duty takes from its cases is bounded: its lives as damage rates, and as largest values its static safety, contact stress, load against its minimum load (as 1/(1 + load/minimum), bounded where the load passes zero; the axial load's holding, which changes with each case's load, is not evaluated in 0.3.0, so a light load fails;

and as a valley check, failing at the load's low point, which the cases bound from above only, a passing one is undecided over the steering duty in 0.3.0, unless a preload holds it in every case), speed over its thermally permissible speed and temperature (where a heat balance is hottest or fails). Declared, with the speed: ν1's two formulas at 1000 rpm (a jump), κ held to 0.1…4 and a_ISO's ranges at κ = 0.4 and 1 (bends), e_C's saturation (a bend); for a bearing that takes its loads as entered, without a heat balance or the high-speed model, with the load too: e_C·C_u/P = 5 (a jump under the original rules, a bend under the standard rules, which hold it at 5), a_ISO's bracket at 0 and its cap of 50 (bends). Marginal where its cases come near one: a bearing given its loads by a shaft or a planetary set (e_C·C_u/P within a factor of 2 of 5: its load need not be monotone in the rack force); one with a heat balance (κ within 25 % of a limit or range, ν·n within 25 % of Palmgren's 2000, where its idle friction, so its temperature, clearance and contact stress, jump, e_C·C_u/P within a factor of 2 of 5). Undecided wherever it is: a ball bearing with the high-speed model (its ball equilibrium, started from several points, may change solution, and its e_C·C_u/P is not monotone in the speed).

No jump: the load zone opening (element loads go as δ^1.5, the lives as δ^5); the most loaded contact moving to another element, ring or lamina (the largest of several: a bend that makes no peak of the contact stress or of 1/s_0); the ring held square (taken only where it is an equilibrium of the free ring too).

Enveloped: a_ISO per lamina, whose jumps at each lamina's e_C·C_u/p = 5 move with the lamina's own load, where no boundary can place them; each lamina holds e_C·C_u/p to 5 (where ISO 281's curves end; the standard rules everywhere, the original rules over steering-derived cases) instead of its a_ISO jumping to 50, continuous and never above the jump's, so the cases can only count more damage. Not rated over the strokes: ISO 281's X and Y step at F_a/F_r = e and ISO 76's X_0 and Y_0 (reported only), the minimum load (a check), zero load (only where the force is zero, a cut).

Ball screw (ISO 3408-5). Declared: the preloaded nut's lift-off at 2^(3/2)·0.6 of the preload (a bend of the life). No other: one step's equivalent load and speed are smooth, and standstill or no force on a nut without preload is no damage, its limit; its static safety, buckling and rigidity are not rated over the strokes.

Shaft strength (DIN 743, FKM). A case's fatigue and static safety are bounded as largest values, and their changes are bends: the resultant moment's zero, |σ| of the static check, DIN 743's σ_mv and the ranges of its amplitude strength, FKM's mean stress ranges (F1 and F2), the least over places; FKM's jump at S_m = 0 under a mean torque (eq. 2.4.20) is not taken over steering-derived cases, whose equivalent mean stress is taken whatever the sign of S_m (never less severe), and a specified assessment that covers the worst plain place leaves the next worst as the plain section (before 0.3.0 that case's plain section was lost, and its safety jumped where its worst place moved onto a covered one).

Each derived damage carries an error bound, the share by which the cases' damage may fall short of the strokes' own; a check that passes by less is marginal, a warning that the derivation cannot decide it.

A shaft's FKM damage over the strokes uses the elementary Miner rule, never below the consistent rule's, with each case's amplitude converted to R = −1 by the line of F2's range II (the larger of (1 − M)·S_a and S_a + M·S_m), which is never below F2's conversion and is convex in a load the stresses are linear in, where F2's bends the other way at S_m/S_a = 1 and 3; the cases at a part's ends then bound each stress type's power sum of it over the part (the chord bound) and its largest value, and the elementary result rises with both, so it bounds the strokes' own. A shaft coupled to its bearings is not linear in the load, and its damage check over the cases is not decided.

Only equal signed loads merge, also across strokes, where the case is named for how many strokes it covers; time shares are in hours and the load magnitude is the driving load's rack force over the first mesh's (see above).

Shaft loads and linked screw forces retain the stroke's sign through individual factors; gravity keeps its direction.

The rack speed sets pinion speed and stroke time; the bearing input speed is scaled in the same ratio as the reference mesh's speed.

Every rack mesh takes the full load of every stroke; no force split between pinions is modelled yet.

Each tooth has a contact window of width $\varepsilon_\gamma \cdot c/z_1$ centred on its displacement, at the mesh's own lift (the widest window: the rack's lift over the duty is never below it).

Every pass adds one cycle per loaded flank at its peak root and contact stresses after applying local lead, lift and geometry; this is conservative.

Each member's root and contact peaks are selected separately, including the corners of each pinion's loads, centre crossings and numerically refined interior extrema; the window is first cut where the dynamic factor changes regime (the resonance ratio crossing the main resonance range's lower edge, 1.15 or 1.5, or a line load of 100 N/mm, found exactly), so each refinement stays within one regime and a narrow resonance range is not missed.

Tooth zero is centred at displacement zero; variable lead is integrated to set tooth phase, and shifting the phase by half a pitch may change the count by one tooth.

Pinion tooth k collects rack teeth j with $j \equiv k \pmod{z_1}$. The pinions on the rack share the rack teeth their zones both reach, and the cycles add there.

The duty is rated over the steering system: each duty point is a state of its balance, with the rack at the point's displacement, the stroke's rack force on it, the rack moving at the stroke's speed in the pass's direction and the vehicle at the stroke's speed (0 when not given, parking). The balance splits the force between the driver's and the motor's paths by the assist characteristic, the variable ratio and each stage's efficiency in the power's direction, with the rack's friction (yokes, bushings, seals) added; the train gives each pinion its force on the rack, and the rack solved under them each pinion's lift at its contact (its yoke seated, lifting or on its stop).

A pinion on a yoke is held against the rack (no lift, double flank contact) wherever its yoke stays seated, its design regime at light loads; the ratings there say so, and it is not repeated as a warning.

Each pinion's teeth are rated at the driving member's load as a lossless mesh passes it, as its own rating is: its force on the rack over η where its pinion drives the rack, the force itself where the road drives the rack back (a restoring stroke's return, η''s direction).

Between duty points each pinion's force and lift are linear. The points are the stroke's ends, its table's corners, a restoring stroke's centre from either side and every pinion's lead corners, and each span between is halved until its chord meets the states solved at its quarter points within 0.05 % of the pinion's largest force in the pass and 5e-5 mm of lift. Each state also carries the levels its regimes change at, where the loads bend: the sensor torque at the assist characteristic's corners (the motor's limit and the sensor's stop among them), each yoke's reaction at its preload and at its stop, the meshes' force along the rack's motion at 0 (the power turns). A span is accepted only where no change lies within reach of a level (its range at the points widened by twice its chord's largest miss there) that the span's ends do not straddle, and at most one is crossed. The chord's miss over such a span is at most twice its largest at those points for a load that is linear, quadratic, has that one bend or one curvature over the span; each pinion's forces are raised by twice the pass's largest sampled miss (at most 0.1 % of its largest force), so its teeth are never rated below its share, and its lift is within twice its largest sampled miss (at most 1e-4 mm). These bounds hold for loads and levels of one curvature over a span, which the sampling cannot prove, so a duty check that passes by less than 0.1 % is marginal, never a plain pass.

A misuse event is a state at standstill (forward efficiencies, no friction, vehicle speed 0); each pinion's static root check takes its force on the rack there, as a standalone mesh's takes the event's force.

Damage uses the linear Miner sum of ISO 6336-6 with $K_A = 1$, because the strokes give the loads.

Each flank takes one cycle per pass: the raw root and contact stress peaks are reported, while damage and safety use the lowest local ratio of permissible stress to stress at each life knee, interpolated and extrapolated on the safe side; a restoring pass through the centre can load both flanks once.

ISO 6336-3:2019 Annex B gives Y_M per tooth from the flanks' cubic-mean line loads; the printed 1 > R > 0 in B.1 is read as negative R.

misuse events are checked for root strength; their contact stress is reported, not checked: ISO 6336's static contact strength is a pitting limit, which a single event does not reach.

Duty contact safeties use only the bins with cycles; misuse adds no fatigue cycles.

With strokes, the duty replaces the mesh's own-load fatigue checks; the nominal rating remains a reference, and peak-load static and contact-ratio checks remain.

Contact-ratio requirements cover the nominal position, every stroke's endpoints and crossed lead corners (including the centre), and misuse positions, even at zero load, at the largest lift of the duty.

Outside ISO's geometry range, duty and misuse root assessments use the larger of ISO and FE stress for each member; a missing FE assessment withholds the duty rating.

With strokes the duty also rates the system's shafts (the rack bar among them), the bearings linked on them that take the load cases, its worm pairs and its ball nut over its states, as over load cases. Each span between two neighbouring duty points of a pass gives five states, its ends and its quarter points, each the project as that state loads it; each quarter of a span takes its time (its length over the rack speed, times the stroke's count), half to each of its ends. A check that takes its worst case (a DIN 743 or FKM safety without a number of cycles, a stress, a deflection, buckling, the critical speed) takes the worst of the states and the system's state; a shaft's stresses in each state are taken as fully reversed (the loads turn round between a left and a right turn, a stroke out and back), which bounds any cycle between the states. A bearing's life is the Miner sum over the states by their shares (ISO 281), FKM's damage over a number of cycles the elementary Miner sum of the states by their revolutions, and the ball nut's screw is rated over the states as its load steps (ISO 3408-5); a worm pair and a drive take the worst of their ratings in the states, each as if it lasted their life (on the safe side for their damage). A mounted gear pair is rated at the system's state and not over the duty: its checks are not decided.

Load cases of the project's own rate the shafts, their bearings, the worm pairs and the ball nut instead, with a warning at the duty, and their checks are not rated over the duty, never a pass.

A support that names a bearing by its designation only links none: nothing rates it, at the system's state or over the duty, and the duty lists it as not rated (the steering examples draw their pinions' bearings so); link a bearing there to rate it.

A check's utilisation between the states follows its chord to within twice its largest miss at a span's quarter points where it is linear, quadratic, has one bend or one curvature over the span, and its mean (a life's) to within 2.75 times that: a check that passes by less than three times its largest miss is marginal, never a plain pass.

The misuse events rate the pinions only.

Validation

6 of 6 cases within their tolerance, in 2 groups.

Closed-form solutions

Ball nut on the rack: the rack force that turns it back

Kind of reference	Closed-form solutions
Implements	—
Reference source	The closed form F = 2000·π·T/(P_h·η′) for 0.3 N·m at a 7 mm lead and η′ 0.936795
Cases	1 of 1 within tolerance

Case	Reference	Axomek	Deviation	Tolerance	
Return force	287.4474 N	287.4474 N	5.1419×10^{-8} N	1×10^{-5} N	**pass**

Steering column and assist: torsion, the balance of driver and assist, universal joints

Kind of reference	Closed-form solutions
Implements	—
Reference source	Closed forms of mechanics: twist ΣL/(G·J) of a stepped shaft, a torsion bar's G·Ip/L and shear, the balance T_d = F/(A_d + k·s·A_a) of a linear assist characteristic, a universal joint's fluctuation 1/cos β − cos β
Cases	5 of 5 within tolerance

Mechanics only; the steering system as a whole has no published worked example to compare with.

Case	Reference	Axomek	Deviation	Tolerance	
Linear assist, rack force 7 kN: driver torque F/(A_d + k·s·A_a)	4.830918 N·m	4.830918 N·m	1.8×10^{-16}	10^{-12}	**pass**
Stepped shaft (Ø20, Ø30/10, Ø16): compliance ΣL/(G·J) over 230 mm	1.0942×10^{-7} rad/(N·mm)	1.0942×10^{-7} rad/(N·mm)	0 %	10^{-12}	**pass**
Torsion bar d 4.2, L 22 mm: stiffness G·π·d⁴/(32·L)	111087.3 N·mm/rad	111087.3 N·mm/rad	0 %	10^{-12}	**pass**
Torsion bar at its 3.5° stop: shear 16·T/(π·d³)	466.4789 MPa	466.4789 MPa	0 %	10^{-12}	**pass**
Universal joint at 22°: fluctuation 1/cos β − cos β	0.1513509	0.1513509	0 %	10^{-12}	**pass**

Not validated against a published reference

- No published worked steering system: only the closed forms of the column, the assist balance and the ball nut are validated
- The steering duty and the load cases it derives

Stated limits

The help states no separate limits for this chapter; see the method notes above and the known limits.

Known limits

- Units (R-387, R-931): the app and axomek read the same units typed, with the same factors, and show the same US customary units; a test holds them to it (`units-parity.spec.ts`), also for a field shown in another unit than its value's. It lists what still differs, on purpose or refused (never read wrongly): a design search's bound on a deflection is typed in µm (mil) in the app, as its check shows it, and in mm (in) by axomek `optimize --require`, which refuses nm there; a bolt's expansion coefficient α_S is typed in 10^{-6}/K in the app, in 1/K in axomek. A bare number in a bearing clearance is in µm in the app (the unit it shows) and in mm in axomek without `--units imperial` (base units, as every bare argument; mil with it, as in the app, but mm again with `--json`, which always reads base units). The bolt's expansion field refuses its own unit typed out ($12\ 10^{-6}$/K reads as two numbers): type `12e-6 1/K`, or a bare 12 with metric units shown (with US units shown a bare number is in 10^{-6}/°F). The text output and the report keep a few units metric that the app shows in US units (m, J, GPa, √MPa, mm³, a steering's feel: `SCREEN_ONLY` in `crates/axm-model/src/units.rs`). Not compared: fields the app shows in % of a plain fraction (a torque amplitude, a modulus share), and the change lists of the Assistant and axomek `diff`, which give a clearance in mm.
- On the steering's ball nut shaft, the friction of the nut's linked bearings shows in the power flow as power leaving that shaft (19 W in the Ball-nut EPS example): the nut's stage passes on what reaches its thread. The totals and efficiencies hold it once, as the report's power loss does.

- A result not determined is hidden display by display, not by one shared rule: in 0.3.0 the displays reviewed (the report, Show the calculation, the CLI's text, the comparisons, the Overview, the cards, the modules' tiles and the Assistant's digest) say "not determined". Those found last are refused instead: the Changes dialog's and Compare projects' check rows and the Overview's key results of keys and bolts say "not determined (see the check)"; the assembly view gives no share of the limit for a check not determined or not computed; and in the report an element's results, and a key's or a fit's results its checks go by (safeties, capacities, permissible values, pressures, shear, the fit's Turning row; with `axomek solve`'s key and fit lines), say "not determined (see the checks)" as soon as one of its checks is not determined, determined ones with them; a key's page refuses its flank pressures and tooth shear stress the same way (its permissible values and capacity, which do not depend on its rating, stay). A bolted joint with a check not determined (beyond VDI 2230's limit G, a duty not computed) has its whole Results block refused the same way in the report's section and in `axomek solve`'s line (its tightening torque, permissible assembly preload, resiliences and embedding with it, which the bolt page still shows); its page refuses its preload needed, load factor, clamp loads, stresses, opening and the joint diagram, but the notes under them, on its page and under the report's section, still quote the eccentric model's figures (M_Sb/W_b, F_Sm/F_0.2min, F_K against F_Kab). Make it pass and `axomek fix` give such a check no value either. A module's page, once one of an item's checks is not determined, shows no tile's caption (it may quote another check's figure) and refuses its tiles and Details tied to no check (a worm pair's load value C / Clim), and the Overview refuses the item's tiles tied to no check the same way, including ones that do not depend on the rating: a worm pair's method, centre distance and ratio, a motor's power and magnetic stiffness read "not determined" too; a later release shows those again. A design search meets no requirement or objective on a result of a design with any check not determined (a requirement's own check may be cleared, and the system life and power loss take every item's), its dimensions apart, nor on a fit's yield safety resting on an estimated yield strength, even where that figure itself would stand: the safe side; a later release decides each figure on its own. A bearing whose life the engine puts outside its rating method (κ below 0.1 where it turns, balls in four-point contact, a hybrid, whose a_ISO is defined for steel only) has all its lives, basic and modified, meet no search requirement or objective, as a minimum, a maximum or an objective, its minimum life cleared or not, and so has the system life with such a member; it says "not determined" without a life check too. The safe side: a hybrid's basic life, or one at a low κ, would stand; in a later release a search requirement on a figure whose own check is cleared is evaluated as that check, each life on its own. A design with a bearing the engine marks outside what it rates (a designation that contradicts it, a slewing ring's proportions) takes no calculated result in a search, its dimensions apart but that bearing's own (inferred from its geometry as modelled), as a design with a check not determined does, whatever its own checks say. A bearing whose designation contradicts it shows only its results' warnings it can vouch for: those at its own path, of the bearing as modelled, are left out (a validation warning at that same path too, such as that its geometry is estimated), its fields' warnings stay. A check not determined is said in a short form wherever its message would be rewritten: what it checks, named from its id, "not determined" and its kind's reason in fixed words, else "(see the check)", never a part of its message, which can hold a name the user wrote and the figures after it; a name that holds an engine mark's words can still pick another kind's reason (no figures). The full engine message, with the reason in context, comes back in a later release, when every display shares one rule. The bearing module's speed chart is not drawn while the bearing's operating nθ is not determined (a rotor's pull, a designation its geometry contradicts), saying why; `axomek speed-limits` then lists the curves' samples without an

operating point. A later release draws the curves apart from that load. A display not yet reviewed may still give such a number beside its check, which says "not determined". A later release routes every display of a check's value through one rule, with a test over every template. Known in 0.3.0, closed in a later release by that one shared display rule: - the Assistant's solve data still carries a check's value and utilisation where the check is not determined (a bearing's thermal speed whose nθ is withheld: its speed and share of the limit, from which the withheld nθ can be worked out); the check is flagged not determined there, the solve tool tells the Assistant never to quote a figure so flagged, and the check's row and every display say "not determined", never a pass; - a rack pinion shaft not rated at its mesh's peak because its only drive is solved for one power direction (the Dual-pinion EPS example's SH5, driven by worm pair WG1) is said, in the verdict's and the report's short reason, to be refused because its torque sources cannot be shared; its check's own note gives the right cause; - a design search takes no result from a design with any check not determined, so a project whose steering duty leaves a warning over its states may find no design meeting a shaft or bearing requirement (the safe side; a later release decides each figure on its own); - `Life0utside::against_bounds` keeps a maximum branch no caller uses (the search reads the hybrid flag directly); a later release removes it; - the Overview's governing check of a module, between two checks not determined, keeps the first in the list (the bearing listed first) where the engine picks by id, so the module tile and the verdict can name different checks, both said not determined; - a variant comparison can mark as best the life, and its check's margin, of a bearing the engine marks unrated (its designation contradicts its geometry), though the bearing's own results say its figures are of the bearing as modelled: compare such a design only after its designation is cleared or corrected; - a comparison of three or more designs can leave out a result row whose neighbouring columns each agree within the tolerance while the first and last do not (10 000, 10 000.8 and 10 001.6 h); `--all` shows it; - a comparison of designs whose life checks all pass on a hybrid's basic life (its modified life outside its rating method, so no best) leaves that check row out of the default CLI comparison and the report, though the minimum lives and the checks' shares differ (a hybrid 6309 at a 1 million h minimum against 4 million h); `--all` shows it. - a hybrid whose modified life fails its minimum keeps that failing check's number in its check row (the fail stands, on the safe side; the number rests on a_ISO, defined for steel bearings only), and so does any life check failing outside its rating method (a steel bearing turning below κ = 0.1, or with its balls in four-point contact): the report's Summary of checks and its Summary page's Governing checks show its share and draw its utilisation bar like any other (a 30207 on 2 / 1 cSt oil against a 20 000 h minimum), though that share cannot be compared with another check's; the governing pick and the orders leave it unranked, and the typed results' report follow-up shows no bar for it. Show the calculation of a hybrid's life explains the modified life, said not determined, not the basic life its passing check goes by; - the Assistant's shortened record of an earlier solve lists a failing life check with its value (a hybrid QJ 207 in four-point contact: 94 476 h against a 1 000 000 h minimum) but without the check's outside-method flag or reason, though the same record says the bearing's life is not determined; the fail stands, and the full solve the Assistant reads carries the flag; - over a duty cycle whose cases stay within the rating method, the Assistant's solve data still gives the nominal summary's modified life where the loads as entered put κ below 0.1 (the Overhung pulley's BR2 on 10 / 2.5 cSt oil at 150 °C nominal, 35 °C in its one case: 107 191 h), with no note that it is outside the method; the duty's life, which the check and every display go by, is right (a nominal summary in four-point contact is marked); - Show the calculation of a check that fails with an unbounded share says it fails "whatever the value of the checked quantity"; for an axial preload that lifts off (a cross-located pair whose housing runs hotter than its shaft) the share is of the lifted-off

state only, and the same check passes once the temperatures no longer lift it; the check's message gives the cause; - "Make it pass" on a check not determined still steers its search by the check's share as modelled (a proposal is made only where the check is determined and passes after the change), and its "tried" list may give that share as the closest reached; - the load-spectrum card's chart still plots a bearing's damage shares where its table withholds them (a case's life outside the rating method with the life check failing); read the table; - the report's summary page labels a passing hybrid's life "modified reference rating life" before the parenthesis that says it is the basic life its check goes by; the bearing section is right; - the agent guide says a hybrid's check `value` is its basic life; where the check fails (`outsideMethod`) it is the modified life it fails on; - a variant comparison of a steel and a hybrid bearing that both pass their life check leaves that row out of the default CLI comparison and the report, though the two columns go by different lives (the row's `labels` say which); `--all` shows it; - the Overview's utilisation chart plots the share of a check failing outside its rating method beside the others, ordered first among the failures but drawn as if comparable; the fail itself is shown; - a hybrid roller bearing on estimated geometry passes on the lower of its two methods' basic lives (correct), but the report's summary page names that life with the other method's label, the Assistant's basis text points at `basicLifeHours` (the picked method's, possibly the higher) instead of the check's value, and a check made not determined by the other method's lower basic life does not quote it; the bearing finder's `basicLifeHours` and the bearing page's basic life rows show the picked method's too; the check's value and status are right; A later release types every result figure as determined or not determined, and every display, ranking and short message reads that type: this closes the ranking, display and short-message limits above for good.

- `axomek check --strict` and `axomek solve --strict` exit 3 for every file with something its method does not assess, which no input clears yet (4 where a check fails and the file was solved completely; a file that was not, an unknown field under `--strict` among it, exits 3 first): a rated key or spline (reversals and load peaks), a rated fit (its shaft seat), a belt or chain drive (its capacity), a fatigue-rated weld (its toes), a yoke on its stop, a support with a designation only and a planetary set without a planet bearing. Check those by hand and use `axomek check` without `--strict` for such files (it still exits 4 on a failing check and 3 on one not determined); a way to mark an item as checked by hand is for a later release.
- Steering duty, for a later release (user test ut4): - A worm pair or a gear pair over a steering duty keeps its own required life (a worm pair rated at its worst state as if that lasted its life, a gear pair over the cases at its own life's cycles); the duty's length checks the ball nut's, the ball screws' and the bearings' lives. - Editing a mesh's Peak rack force for static rating in the duty's refusal card makes the duty not computed again, so the card (and the field in it) goes until Compute duty runs; with two refused meshes, the second is edited after recomputing, or on its mesh's own panel. - The drivetrain's natural frequencies (system modes) are dropped as out of date when another shaft is shown, though they do not depend on it; Compute them again. - While the duty computes, a check it decides reads "computing…" in every list of checks the app shows, also another report's (a version compared) of the same kind; the project's verdict line keeps the engine's count (a failure at the state counts) and says the duty is being computed. - A design search's work budget counts a remembered result of the steering duty at every request within one solve, so a design that runs on and on is stopped later than it could be (its time limit and Cancel still stop it); the engine's other remembered results (a gear tooth root's or a housing's finite elements) are not counted, which the budget's floor of 5000 steps covers.

- The assistant and the steering duty computed on request, for a later release (reviews of rr1-assistant-edits): whether the results include it is read against what is shown, so a duty that came in while the assistant answered is not noted under its reply when a preview is shown at the end (the next message is told). A later release should read it against the committed project.
- Steering misuse events (user test ut4): the misuse events rate the pinions' tooth roots (their contact stress is reported, not checked) and the pinions' shafts statically; the ball nut and its bearing are rated in the steering state and over the duty's strokes, not at a misuse event's peak, and the elements keep their own peak factor for a static peak. A worm pair that drives an assist pinion (a plastic wheel too), the other shafts and the bearings are not rated at a misuse event either, and only the report's method notes say so: the results do not list them as not checked for it. A mesh's peak rack force for static rating is a case of its own, the whole force through that pinion with the assist failed, rated for root and contact, so the same kerb strike can pass as a misuse event and fail there; the checks do not name it as the assist-failed case. Both err on the safe side or are disclosed; listing them on screen is for a later release (RR1 U-A25).
- A shaft's DIN 743 fatigue check over a steering duty is an endurance check at the duty's worst state, however few times it occurs: a shaft loaded hardest only when parking at the lock (60 000 strokes in user test RR1, SH5 at 1.162 against 1.2) fails it where a finite life could pass. On the safe side; a finite-life shaft rating over the strokes is for a later release (RR1 U-A26).
- Keyboard and screen readers, for a later release (user tests ut3 and RR1): the app's screen reader checks read Windows' UI Automation from a real Edge window (`npm run smoke:screen-reader`, on demand, not in CI) and the browser's accessibility tree; no run with NVDA or JAWS themselves has been made, and a check by a person with one is still advised. A check's rule written as a formula in its name (a bearing's minimum load, "k_r 0.025·(ν·n/1000)^(2/3)·(d_m/100)²") is read out symbol by symbol. A press on a fix button while a list shows an option not yet applied applies the list first, and the click is lost: press again once the suggestions are solved. A chart's "Show the values as a table" button shows only with the keyboard focus on it or the pointer over the chart (always shown only where the main pointer is touch: on a touchscreen laptop whose main pointer is a trackpad it starts hidden), so voice control that numbers what is shown may skip it on a computer with a mouse. The edges between the columns have no single-pointer way to resize them other than dragging (the keyboard has its own; WCAG 2.2 2.5.7 asks for one), and their 24 px targets reach from the border into the drawing's and the results' left side, so a control within 24 px of that border is pressed by its edge instead. On a window too narrow for both chosen widths, a column's edge does not go wider than it is shown (narrow the other column first). With a part shown in the Shaft view's results column, the inputs' edge reckons with the results width chosen rather than the part's, so it may stop short of the room there is, and a drag on a window too narrow for both may change the results width kept. A refusal already there as the Shaft page opens (a shaft without supports solved on the Assembly page first) is in the page's status region from the start, so a screen reader need not say it on its own: it is read with the results.

- While an edit solves, the app shows the results of the state before it (another project's never). They are dimmed and marked "Updating…" in the results columns, the drawings and their cards, the Steering and Assembly pages, and a balancing load's torque. Some places are not dimmed: the status marks in a module's list of parts, the figures an editor shows (a linked bearing's solved loads, the shaft's strength panel), the dialogs and the report. The header's "Updating results…" covers those. A solve that lands within 0.15 s is not marked at all, so that the page does not flicker (R-943). The system modes and the steering duty, computed on request, follow the inputs rather than the project: after another project opens they say "Out of date" until computed for it (never counted as computed), and a steering duty computed for a project of the same inputs counts for it. After a licence change while the Overview is open, its power flow can show the plan before's figures unmarked for as long as its own job runs.
- A teeth end (`teethEnd`) is rated by DIN 743-2's notch factors of a shaft with involute teeth without a hub (4.2.2) and an eq. (9) shoulder or groove of its shape, not by a published method for a hob's run-out, which no source at hand gives (see "Shaft strength references"); the mesh's own load on the teeth is the rack mesh's or gear pair's tooth-root rating. A worm's thread end takes none (a strength notch instead). The Shaft module's 3D draws a rack pinion's teeth cut into the shaft; a gear pair's pinion cut into its shaft is drawn there at its root diameter as before (the Assembly and Steering views draw its teeth). The 3D views draw a single toothed section's ends; teeth over several sections at the root diameter are rated over all of them but drawn on the one at the mesh.
- A ball screw on its shaft (`screws[].mounting`) loads it with the nut at one place (`mounting.position`), not along its stroke; its end fixity is checked against the bearings' places and axial roles, not their tilt stiffness, and its unsupported length is not compared with the bearing span. On the safe side, as the screw's states rate the shaft: a load step is taken as driving (the larger torque), and the screw shaft's fatigue (and that of every shaft joined to it) takes its whole torque and axial force as fully reversed over two or more cases even where they never turn round. Refused rather than combined: a mounted screw with a motion profile (its ramps' bearing damage needs the life integrated over the speed), one with steps of its own in a project with load cases or steering strokes, and a second screw with steps of its own beside one whose steps are the cases (a later release).
- A project saved before the driving side (`drivingSide`) reads as left-hand drive whatever its gear: the field is the side the geometry is said to be, and switching mirrors from it. Copies of the Steering rack, Column EPS and Dual-pinion EPS examples saved before had the right-hand drive gear's mirror image (their column ran forwards, to −z), so they read left while their gear is not. Tell the side from the gear: its steering pinion towards the rack's left end (x = 0) with its column rising rearwards (+z) is left-hand drive. The switch also changes the inputs checker remarks are about, which then read as outdated, as after any edit.
- A column-EPS gear's input switches from the column's torque to a rack force (`setGearInput rackForce`, the Steering module's Column EPS gear page), but not back: `torque` is refused with what to do (remove the steering system, put the torque on the pinion shaft by hand). Carrying a rack-force state back exactly (its rack friction as fixed loads, which a later switch, the static peak and the misuse states must not take again) is not done yet. The switch is refused either way in a project with design variants, whose designs would each need their own state (a later release).
- The same switch is refused where the rack mesh takes its torque from the train and the rack back-drives a pinion mounted to drive it (the train's power flows from the rack): the state would take the mounting's sense. Set the steering system by hand with its state's rack force and rack speed signed as the rack is loaded and moves (a later release).

- A column torque of 0 on a Column EPS gear's pinion shaft hides the Column torque field in the Steering module (its load is still there): enter another torque in the Shaft module, or undo (a later release).
- A steering's ball nut (`steering.system.assist.ballNut`) is no radial support of the rack itself (0.3.0): the rack needs a support there (a bushing with the nut bearing's stiffness, as the "Belt-driven ball-nut EPS" template has). The torque the nut passes into the rack's thread is not put on the rack (it is held against turning by the pinion's teeth and the yoke, whose ratings leave it out), and the nut's travel is not checked against the threaded length: keep the thread long enough for the nut over the rack's travel. On-centre, the feel's friction torque takes the preload's and the nut bearing's drags but not the motor's, which only the return force counts; it takes the nut's drag with the motor idle (the bearing's friction under that drag at the creep speed), so where the assist curve gives torque within the friction band the band comes out a little wide (a 1000 N preload on the template's nut, before its bearing's drag was counted: 0.601 N·m against 0.591 N·m solved at the band's edge). The steering's balance counts the friction of the bearings linked on the nut's shaft only (as a drag at the nut, see "From 0.3.0"): a bearing linked on another shaft of the steering's drive (the column, the pinion, the motor) is in the power loss but its friction is not in the driver's and the motor's torques, which come out low by it (no example links one). The nut bearing's friction in the balance is its rating's as the solve rates it (its housing's seats moved as the solve moves them); should the two differ, the solve warns at `steering.system.assist.ballNut`. A bearing linked on the nut's shaft whose outer ring sits in another shaft (a bearing between shafts) refuses the steering's state: its friction turns with the two shafts' relative speed and may drive the nut, which the balance does not hold; seat it in the housing, or unlink it to take `bearingDrag`. For a later release. Its thread is taken right-handed (a left-hand one only turns the nut the other way). Without a duty its screw is rated in the steering state alone, so its life is that state's; a steering duty with strokes rates it over the duty's states (0.3.0). The rack's misuse events leave the nut's force out: the meshes hold the event alone, on the safe side for their teeth and the bar between them and the joints, while the nut and its bearing are rated in the state only.
- Modules left out of the verdict (`report.contents.exclude`), until a later release, all on the side of saying more: a verdict whose kept checks pass close to their limits reads `partial`, not `close`; a torsion bar's "not checked" line and the first shaft's issues at `settings.*` or `rack.*` stay in the verdict when steering or the shaft is left out; a module switch's issue count is that module's alone, which can differ from the verdict's while others are off; `axomek variant compare` and the design search (`axomek optimize`) count the checks left out; a matched pair's warning (`pairs.*`) stays in the verdict when its bearing is left out; `axomek tree`'s root has no partial state (the nodes left out say `leftOut`).
- A plastic worm wheel's flank band rates the coolest temperatures at the whole life's load cycles, which errs on the safe side for a wheel that warms up within minutes. The data start at 20 °C, so a colder start leaves the flank not determined rather than rated (a cold-start rating below 20 °C, from moduli and flank strengths measured colder, is for a later release; the Column EPS assist gear example shows it), and with the lowest wheel temperature left empty the band starts at 20 °C: a colder start is then not looked at. A given wheel modulus E_2 holds at ϑ_M only, so its flank is not determined unless the lowest wheel temperature is given equal to ϑ_M.

- A steering duty's work is limited, so that neither its derivation nor its rating can run without end (0.3.0). Bounding the linked modules over the strokes stops at a fixed amount of work (the setup that ran for hours stops in some 35 s natively; every derivation the property tests' random linked setups and the steering tests decide takes at most some 20 s; the limits allow at most some 2.5 min, for a gear pair whose true-safety search is slow throughout): past it the load cases are the strokes' own, without the linked modules' refinement, every linked module's checks over them are undecided (never a plain pass) and `steering.duty` warns why and what to change. The tooth-by-tooth rating stops at its own limit: past it the duty is refused at `steering.duty`, saying what to change, as a search cut short would understate the teeth's damage. A duty on a constant lead, or with its tooth geometry within ISO 6336's validated range, rates in a few seconds (the steering-duty template in 2.7 s natively). Only a duty with FE root assessments (the geometry outside ISO's range) over a variable lead is slow: each new lead needs its own assessment (some 130 ms), so its rating can take some 52 s natively before the limit, and with its derivation a solve of such a duty up to about 90 s natively, longer in the browser; a 0.1 mm window there takes some 22 s. To make it fast, use a constant lead or bring the tooth geometry within ISO's range.
- The app's computed steering duty belongs to the project object it was computed for: showing another shaft (which orders the project anew without editing it) makes it out of date, and a run under way is cancelled (0.3.0). On the safe side: Recompute duty, or go back to the shaft shown. Keying it by the project's content comes in a later release.
- Where a rack mesh is given its pinion torque and no efficiency (its own rating then takes 0.92), the steering duty rates its force tables at the default for a known rack force, 0.85: a stroke the duty card starts at the mesh's rated rack force is rated 0.92/0.85, 8 %, above the mesh's own driving load, and linked items whose loads follow the rack's actual force (a ball screw) are scaled 8 % above it too. On the safe side; exact where the mesh's efficiency is entered.
- The steering duty rates every stroke's force table as the pinion driving the rack (F/η), also a restoring stroke's and a return pass, where the road may drive the rack and η''s direction would rate the teeth at F and the pinion at $F \cdot r \cdot \eta'$: on the safe side. Misuse events are rated at F.
- Over steering-derived load cases a shaft's FKM damage check (with a number of cycles) is undecided where it passes when the shaft's loads follow the speed (0.3.0): every shaft of a mounted stage (a gear or worm pair, a rack mesh, a drive), whose mesh efficiency is rated at each case's speed and enters a member's torque, and whose belt's pull rises with the square of the speed, and every shaft joined to one by a coupling or a bearing between shafts. The cases' chord split bounds that damage only where the stresses follow the force alone. In belt-drive with SH1's FKM over 800000 cycles and one-way strokes over a 126–130 mm/rev lead, the cases passed 6.681 at 6.681179 where the strokes give 6.680947.
- A steering duty whose strokes reverse the rack force (a return pass against a resisting load, a signed load below zero) is refused at `steering.duty` beside anything the load cases reach that a case cannot reverse (0.3.0): a mounted gear or worm pair or rack mesh, a drive, a coupling, a rotor, a bearing with loads of its own, a gear pair rated over the cases. The derived cases reverse only the entered shaft loads and ball screw forces, so the others would be rated at the forward loads on the reversed passes (a rotor's torque, unreversed, cancelled the reversed input in one review's case: 18.444302 where 9.222151 is right). In gearbox-single-stage with a rack mesh and resisting strokes with return passes, BR1 passed 5.5 at 5.778100 where the reversed pass gives 5.260948. One-way strokes are rated; reversing stages over the duty come in a later release.

- Over steering-derived load cases a linked gear pair is undecided where it passes (0.3.0) when its KHβ comes from its solved shafts (`misalignmentFromShafts`), or when the connections resolve the project (a shaft speed, a train-mode stage or a coupling): the solve then rates each case's bin at the torque and speed that case's train gives the pair, the duty's bounding at the pair's torque times the case's factor. A standalone pair, or one in a project with no shaft speed, train-mode stage or coupling, is bounded as before. (A project of given stages with a shaft speed is marked too, where the two ratings agree: on the safe side.)
- A steering block (strokes or misuse events) beside a rack member or a rack mesh mounted on one, without a steering system, is refused at `steering` (0.3.0): without a system the duty rates standalone rack meshes. Its derived load cases would load the rack member, whose yokes change regime and whose spring braces hold or give way with the force along the strokes, and the cases do not bound those states between them; and it would rate a mounted mesh's teeth without the rack's lift at the contact. The `steering-rack` template with a stroke now gives that error and derives no load cases. Give the project a steering system (the duty is then rated over its balance), or rate the duty in a copy whose rack meshes stand alone (the `steering-duty` template).

- A steering duty over a steering system (0.3.0) rates the rack meshes' teeth, and with strokes the system's shafts, the bearings linked on them that take the load cases, its worm pairs and its ball nut over its states (see Steering system). Not rated over the duty (a later release): a mounted gear pair (rated at the system's `state`, its checks not rated over the duty, never a pass: its rating takes its solved shafts' KHβ, which the states do not give it yet), and likewise fits, keys and housings. A rotor's air gap is not rated over the states (its band between them is not drawn; never a pass: rate it at the duty's heaviest state). A bearing's warning over the states (a minimum load undecided at its valley) is never satisfied, and its gyroscopic slip (the high-speed model's) is the system's state's, a pass not rated over the duty. Beside such a duty a design search's metrics are refused but the shaft's mass (their values over the states have no bound between them, and some items' results are the system's state's): search with requireChecks, whose checks take the bands. The ball nut's life is the states' Miner sum (as a bearing's), its static safety the worse of the states' and the system's own state's. Left at the system's `state`: the couplings, a bearing with `useLoadCases` false, and everything at a misuse event (the events rate the pinions only; the elements keep their own `peakFactor` for a static peak, which over the duty multiplies every state's loads, as over load cases). A worm pair and a drive take the worst of their states, each rated as if it lasted their life: on the safe side, as DIN 3996 holds no load spectrum here. A shaft's states are each rated with their stresses fully reversed (as a mounted ball screw's states are), on the safe side where a stroke does not turn round. The band of a bearing's checks is drawn from its life, static safety, contact stress and thermal speed in the states, relative to each check's utilisation. Two supports in seats of one housing (whose seats push each other with the bearings' loads) leave the elements unrated over the duty (a warning at `steering.duty`, their checks never a pass): the states do not couple the seats again (a later release). The project's own load cases beside such a duty rate the shafts and bearings instead (a warning at `steering.duty`), and their checks are then not rated over the duty (never a pass): remove the load cases to rate them over it. The elements' states are some four times the duty points (284 for the Dual-pinion EPS's duty), each a state solve and a solve of every shaft: at most 1200 (`system_cases::CASE_LIMIT`) and at most 12,000 solves (`WORK_LIMIT`: each state one, and one for each shaft and each bearing on a shaft; some 25 s natively and 40 s in the browser), past which the elements are not rated over the duty (a warning at `steering.duty`; their checks never pass) while the teeth are. Measured under load on the build machine (one process, natively): the Dual-pinion EPS's duty 9.3 s for the teeth and 19 s in all; the Belt-driven ball-nut and Column EPS examples given its duty 2.4 s and 2.8 s for the teeth, 7.0 s and 5.2 s in all (168 and 88 states); the browser's engine took about the same under that load (17.9 s, 7.6 s and 6.3 s). It refuses a stroke's `torque` table (the system splits the rack force, not a pinion's torque); a rack mesh off the system's rack is not rated over the duty (a warning; it keeps its rating at its own load). Each duty point is a state solve (some 15 to 30 ms natively), at most 4000 per duty. Rating the teeth of two pinions is the slow part: the Dual-pinion EPS template's duty (a parking, a manoeuvring and an on-centre stroke, both ways) takes some 4.7 s natively and 7.3 s in the browser's engine on the build machine (some 20 s natively on a laptop, 4 s of it the state solves); the app computes it on request, with progress and Cancel (0.3.0), and solves each edit without it. Positions past the rack's travel are refused (each point is a state of the system, within its stops). The lift is rated within 1e-4 mm of the states' between points (the contact ratios within some 4e-5), not raised; the forces are raised to bound theirs. Misuse events are rated at standstill (vehicle speed 0, forward efficiencies, no friction).

- The steering duty rates only the bearings linked to the system's shafts' supports. A support that names a bearing by its designation only links none: nothing rates it, at the system's `state` or over the duty; the duty lists it (`elements.unlinked`) and the summary names it as not assessed. The steering examples draw their bearings so: the Dual-pinion EPS's 6001 and HK2816 on the steering pinion and 6001 and 6203 on the assist pinion's shaft, the Column EPS's and the Steering rack's likewise, every other support of theirs (the column, the intermediate and input shafts, the motors) plain. Only the Belt-driven ball-nut EPS links a bearing, its nut bearing BR1 (see "From 0.3.0"). So the Dual-pinion EPS's duty rates no bearing. The designations are drawn, not sized: linked as drawn (6001 preset, default grease), the Dual-pinion EPS's assist-pinion 6001 at S51 would not carry its parking strokes (static safety some 0.6). Sizing and linking the examples' bearings is for a later release; until then link bearings at those supports to rate them over the duty (a needle bearing such as the HK2816 as a bearing of your own).
- Over a steering system's duty states (0.3.0), all on the side of saying more or of reporting less, for a later release: a rack buckling span that exists in some of a span's states and not in others (the bar moves under the supports) draws no marginal band over that change; a shaft standing still at the system's `state` takes FKM's checks over a number of cycles at the worst state for all its cycles instead of their Miner sum; a state that gives the ball nut no thrust can leave its rating over the duty refused; the system's own state, kept among the worst-case checks, is rated with its mean and amplitude stresses where the duty's states take theirs fully reversed (a state heavier than every duty point is rated more leniently than they are); a refused duty marks every drive's checks, also those of a drive off the steering's train; a bearing's band for its checks other than life and static safety is drawn from those quantities' bands rather than its own; `atState` does not list bearings off the shafts or planet bearings (rated at their own loads), nor the couplings.
- Over steering-derived load cases a linked bearing's minimum load check does not evaluate the axial load's holding (0.3.0): a radial load below the minimum fails there even where the axial load keeps the elements loaded; a preload still holds, and the bearing's own loads keep the exception. Precise handling comes in a later release.
- Over steering-derived load cases the checks that fail at a quantity's low point (valley checks: a linked bearing's minimum load; its gyroscopic slip, which only the high-speed model gives, is undecided with it) are undecided where they pass (0.3.0): the cases bound loads and stresses from above, not their lowest values between the cases. A preload that holds in every case keeps the check, and a module at its own loads is rated as entered. The checks at a peak (stresses, lives, safeties, temperatures, speeds) are bounded. Precise valley bounds come in a later release.
- A yoke's preload is not checked against the separating force of the on-centre states (the rack may rattle on centre with a preload below it): the yoke panel checks only its mesh's contact ratios at the lift and warns when the yoke sits on its stop. Compare the preload with the mesh's separating force by hand; the check comes in a later release.
- The steering duty finds a quantity's largest value over each part of a stroke from samples: 8 even ones, 12 toward each end and seven rounds of a finer grid about the largest. A smooth peak narrower than the even grid's step (1/9 of the part) and away from the part's ends can pass between them; the jumps of a module's regimes cannot (each is located and sampled on both sides). Strokes whose force tables have such narrow peaks are best given a point at the peak.
- **Add a steering** puts the rack's meshes in the steering's train and takes the torques off the driver, the shafts coupled to it and the pinions' shafts. A gear pair, worm pair or drive between the driver and the rack still rated at its own torque is left as it is. The steering then reads "not solved", and its error offers the fix (`torqueMode` `train`). Apply that fix, or set it in the item's module.

- The app opened at a copied or reloaded Steering address (`#/steering/<type>/<section>`) opens that page once the project's first results are in, even when another module was chosen meanwhile; leaving Steering before then should let the address go, for a later release (land-t review). An example that lands on Steering no longer does this.
- Compare variants writes a steering-duty check as "not computed" (or with its value) in every column by the first column's state: a design whose steering duty is computed beside one whose duty is not shows both alike, for a later release (each column's own message; review of RR1 branch 11). The status word in each column is that column's own.

Known limits of this version (all modules)

The list of known limits of this version, in full. Those that belong to a chapter are repeated there.

- The lint's number-reader rule (`axomek/number-reader`, R-388; a development check, nothing users see) finds `parseFloat` anywhere and `Number`, `parseInt` or `+` of an input's or text area's `value` read where it is used (any type assignable to `HTMLInputElement` or `HTMLTextAreaElement`, generics included). It does not see the value passed on first (a variable, a destructured `{ value }`, a function's argument), a key computed at run time, `Number.call`/`apply`, an element typed `any`, or Angular template expressions. Reviews catch those; read typed numbers with `core/format.ts`.

- Own templates and parts (R-926), each on the safe side (nothing is lost): - A change made in Axomek 0.3.0 to 0.3.2 and still waiting at the update may be shown as refused although nothing else changed the item: that version does not say which version its change was made over. So is a part made in an old tab and changed in the new app before the first sync, and a change in the new app of a part an old tab sent itself. - A change made before this device had brought the list up to date (the app updated while offline, or a part made signed out and changed before the first sync) is shown as refused: Axomek cannot tell which version it was made over. The card says so. - A refused change stays on this device, said on a card, until it is saved as a copy under a new name ("Gear (mine)"); there is no other answer yet (a later design), so a card can only be cleared that way. A refused delete has nothing to save: its card stays, saying so; delete the item again yourself if you still want it gone. A card is never changed by a later refused change of its item (an old tab's edit or delete made after it synced, say): that change gets a card of its own, so one item can show several cards, each keeping its own change. Only an old tab's newer version of the very change a card shows takes its place there. Saved as a copy, yours never replaces the account's item: replace or delete the other version yourself if yours should take its place. - A part (or template) renamed offline in Axomek 0.3.0 to a name another device took meanwhile: the old name is gone and the renamed part waits on its card; "Save mine as a copy" keeps the renamed part. That version sends a rename as a delete and a new item, which the account decides one by one. - If the updated app cannot trim an older Axomek's waiting changes (the file held by another program), a change that version then replaces with a newer one keeps its card beside the newer one's: two cards to save, the older of them no longer wanted. Nothing is lost. - A copy that would take the list to 4 MB, counted as the account counts it (with waiting deletions not yet counted as freed), is not saved and the card stays: delete other items, then save it again. A deletion is never refused for its size. - For a copy within a few kilobytes of 4 MB, this count can come out a little above the account's, and the copy is refused although the account would take it: when the copy takes the name of an item deleted earlier ("Gear (mine)"), whose deletion the account then forgets, or when this computer's clock is behind and deletions have just aged out on the account. The design stays on its card on this device; nothing is lost. - In the web app, replacing a part or template of several MB can be refused by the browser's storage limit: until it syncs, this device keeps the account's version beside yours. Nothing is changed or lost; delete the old version, let that sync, then save yours. - While a replacement of a large item waits to sync, both versions are counted, so saving another large card as a copy can be refused until that replacement has synced. - A copy saved from a card can still be refused as too large when it syncs, if another device added to the list meanwhile: nothing is lost, the copy stays on this device waiting, with a sync error. Free space on any device (delete items), then sync again. - A list edited right up to the 4 MB limit while offline can be refused as too large when it syncs (the account also counts what it keeps about each item): the changes stay on this device with a sync error, and with many changes waiting a deletion made here may not get through either. Delete items on another device (or in the web app), then sync or reload here. - A sync sends at most 16 MB of waiting changes at a time (a waiting rename carries its whole item), the rest with the next. When more than that waits (many offline renames of large items), a batch can end on a list the account refuses as over 4 MB although the changes after it (a deletion) would bring it under: the changes stay on this device with a sync error. Delete items on another device (or in the web app), then sync here (review C1 of ownlist-fu-033). - A change too large to send at all (only a copy made outside this version's 4 MB limit can hold one) is kept on a card on this device and never sent; saving it as a copy is refused as too large. A build from before 0.3.3 (a development or staging build of the new copy) left open beside it says that card as changed elsewhere; nothing is lost (review D3). - A

damaged record of which account the templates on this device belong to (not JSON, or not naming an account by name), or one that cannot be read, keeps the templates made signed out from being shown, saved or taken by an account signing in, saying so, rather than show them as anyone's (they may be an account's of 0.2.5); this holds while there are none too, so nothing can be saved signed out until the record is put right. Signed in, the account's own are listed and kept as before (reviews D1, E1, E2, F1, F5 of ownlist-fu-033). - Privacy, until the sign-in and owner-record design is reworked in 0.3.4 (owner-accepted, ownlist-fu-033): - A templates owner record with no account in it at all (only after damage or editing) reads as naming none: an account's templates of 0.2.5 still under it can be shown signed out, or taken by the next account to sign in (review G1). - In the browser, an owner record stored as the text `null` (damage only) reads as no record, with the same effect (review H1). - Those made signed out go to the first account that signs in, at that sign-in only (as before 0.3.3). In the browser, when the browser's storage cannot be read at that moment, they are not taken and not tried again: the next account to sign in in that browser can take them (review G2). In the desktop app, when their file is held by another program at that moment, they are not taken and stay those made signed out, also for later accounts (review M1). - In the desktop app, when the first account's own copy here cannot be read at that sign-in (held by another program for a moment), taking those made signed out writes them over it: that account's unsent changes on this computer can be lost (R-971). - In the desktop app, when moving an account's templates of 0.2.5 fails (its own copy cannot be read for a moment) while the record says no account has signed in yet, the next account to sign in can take those templates (review H2). - In the desktop app, an owner record that cannot be read at the first sign-in (held by another program for a moment) counts as one saying an account signed in before: those made signed out are not taken then and not tried again, so a later account can get them (review H4). - With that record damaged, `axomek templates --own` signed in does not sync first: it lists this computer's copy with a warning (`templatesNotSynced`), as when offline; the apps sync as before (review G3, owner-accepted). - A save refused for its size just after the browser's storage could be read again leaves the message that it cannot be read until the list is read again (reload the page) (review H5). - In the desktop app, when another program holds the list's file for a moment just as a sync keeps what it brought, that sync fails with the system's own words (nothing is changed or lost), and the message can stay until the next sync, even after the list was read again (review I3, accepted by the lead). - In the desktop app, deleting an account while the templates' owner record is damaged is said to have failed although it went through: the account is deleted and signed out here, and its copies are removed except as the known limits next to this one say. A templates file of Axomek 0.2.5 whose damaged record cannot say whose it is stays on disk, and another account's file held open can stop the cleanup (the next line). The app shows the account signed out once it is opened again (review I1, accepted by the lead). - On a shared desktop, deleting one account can be reported as failed, and the account still shown signed in, when another account's old templates file is held open by another program; the deletion did go through; reload to see it (batch-52 merge review, accepted by the lead). - In a signed-out tab whose storage was unreadable while the list was deleted elsewhere, the deleted items stay shown after the warning clears; reload the page (review K1, accepted by the lead after the council). - Signed out, `axomek templates --own` can list nothing when the list is withheld (its owner record cannot be read or is damaged); an empty list there may mean withheld (review K2, accepted by the lead after the council). - A tab of 0.3.0 to 0.3.2 left open lists only its own waiting changes once the new app has run: deleting there a part it does not list does nothing, and the part shows again at that tab's next sync. While such a tab syncs, the browser holds its full copy beside the new one. - 0.3.0 to 0.3.2 ask to be updated when they re-add a deleted name,

edit a template 0.2.x wrote that was never changed since, or edit after a lost answer in some orders; the change stays on that device. Two windows of 0.3.0 on one device can still overwrite each other (that version's own behaviour; the fix is the update). - A template changed in 0.2.x stays on that device until it is updated, and 0.2.x's own merge may still prefer a newer copy from the account. A template deleted in 0.2.x while offline and saved again since (on another device, or in the updated app) stays: the newer save wins, with no notice. Until such a delete is sent, the app and `axomek` list templates as if it had gone through; the account may still decide otherwise (a 0.2.x copy so old that the account no longer remembers what it had), and the list shows the account's decision after the next sync. - An item made signed out goes to the account at the first sign-in only under a name the account does not remember. Under a name the account still remembers (listed, or deleted not long ago: deleted ones are forgotten after a while), it is kept in this device's signed-out list, unsent, without a notice, however newer it is (as before 0.3.3). - A list near the 4 MB limit can stop 0.2.x from saving changes to its templates on that device: the account's file it downloads carries the newer versions' extra fields, which count against 0.2.x's own size check. Update the app. - `axomek templates --own` and `axomek new --template` read a snapshot: run at the moment an app syncs or first moves this device's list to the new copy, or while an older Axomek still holds an unsent change, they can list the account's version without the newest change made on this device, or call a just-saved template unknown. `axomek` only reads, so nothing is changed or lost; run it again once the app has synced. - A browser too full to hold the list twice for a moment keeps it as it was, unchanged, until space is freed. - After a desktop downgrade to 0.3.x, the new copy waits unread until the next update. - A part re-added after more than 90 days offline (or 500 deletes later) in the same run of changes that deleted it is shown as refused. So is the last Undo of Delete, Undo, Delete, Undo of one part made offline before a sync: nothing is lost (save it as a copy), but it comes back under a new name. - A part created in 0.3.0 to 0.3.2 whose save reached the account before the 0.3.3 service but whose answer was lost, then changed offline in that version, can come back after the update with that later change when another device deleted it meanwhile and the delete is no longer remembered (after 90 days or 500 deletes). Nothing of anyone's is lost: delete it again if it is not wanted (R-926, council-accepted).

- Open in the browser with storage refusing drafts (R-385, webopen-034, for filesafety-036): Open picks the file first and asks "Save changes?" after, reading the file again after each answer. An edit that lands in the last moment of opening (while the browser's list of kept projects is read) is asked about once more; answering Save there when the file picked is the very file being saved opens the file as it was read just before that Save. Nothing is lost: the file on disk has the saved edits, and where the browser writes back to that file the next Save says it changed. Open the file again to see it as saved.
- Privacy choices and tabs left open from an earlier version (owner decision 2026-11-24): changes made in a browser tab left open from 0.3.0 don't reach the new version; reload old tabs after updating.
- Privacy choices across tabs (owner-accepted): a choice saved in one tab reaches the other tabs after a short delay (a few milliseconds, not measured). A statistics batch or replay chunk whose last check ran just before another tab's withdrawal landed can still be sent, and a request already started cannot be recalled. Where the browser refuses to save a withdrawal (storage full or blocked), it holds in that tab's memory only: an earlier opt-in from another tab arriving late can undo it there. Reload the tab, or free storage, and withdraw again. A tab still showing an earlier account that hears another account's choice before its own sign-in state updates can clear the shared queue, so that account's events not yet sent are lost (never sent).

- Welcome email counts (welcome-catchup-032): an account whose deletion stalled after its mark (Auth user and notice record still there) can be counted in a run's `overCap` or `failed`, which can then be one too high. Its claim still stops any email to it.
- Privacy prompt with storage refusing writes (R-948, small-033): an answer the browser would not save closes the prompt for that tab's page, with nothing on. "Ask me again" in another tab reaches it only when that tab removes a saved choice (the browser tells other tabs of nothing else), so with no choice saved anywhere that tab asks again only at its next start or reload. Nothing is collected meanwhile.
- "No thanks" and "Ask me again" (owner-accepted): they save the two choices as two writes, "off" first, not as one; a page closed between the two keeps the second choice as it was.
- Desktop app (owner-accepted 2026-11-23, R-922): a feature-statistics batch already handed to the app's shell, while the shell fetches its sign-in token (usually under a second, at most about 35 s), can still be sent after Feature statistics is switched off.
- Update counts (Analytics › Updates) are approximate, and the console says so: they count the desktop app's own update events only. An update that asks for consent again sends none (and the install id is deleted while analytics is off, by design), so it is not counted; nor is one whose event aged out of the app's queue before it was sent. No update is inferred from a device's versions (R-919). A batch the app sent again (its answer lost) counts once, on the first copy's day, when the page read both copies; when the days before the range were read only in part (the page says so), a copy sent again just after UTC midnight can count once more, or on the later day (review of updates-032).
- Units (R-387, R-931): the app and axomek read the same units typed, with the same factors, and show the same US customary units; a test holds them to it (`units-parity.spec.ts`), also for a field shown in another unit than its value's. It lists what still differs, on purpose or refused (never read wrongly): a design search's bound on a deflection is typed in µm (mil) in the app, as its check shows it, and in mm (in) by axomek `optimize --require`, which refuses nm there; a bolt's expansion coefficient α_S is typed in 10^{-6}/K in the app, in 1/K in axomek. A bare number in a bearing clearance is in µm in the app (the unit it shows) and in mm in axomek without `--units imperial` (base units, as every bare argument; mil with it, as in the app, but mm again with `--json`, which always reads base units). The bolt's expansion field refuses its own unit typed out (`12 10⁻⁶/K` reads as two numbers): type `12e-6 1/K`, or a bare 12 with metric units shown (with US units shown a bare number is in 10^{-6}/°F). The text output and the report keep a few units metric that the app shows in US units (m, J, GPa, √MPa, mm³, a steering's feel: `SCREEN_ONLY` in `crates/axm-model/src/units.rs`). Not compared: fields the app shows in % of a plain fraction (a torque amplitude, a modulus share), and the change lists of the Assistant and `axomek diff`, which give a clearance in mm.

- Design variants (for a later release; release decision): files with variants open there. 0.3.0 opens no file that holds any, in the app (a file, a cloud project, a browser draft, a report's attachment), `axomek`, MCP and the Assistant alike, and leaves the file unchanged: "This project has design variants, which come in a later version of Axomek. Open it there; this version leaves the file unchanged." (code `newerFile`). It is judged on the file as written, before any upgrade: a variant in `variants.list`, an `active` variant, `baseValues`, or a malformed `variants` member is refused; none or an empty one opens. A project that holds variants all the same is edited by no command and saved by nothing (the same message). An open file replaced on disk by one with variants is let go of: the design shown stays, saved nowhere, and nothing is saved to that file. The Assistant declines to compare designs. Making, showing, renaming, removing or resetting a variant, and `--variant` other than `base`, are refused too: "Design variants come in a later version of Axomek." To try a change, save a copy of the project. The promise is narrow (owner, 2026-11-21): 0.3.0 never creates design variants. Files with variants exist only from pre-release builds; 0.3.0 refuses to open them and leaves them unchanged, but does not otherwise guard them: chosen as the target of Save as, or replaced by undoing a change in a folder where one took your project's place, such a file is replaced like any other. Keep such files aside until a later release.
- A motor bearing's minimum load and gyroscopic slip over its rotor's static eccentricity (user test rr1-motor; see "From 0.3.0"). While any rotor in the project pulls, their pass (or a warning for a bearing its axial load holds) counts only in the case rated in full, where all of these hold: - an uncoupled shaft alone: Couple linked bearings off, in no group (a rigid coupling, a bearing between shafts) and in no housing whose seats move each other; - no load cases; - the minimum load checked on the radial load F_r alone (not the equivalent load P of angular contact, tapered roller and thrust bearings), one bearing on its support; - no axial load on the shaft (an axial load shares out anew and can hold or let go of a bearing between the states); - the least loaded state (the least radial load, linear in the pulls without coupling) solved exactly, and the bearing loaded at the eccentricity. Every other such pass or held warning is not decided, saying why: "enter the eccentricity 0 to see it" (the Electric motor example, coupled, with an eccentricity entered, is one; it ships centred). Every bearing's gyroscopic slip margin is not decided in every case while any rotor pulls: the least radial load does not bound its balls' gyroscopic margin (a later release bounds the margin itself). A later release: a bound on the coupled shaft's least bearing load between centred and the eccentricity (the bearings' load–deflection laws bound how far load moves between them), solved with the groups' and housings' own equilibrium in each state and per load case; P, the axial holding and the balls' loads minimised themselves.
- While any rotor in the project pulls (a static eccentricity), the life, static safety, thermally permissible speed (DIN 732) and case depth of every bearing in the project are not decided (a pass or a held warning): a pull against another load leaves a bearing less loaded at the eccentricity than centred, and no bound from the supports' loads holds for a bearing's rating. Enter the eccentricity 0 to compare. A later release rates them at both states.
- With a rotor's pull, a one-way bearing's axial direction check can still pass in a narrow window between the states, as its 5 % threshold scales with the radial load; and a shaft's own checks (static and fatigue strength, deflection) are rated at the eccentricity as given, where a pull against the weight bends the span less than centred. A later release rates them at both states.

- A bearing's thermally permissible speed nθ is hidden beside any thermal speed check that is not determined, one whose speed is not its shaft's too, while the bearing's speed card still draws it (on the safe side: a figure left out); a later release hides it only where it is not decided.
- The rotor-pull refusals above apply to every bearing in the project once any rotor in it pulls, a gearbox's output shaft's too, though no pull reaches it; a later release takes only the shafts a pull reaches.
- Where a train's speeds come from its items and two of them turn one shaft opposite ways (a warning at the later one's sense, with its fix): - the pair the train took as driven the other way round is rated so meanwhile, and its mesh loss with it (the Two-stage helical gearbox with G2 alone reversed in its file: total 3 614 W against 3 610 W), though the power in and the efficiencies are not given; - a drive or pair added or mounted beside a train that turns its shaft negatively starts at its default sense, positive, and so gets the warning until it is reversed (the fix) rather than taking the train's sense; - a rack mesh's sense reversed is not carried on to the rest of its train (a gear pair's, worm pair's or drive's is), only warned of.
- On the steering's ball nut shaft, the friction of the nut's linked bearings shows in the power flow as power leaving that shaft (19 W in the Ball-nut EPS example): the nut's stage passes on what reaches its thread. The totals and efficiencies hold it once, as the report's power loss does.
- Renaming and deleting project files in the desktop app's Projects screen (RR1 TA-F19 reviews), for a later release: - A window that has read a file and not yet claimed it (milliseconds, while it opens) is not seen by a rename from another window: it then edits the old name while the renamed file stops (nothing is lost). - A file is known by its path in lower case on Windows, which is not the file system's own case folding: in a folder set case-sensitive (or a WSL share), `Pump.axomek` and `pump.axomek` count as one file (opened read-only, a rename or delete refused); a rename of the other one from the list of the window that edits the first moves that window to the renamed file, whose next save is then refused as a change on disk. - The window's own file renamed from the list under another spelling (letter case, a mapped drive) is followed, but the recent files keep the spelling it was opened by under the old name, and the project's own name is not renamed with it.
- Pictures on Your projects and Home: a project kept before 0.3.0 drew a picture for every project opened (a browser draft, a desktop project not in a file yet, a recent file) shows its module's icon until it is opened once. Drawing those pictures in the background while the list is shown waits for a later release: reading the desktop's files without holding up the app, and keeping drafts' pictures without taking their unsaved work's room or racing other tabs and windows.
- A bearing life that is not determined (see "From 0.3.0") is said so in place of its number, with these limits in 0.3.0: its basic life is hidden with it, also where only the modified life is undetermined (κ below 0.1, a hybrid's a_ISO), which is on the safe side; without a life limit only a contradicting designation or a slewing ring's proportions mark it, so κ below 0.1 then shows a plain life (the verdict lists that life as not checked; a hybrid's modified life reads "not determined" with or without a limit); and the variant comparison's CSV writes "not determined" where a number stood.

- A result not determined is hidden display by display, not by one shared rule: in 0.3.0 the displays reviewed (the report, Show the calculation, the CLI's text, the comparisons, the Overview, the cards, the modules' tiles and the Assistant's digest) say "not determined". Those found last are refused instead: the Changes dialog's and Compare projects' check rows and the Overview's key results of keys and bolts say "not determined (see the check)"; the assembly view gives no share of the limit for a check not determined or not computed; and in the report an element's results, and a key's or a fit's results its checks go by (safeties, capacities, permissible values, pressures, shear, the fit's Turning row; with `axomek solve`'s key and fit lines), say "not determined (see the checks)" as soon as one of its checks is not determined, determined ones with them; a key's page refuses its flank pressures and tooth shear stress the same way (its permissible values and capacity, which do not depend on its rating, stay). A bolted joint with a check not determined (beyond VDI 2230's limit G, a duty not computed) has its whole Results block refused the same way in the report's section and in `axomek solve`'s line (its tightening torque, permissible assembly preload, resiliences and embedding with it, which the bolt page still shows); its page refuses its preload needed, load factor, clamp loads, stresses, opening and the joint diagram, but the notes under them, on its page and under the report's section, still quote the eccentric model's figures (M_Sb/W_b, F_Sm/F_0.2min, F_K against F_Kab). Make it pass and `axomek fix` give such a check no value either. A module's page, once one of an item's checks is not determined, shows no tile's caption (it may quote another check's figure) and refuses its tiles and Details tied to no check (a worm pair's load value C / Clim), and the Overview refuses the item's tiles tied to no check the same way, including ones that do not depend on the rating: a worm pair's method, centre distance and ratio, a motor's power and magnetic stiffness read "not determined" too; a later release shows those again. A design search meets no requirement or objective on a result of a design with any check not determined (a requirement's own check may be cleared, and the system life and power loss take every item's), its dimensions apart, nor on a fit's yield safety resting on an estimated yield strength, even where that figure itself would stand: the safe side; a later release decides each figure on its own. A bearing whose life the engine puts outside its rating method (κ below 0.1 where it turns, balls in four-point contact, a hybrid, whose a_ISO is defined for steel only) has all its lives, basic and modified, meet no search requirement or objective, as a minimum, a maximum or an objective, its minimum life cleared or not, and so has the system life with such a member; it says "not determined" without a life check too. The safe side: a hybrid's basic life, or one at a low κ, would stand; in a later release a search requirement on a figure whose own check is cleared is evaluated as that check, each life on its own. A design with a bearing the engine marks outside what it rates (a designation that contradicts it, a slewing ring's proportions) takes no calculated result in a search, its dimensions apart but that bearing's own (inferred from its geometry as modelled), as a design with a check not determined does, whatever its own checks say. A bearing whose designation contradicts it shows only its results' warnings it can vouch for: those at its own path, of the bearing as modelled, are left out (a validation warning at that same path too, such as that its geometry is estimated), its fields' warnings stay. A check not determined is said in a short form wherever its message would be rewritten: what it checks, named from its id, "not determined" and its kind's reason in fixed words, else "(see the check)", never a part of its message, which can hold a name the user wrote and the figures after it; a name that holds an engine mark's words can still pick another kind's reason (no figures). The full engine message, with the reason in context, comes back in a later release, when every display shares one rule. The bearing module's speed chart is not drawn while the bearing's operating nθ is not determined (a rotor's pull, a designation its geometry contradicts), saying why; `axomek speed-limits` then lists the curves' samples without an

operating point. A later release draws the curves apart from that load. A display not yet reviewed may still give such a number beside its check, which says "not determined". A later release routes every display of a check's value through one rule, with a test over every template. Known in 0.3.0, closed in a later release by that one shared display rule: - the Assistant's solve data still carries a check's value and utilisation where the check is not determined (a bearing's thermal speed whose nθ is withheld: its speed and share of the limit, from which the withheld nθ can be worked out); the check is flagged not determined there, the solve tool tells the Assistant never to quote a figure so flagged, and the check's row and every display say "not determined", never a pass; - a rack pinion shaft not rated at its mesh's peak because its only drive is solved for one power direction (the Dual-pinion EPS example's SH5, driven by worm pair WG1) is said, in the verdict's and the report's short reason, to be refused because its torque sources cannot be shared; its check's own note gives the right cause; - a design search takes no result from a design with any check not determined, so a project whose steering duty leaves a warning over its states may find no design meeting a shaft or bearing requirement (the safe side; a later release decides each figure on its own); - `Life0utside::against_bounds` keeps a maximum branch no caller uses (the search reads the hybrid flag directly); a later release removes it; - the Overview's governing check of a module, between two checks not determined, keeps the first in the list (the bearing listed first) where the engine picks by id, so the module tile and the verdict can name different checks, both said not determined; - a variant comparison can mark as best the life, and its check's margin, of a bearing the engine marks unrated (its designation contradicts its geometry), though the bearing's own results say its figures are of the bearing as modelled: compare such a design only after its designation is cleared or corrected; - a comparison of three or more designs can leave out a result row whose neighbouring columns each agree within the tolerance while the first and last do not (10 000, 10 000.8 and 10 001.6 h); `--all` shows it; - a comparison of designs whose life checks all pass on a hybrid's basic life (its modified life outside its rating method, so no best) leaves that check row out of the default CLI comparison and the report, though the minimum lives and the checks' shares differ (a hybrid 6309 at a 1 million h minimum against 4 million h); `--all` shows it. - a hybrid whose modified life fails its minimum keeps that failing check's number in its check row (the fail stands, on the safe side; the number rests on a_ISO, defined for steel bearings only), and so does any life check failing outside its rating method (a steel bearing turning below κ = 0.1, or with its balls in four-point contact): the report's Summary of checks and its Summary page's Governing checks show its share and draw its utilisation bar like any other (a 30207 on 2 / 1 cSt oil against a 20 000 h minimum), though that share cannot be compared with another check's; the governing pick and the orders leave it unranked, and the typed results' report follow-up shows no bar for it. Show the calculation of a hybrid's life explains the modified life, said not determined, not the basic life its passing check goes by; - the Assistant's shortened record of an earlier solve lists a failing life check with its value (a hybrid QJ 207 in four-point contact: 94 476 h against a 1 000 000 h minimum) but without the check's outside-method flag or reason, though the same record says the bearing's life is not determined; the fail stands, and the full solve the Assistant reads carries the flag; - over a duty cycle whose cases stay within the rating method, the Assistant's solve data still gives the nominal summary's modified life where the loads as entered put κ below 0.1 (the Overhung pulley's BR2 on 10 / 2.5 cSt oil at 150 °C nominal, 35 °C in its one case: 107 191 h), with no note that it is outside the method; the duty's life, which the check and every display go by, is right (a nominal summary in four-point contact is marked); - Show the calculation of a check that fails with an unbounded share says it fails "whatever the value of the checked quantity"; for an axial preload that lifts off (a cross-located pair whose housing runs hotter than its shaft) the share is of the lifted-off

state only, and the same check passes once the temperatures no longer lift it; the check's message gives the cause; - "Make it pass" on a check not determined still steers its search by the check's share as modelled (a proposal is made only where the check is determined and passes after the change), and its "tried" list may give that share as the closest reached; - the load-spectrum card's chart still plots a bearing's damage shares where its table withholds them (a case's life outside the rating method with the life check failing); read the table; - the report's summary page labels a passing hybrid's life "modified reference rating life" before the parenthesis that says it is the basic life its check goes by; the bearing section is right; - the agent guide says a hybrid's check `value` is its basic life; where the check fails (`outsideMethod`) it is the modified life it fails on; - a variant comparison of a steel and a hybrid bearing that both pass their life check leaves that row out of the default CLI comparison and the report, though the two columns go by different lives (the row's `labels` say which); `--all` shows it; - the Overview's utilisation chart plots the share of a check failing outside its rating method beside the others, ordered first among the failures but drawn as if comparable; the fail itself is shown; - a hybrid roller bearing on estimated geometry passes on the lower of its two methods' basic lives (correct), but the report's summary page names that life with the other method's label, the Assistant's basis text points at `basicLifeHours` (the picked method's, possibly the higher) instead of the check's value, and a check made not determined by the other method's lower basic life does not quote it; the bearing finder's `basicLifeHours` and the bearing page's basic life rows show the picked method's too; the check's value and status are right; A later release types every result figure as determined or not determined, and every display, ranking and short message reads that type: this closes the ranking, display and short-message limits above for good.

- `axomek check --strict` and `axomek solve --strict` exit 3 for every file with something its method does not assess, which no input clears yet (4 where a check fails and the file was solved completely; a file that was not, an unknown field under `--strict` among it, exits 3 first): a rated key or spline (reversals and load peaks), a rated fit (its shaft seat), a belt or chain drive (its capacity), a fatigue-rated weld (its toes), a yoke on its stop, a support with a designation only and a planetary set without a planet bearing. Check those by hand and use `axomek check` without `--strict` for such files (it still exits 4 on a failing check and 3 on one not determined); a way to mark an item as checked by hand is for a later release.

- A test over every example and five more cases (a hybrid, a four-point contact bearing, a designation its geometry contradicts, the electric motor pulled off centre, a life below κ 0.1 failing beside a determined fail) replaces every figure the engine withholds and checks that no display shows it and no verdict or ranking moves with it (the "poison" harness). What it still finds is listed, each with its reason, in `crates/axm-cli/tests/poison/expected-fail.json`. Beside the limits above it found: the report's appendix of calculations gives a figure not determined as a number (a contradicted bearing's life, the pulled motor's critical speed); `axomek diff --json` and Compare projects give the check and result rows of such a life their numbers without an `outsideMethod` flag; the design search's `--json` gives a candidate's `objective` with no flag of its own where it is such a figure or one not determined (its `undetermined` names the metric; the search does not choose by it); `axomek find-bearing --json` lists a candidate's checks in an order its outside-method life's share moves (the candidates' order does not); and `axomek solve --json` gives `shaft.governingCriticalSpeed` with no flag where the check `minCriticalSpeed` is not determined (read that check's `undetermined`). A bearing's contact stress shows as a number where the static safety that goes by it is not determined: in the Overview's key results, `axomek solve`'s `p max` and the report's Max contact stress (read the static safety's "not determined" beside it). The app's checks panel picks µm or mm for the limit of a length check below 1 mm by the value as modelled, also where the check says "not determined" (no fixture has such a check yet). The harness itself does not tell a leak that moves to the same words on another line, a further difference inside a line whose wording already differs, or a leak moving to another item at the same place of a list from the one its table lists; nor, in the report's decisions, items whose ids are numbers alone ("1", "2").
- A gear's mass is warned of only far off its geometry's (above twice a solid disc, below a quarter of the disc less the shaft): a mass left from a gear up to about twice smaller, or from one only a little larger, is not, and the critical speed then reads somewhat too low or too high. Check each gear's mass after resizing an example.

- An electric motor's critical speed and magnetic stability with a static eccentricity on coupled bearings (review of ut4-motor-critical-speed; see "From 0.3.0"). In 0.3.0: - A pull not along a bearing's other loads (more than 1° apart: a belt or a gear's force across the weight) leaves both not rated: the bearing's load turns as the eccentricity grows, and its stiffness along y or z can be least in between, which neither the states taken nor a bound with each bearing equally stiff every way at its least load bounded (a reviewer's case read 9 988 rpm where a state inside the bound gives 9 604 rpm). This blocks a belt- or gear-driven motor with an eccentricity from the coupled critical speed, and 0.3.0 names no way around it: no stiffness entered on the supports (with coupling off) is known to be safe, as the least one lies somewhere between centred and pulled (keep the eccentricity: its pull counts in the bearing loads and the air gap; 0 would drop it from them). - A state the coupling cannot settle exactly (bearings loose in their clearance, e.g. the Electric motor example without its wave spring, centred) leaves a passing result not decided (a warning). A motor whose bearings carry no load with the rotor centred (the weight off, a vertical motor without preload) has both not rated, with a hint (a horizontal motor: turn the weight on; a vertical one, or one held only by a preload, 0.3.0 cannot rate). The hints of the not decided and not determined results name no stiffness to enter: none is known to be safe there (review: a recipe of the bearings' stiffness without the pull could pass a motor that fails). - A coupled bearing under an axial load (a spring preload, as in the Electric motor example, or the shaft's own axial loads) or preloaded from within (an X or O pair's preload, a negative operating clearance) leaves a passing result not decided (a warning): its radial stiffness need not grow with its radial load, so the states taken bound nothing between them (a reviewer's case: e = 0.2 passed 15 000 rpm at 15 908 rpm where e = 0.15 gives 13 878 rpm). - A spring preload is refused on any shaft joined to another by a bearing between them (shafts solved together do not carry it), also where the spring would act on its own housing. - The unbalance response is the given eccentricity's, with a warning: at a smaller one it can be larger. - Up to four pulling rotors on a shaft; more have both not rated. The axial modes of a state keep the locating bearings' axial stiffness and the housings' seat motions of the pulled state. A critical speed or magnetic stability that is not rated fails, shown as "not rated" in place of its placeholder value (0 rpm, a share of 100 %), and What matters most leaves it out, saying why. Planned for a later release: the rotor pulled within its bearings' clearance to where they hold it (it never sits centred there), in place of the centred state; and a sweep of the eccentricity with a proven bound between its steps, for a pull across the bearings' loads, for bearings under an axial load (a spring preload) and for the unbalance response.
- A speed set again as it is, for a later release (user test 4 reviews): on a pair, worm, drive or rotor it makes its shaft the train's input; on a shaft that follows another (`updateShaft`) it is given beside the input instead, and the next ratio change makes the two disagree, an error that names both. The app never sends an unchanged value, so there a speed becomes the input only when it changes. The note under a given speed calls it "the train's input" when any shaft in its placement group follows a speed, also one of another train. In the base, with a variant whose ratio, sense or case speeds differ, a speed sent unchanged on an item is refused like a new one. The stale-speed warning's hint prints the speed to six digits, so typing it by hand (1410.53 for 1410.526…) moves the input where the Fix button, which sends it exactly, does not. When the input moves, a measured spectrum's case names keep the old input's speed ranges, and a series too long to keep keeps its peak speed in the description. The base's variant check compares each variant with the input scaled as before, so it also refuses a move on another train than the cases' reference, or one whose variant would only read the cases at the new input, both on the safe side. A rotor whose speed was not entered (0) and is typed as its shaft's speed takes it without moving the input.

- Speed inputs and variants, for a later release (user test 4 reviews): a speed typed in the base that moves the train's input applies to every variant that does not override it, so a variant with another ratio and no case speeds turns its other shafts by its own ratio from the speed typed (the fan's 900 rpm typed: its motor at 1440 rpm, not the base's 1800), and the message names only the base's speeds, not the variant's. A variant that unmounts the drive or puts it on another shaft is not checked, so after such a move its old input shaft can be left with no speed, its items and bearings at their own. With a variant active, the note under a following shaft's speed still says a speed typed there makes it the input, which the app then refuses with the reason. In the base, adding an item with a speed (a rotor at 900 rpm) is refused whenever a variant exists, as the addition is replayed with each variant active, also one with nothing that conflicts, and the refusal says that variant is active: add it without a speed, then type the speed.
- Steering duty, for a later release (user test ut4): - A worm pair or a gear pair over a steering duty keeps its own required life (a worm pair rated at its worst state as if that lasted its life, a gear pair over the cases at its own life's cycles); the duty's length checks the ball nut's, the ball screws' and the bearings' lives. - Editing a mesh's Peak rack force for static rating in the duty's refusal card makes the duty not computed again, so the card (and the field in it) goes until Compute duty runs; with two refused meshes, the second is edited after recomputing, or on its mesh's own panel. - The drivetrain's natural frequencies (system modes) are dropped as out of date when another shaft is shown, though they do not depend on it; Compute them again. - While the duty computes, a check it decides reads "computing…" in every list of checks the app shows, also another report's (a version compared) of the same kind; the project's verdict line keeps the engine's count (a failure at the state counts) and says the duty is being computed. - A design search's work budget counts a remembered result of the steering duty at every request within one solve, so a design that runs on and on is stopped later than it could be (its time limit and Cancel still stop it); the engine's other remembered results (a gear tooth root's or a housing's finite elements) are not counted, which the budget's floor of 5000 steps covers.
- The assistant in apps older than 0.3.0 (0.2.7): the account service serves its requests on the model and effort Axomek sets for the account service, with up to 16000 tokens, thinking included, whatever model such an app's settings choose and its panel names; and such an app waits only five minutes for an answer through the account, so a very long step can time out there: the account service still waits for the answer and counts it against the day's usage limit, though the app never shows it. Updating the app fixes both.
- The assistant while a preview is shown (a drag, or a dialog such as Mount left open; reviews of rr1-assistant-edits). It never takes a preview's results for the project's, so it waits for the preview to go: up to a minute for the message, for each solve, and before and after each edit, so a turn can run long. After a minute it says the project is still being changed (also when nothing was committed), and an edit is not made. These err on the safe side.
- The assistant and the steering duty computed on request, for a later release (reviews of rr1-assistant-edits): whether the results include it is read against what is shown, so a duty that came in while the assistant answered is not noted under its reply when a preview is shown at the end (the next message is told). A later release should read it against the committed project.

- Administrators' emails in records from before 0.3.0, for a later release (reviews of rr1-privacy-texts). Since 0.3.0 the records of what an administrator did hold only their account id (`admin-names.ts`). Deleting an administrator's account takes their email out of the older records found by the emails it is known under (the current one and those of its audit entries; `admin-forget.ts`), so two kinds can keep it: a record written under an earlier email no audit entry names, and the records of an administrator deleted before 0.3.0 is deployed (their `revokedBy`, `by`, `updatedBy` and `doneBy`, and other administrators' revoke entries; the daily clean-up scrubs only the audit log). With one administrator who never changed their email, neither exists; a later release should rewrite the old fields to account ids once at deploy. Deleting an administrator with many accounts acted on runs a transaction per record at once, which may need a retry (safe) before it finishes; every account's deletion runs that clean-up by email, so someone who later signed up under an administrator's earlier email would, deleting their account, relabel the administrator's old records as "a deleted administrator". The daily clean-up reads every audit entry of a deleted account each day (correct and idempotent; a version mark on scrubbed entries would let it skip them).
- Earlier cloud saves copied by a rename or a restore before 0.3.0 is deployed have no `customTime`, so the Storage lifecycle deletes them 91 days after the copy rather than after they were replaced (at most 90 days late, until 91 days after the deploy); the new lifecycle rule takes effect only once `tools/release/apply-storage-lifecycle.sh` is run for staging and production at release prep.
- Gear flank modifications, for a later release (user test ut4): K_Hβ leaves lead crowning, helix slope modifications and end relief out (ISO 6336-1 method C without Annex C, nor the reduced width of 7.5.3.5.3 for a crowning or end relief above Annex B's amount); an over-crowned pair loses the f_ZCa credit and a counted warning compares its tooth contact's K_Hβ, but the rating does not take that K_Hβ. A helix slope modification or end relief earns f_ZCa 1.07 whatever its amount (Annex B.3's amount of end relief is not checked); the tooth contact warning covers them. `details.helixModification` "crowning" (ISO 6336-1 Table 12, B1 = B2 = 0.5) halves the misalignment in K_Hβ whatever lead crowning the gears are given, none included; the tooth contact warning covers a crowning given, not one claimed by that setting alone. A pair with lead modifications costs one contact analysis per solve (two with a misalignment given beside the shafts'), Make it pass's trials included; no example has them, so the solve work table does not hold that cost.
- Make it pass and the gear warnings, for a later release (user test ut4 review): the per-gear face width warning (`gears.<id>.details.faceWidths`) and the tooth contact warning (`gears.<id>.toothContactFaceLoad`) exist only while they warn, so Make it pass and `axomek fix` cannot find a change that clears them (a new face width b clears the per-gear widths, but no trial reads that as passing). Change the field the warning names by hand. The same holds for a bearing's axial direction check (`bearings.<id>.axialDirection`): it exists only while it fails, so Make it pass and `axomek fix` cannot find a change that clears it, and no trial turns a bearing round (Opus review of land-t). Turn the bearing round, or choose one that takes axial load both ways, by hand.
- Bearing loads, for a later release (user test ut4): bearings have no application or shock factor (f_d) of their own; their loads are taken as entered, in the life and the static safety, and the report does not print one. For a machine with shocks, multiply the loads yourself or add a load case at the peak (the bearing help says so).

- Rotating loads, for a later release (user test ut4, Codex review of ut4-small): a bearing's loads are taken as fixed in direction. A load that turns with the shaft (an unbalance, an exciter) has no setting of its own: **Outer ring rotates** gives its load zone and life, but also the cage and element speeds, the friction and the high-speed model's centrifugal and gyroscopic forces of a turning outer ring, which the bearing does not have. The app and the help say to check the life both ways and keep the shorter. A load of unknown direction, or both senses of rotation, is not a study of its own: give each direction worth checking as a load case (a negative factor reverses a load).
- Slewing rings are out of scope (user test ut4): the four-point contact type rates the standard series (QJ) as a rolling bearing with rigid rings; a slewing ring's rings are bolted and flexible, and it is rated by its maker's curves. The app and the help say so, and a four-point contact bearing of a slewing ring's proportions (D_pw/D_w above 15) has its checks not determined (from 0.3.0). A thin-section four-point bearing of such proportions is not determined either (the safe side).
- Belt and chain capacity (user test ut4): a belt drive checks only its pretension (against slip, or a slack strand), never what it can transmit: its power rating, tooth shear or tensile strength; a chain is not rated for wear and fatigue. The unchecked list names each drive's capacity as not rated. Take it from the maker's data.
- A belt's shaft load is the running one, its strands less their centrifugal tension F_c; at standstill a belt (friction or timing) pulls with its whole pretension, about 2·F_c more (a fast V-belt in user test RR1: 1.11 kN running, 1.40 kN at standstill). The shafts' and bearings' static checks take the running load times its peak factor, so the standstill pull is not checked; the drives help says so. Static checks at the larger of the two are for a later release (RR1 U-A21).
- Steering misuse events (user test ut4): the misuse events rate the pinions' tooth roots (their contact stress is reported, not checked) and the pinions' shafts statically; the ball nut and its bearing are rated in the steering state and over the duty's strokes, not at a misuse event's peak, and the elements keep their own peak factor for a static peak. A worm pair that drives an assist pinion (a plastic wheel too), the other shafts and the bearings are not rated at a misuse event either, and only the report's method notes say so: the results do not list them as not checked for it. A mesh's peak rack force for static rating is a case of its own, the whole force through that pinion with the assist failed, rated for root and contact, so the same kerb strike can pass as a misuse event and fail there; the checks do not name it as the assist-failed case. Both err on the safe side or are disclosed; listing them on screen is for a later release (RR1 U-A25).
- A shaft's DIN 743 fatigue check over a steering duty is an endurance check at the duty's worst state, however few times it occurs: a shaft loaded hardest only when parking at the lock (60 000 strokes in user test RR1, SH5 at 1.162 against 1.2) fails it where a finite life could pass. On the safe side; a finite-life shaft rating over the strokes is for a later release (RR1 U-A26).

- Keyboard and screen readers, for a later release (user tests ut3 and RR1): the app's screen reader checks read Windows' UI Automation from a real Edge window (`npm run smoke:screen-reader`, on demand, not in CI) and the browser's accessibility tree; no run with NVDA or JAWS themselves has been made, and a check by a person with one is still advised. A check's rule written as a formula in its name (a bearing's minimum load, "k_r 0.025·(ν·n/1000)^(2/3)·(d_m/100)²") is read out symbol by symbol. A press on a fix button while a list shows an option not yet applied applies the list first, and the click is lost: press again once the suggestions are solved. A chart's "Show the values as a table" button shows only with the keyboard focus on it or the pointer over the chart (always shown only where the main pointer is touch: on a touchscreen laptop whose main pointer is a trackpad it starts hidden), so voice control that numbers what is shown may skip it on a computer with a mouse. The edges between the columns have no single-pointer way to resize them other than dragging (the keyboard has its own; WCAG 2.2 2.5.7 asks for one), and their 24 px targets reach from the border into the drawing's and the results' left side, so a control within 24 px of that border is pressed by its edge instead. On a window too narrow for both chosen widths, a column's edge does not go wider than it is shown (narrow the other column first). With a part shown in the Shaft view's results column, the inputs' edge reckons with the results width chosen rather than the part's, so it may stop short of the room there is, and a drag on a window too narrow for both may change the results width kept. A refusal already there as the Shaft page opens (a shaft without supports solved on the Assembly page first) is in the page's status region from the start, so a screen reader need not say it on its own: it is read with the results.
- While an edit solves, the app shows the results of the state before it (another project's never). They are dimmed and marked "Updating…" in the results columns, the drawings and their cards, the Steering and Assembly pages, and a balancing load's torque. Some places are not dimmed: the status marks in a module's list of parts, the figures an editor shows (a linked bearing's solved loads, the shaft's strength panel), the dialogs and the report. The header's "Updating results…" covers those. A solve that lands within 0.15 s is not marked at all, so that the page does not flicker (R-943). The system modes and the steering duty, computed on request, follow the inputs rather than the project: after another project opens they say "Out of date" until computed for it (never counted as computed), and a steering duty computed for a project of the same inputs counts for it. After a licence change while the Overview is open, its power flow can show the plan before's figures unmarked for as long as its own job runs.
- The keyboard-access check (`apps/desktop/scripts/keyboard-access.mjs`, quick CI) is a CI aid, not a proof: it reads what the templates and the components' metadata say, not what runs. It does not read metadata written in shorthand: a host's role and tabindex given as `host: { role, tabindex }` (the host then counts as no control, so Escape alone passes its click), or `{ templateUrl }` (that template is skipped). It looks a template constant up by its name in the file, so a constant of the same name in a narrower scope can be read instead. And it counts what an `<ng-template>` holds as shown with what is around it, so a list's keyboard owner there passes the list's items even when no outlet renders it. The app has none of these patterns today; the browser tests check real keys on the main paths.
- Shaft drawing: a bearing against a shoulder takes its seat class only where the bearing's bore chamfer r_s min is known: ISO 15's diameter series 0, 2 and 3 of single deep groove, self-aligning, four-point and double row angular contact ball bearings and cylindrical and spherical roller bearings. A single angular contact or tapered roller bearing (whose front face's chamfer is smaller and whose facing the model does not tie to the shoulder), needle and thrust bearings and other plans leave that seat plain, with a hint to check the chamfer.

- The 2D shaft view's bearing half-sections, all for a later release: on a rack shaft, the room kept for a bearing's texts is worked out at its support's undisplaced position, so at another rack position a larger seat can bring them closer to the view's edge; below about 132 px of view height there is no room left for those texts and the radial scale jumps to the shaft's own; on a thin shaft a bearing's texts can overlap the load labels next to it; the ground hatch and the texts under a support are centred on the support, not on a wide or off-centre pair; and the view's tests check that a half-section is reused while nothing it is drawn from changes, less that each change draws it anew.
- Searches in a browser: Make it pass, What matters most and the design search run over the pool of engine workers; the bearing finder, sweeps, the preload study and the comparison of variants still run in one worker, one design after another (the desktop app and the command line run them on the pool). A design that runs on and on in a browser has no work budget there (WebAssembly cannot unwind): the search's time budget ends it, its helper stopped, and Cancel does at once.
- A search on a small project whose designs solve in milliseconds gains little from more threads: its own steps between the designs (the project solved as it is, with detail) take most of its time.
- A design's work budget (50 times the project's own solve, at least 5000 costly steps) is measured on the project as it is, whose solve can find a result remembered from an earlier search and count less: the budget then comes out lower than it would be fresh. Only a design near 50 times the project's work could be cut where it would not be otherwise (and then shows "no result", never a pass).
- Make it pass's keep takes any path: one that names nothing (a typo, `sections.S9`) keeps nothing and is not refused.
- A design search on several threads may also solve, ahead of it, designs it then does not evaluate (at most `maxEvaluations` more); its result and `evaluations` are those of one thread.
- `fix --apply N --max-seconds S` is not refused, though a search stopped by its time can number its proposals otherwise than the run before: apply a proposal by its `commands` (the guide says so).
- `--threads` and `AXM_THREADS` are not capped at the computer's cores (the app's setting is): a large number starts that many threads.
- Cancel stops a housing FE solve between its steps, not inside the factorisation (the matrix library cannot be interrupted): on the largest model measured that is about 2 s on 4 threads (5 s on one), on a model near the 4 GB limit far longer (a later release).
- A refusal's fix buttons, all for a later release: for an item whose id does not start with a capital letter, a suggestion is found only when it changes that item itself; in a locked project the buttons are disabled without saying why; the item solves a suggestion takes while something is refused are not counted in the solve-work table; the ball screw's lead table follows ISO 3408-1's nominal leads, not yet checked against the standard. A planet bearing whose outside diameter reaches the planet's root diameter is refused and its field marked, with no button (no number to search).
- Spherical roller bearings: the rollers' end and shoulder contact (a guide ring, ribs) and their skew are not modelled; ISO 16281:2025 does not rate them either.
- Four-point contact ball bearings with balls in four-point contact (a radial load) have no rating in ISO 16281:2025: their life is shown and its check is not determined.
- ISO 281 rates a bearing at a 45° contact angle differently as a radial and as a thrust bearing, so lives jump there (ISO 16281:2025, Annex F).

- One-way bearings, for a later release (Opus review of ut4-bearings): a single angular contact ball bearing free to tilt (on a shaft) whose axial load along its facing is below F_r·tan α (as a floating bearing, or locating with next to no axial load) is rated with its ring tilted to balance it, as the solver finds it, and passes; a real one needs a second bearing or a preload to keep its rings together. Held square, or a tapered roller bearing, it is refused. A thrust ball or roller bearing whose axial load reverses between load cases is rated each way (no field says whether it is single- or double-direction). The axial direction check fails only a load against the facing. On a floating support the bearing finder keeps a one-way candidate's preset facing and does not turn it to face the support's spring preload (`springPreload`): facing against the spring by more than the check's allowance (5 % of F_r·tan α), the candidate fails its axial direction check and is not offered as passing, though turned round it could pass, on the safe side (Opus review of the ut4-motor-critical-speed merge). Turn such a bearing round by hand (`facing`).
- A roller bearing counts as having its geometry entered only when its roller diameter or count and its roller length both differ from the estimate it came with: a maker's value equal to the estimate, or a roller bearing imported before 0.3.0 (whose estimate does not record its roller length) until it is imported again, keeps the lower of the two lives, on the safe side. A catalogue file with the elements but no roller length records the estimated length (`estimate.elementsEntered`), and a bearing whose designation leaves its preset keeps the preset's estimate (each one of a chain of renames, `earlierEstimates`), so neither counts as entered on an estimated roller length. A roller bearing with no estimate known (imported or renamed before 0.3.0 under a name of its own) counts as estimated.
- The report preview (its pages only; Print, PDF and Word lay out their own): with a cover page, a cover longer than a page (a very long description) is not broken between its lines, so lines of it sit under the grey gap between two sheets; and a table longer than a page that starts part-way down a page starts on the next one, its heading left behind (a later release).
- A teeth end (`teethEnd`) is rated by DIN 743-2's notch factors of a shaft with involute teeth without a hub (4.2.2) and an eq. (9) shoulder or groove of its shape, not by a published method for a hob's run-out, which no source at hand gives (see "Shaft strength references"); the mesh's own load on the teeth is the rack mesh's or gear pair's tooth-root rating. A worm's thread end takes none (a strength notch instead). The Shaft module's 3D draws a rack pinion's teeth cut into the shaft; a gear pair's pinion cut into its shaft is drawn there at its root diameter as before (the Assembly and Steering views draw its teeth). The 3D views draw a single toothed section's ends; teeth over several sections at the root diameter are rated over all of them but drawn on the one at the mesh.
- A hob run-out longer than the section it runs into (the hob's arc reaching the next section) is not checked, and its radius has no upper bound: the "Run-out radius" point then reads a gentler shoulder than the steps make, but the toothed shaft's own point, which no radius changes, is rated beside it. A 5 mm neck behind a 4 mm step (13 mm of run-out at 25 mm) is such a case: give the neck its length, or a relief groove.

- A placed key's keyway (see "From 0.3.0") is listed in the shaft's strength results at every point it is assessed, each mesh station along the key and the middle between two, which makes the list long for a shaft with long keys (the Drive shaft example's two keys give some 120 points); its lowest safety is the one the checks take. Between two sampled points the shaft's own weight may lift the moment by up to $q \cdot h^2/32$ (q its weight per length, h an element's length): a strength check with a placed keyway point whose safety that bound could take below its limit (through the lower of the bending amplitude strength and fatigue limit, the yield strength or, for a hard layer, its cracking criterion) is not determined (a warning, never a pass), saying so; on a shaft that does not rotate, where the missed stress is a mean stress whose effect is not bounded, a fatigue check with a placed keyway under the shaft's weight is never determined (take the key off the shaft and enter its keyway as a notch); refine the mesh (`settings.maxElementLength`) or enter a keyway notch where the moment along the key is largest. The exact search between the points comes in a later release.
- Bug reports: the inbox's source filter finds only reports that name a source, so those that arrived before 0.3.0 show under every source only. The reproduce check opens and solves the attached project only, and decides only from an error code (the report's) or an engine crash: it says "fixed" only when the code is gone, the report's command (or its last step) is a plain solve or validate of one named file (no pattern, no check, no variant, case or other option but the output's), and the run opened the project without warnings (none of newerFile, ids renamed), the reported code is the engine's (not a file's, the account's or the app's engine host's, such as io, signInRequired or native) and it computed everything (no solve error, nothing incomplete, no item's error); a wrong result reported in words alone, or another operation (an export, an edit), is "could not run", with the engine's output beside the report's expected and actual for the administrator to judge; it never says fixed on a guess. After a crash or freeze in one version and an update to another, no report is offered (it would go with the new version's number), nor when the log cannot tell which session ended badly (the run's mark names no session the log has, or more than one, as when two tabs or desktop windows ended at once; or a freeze kept while the page closed names none). On the web a run counts as ended badly only once its mark has had no beat for a minute, so a tab that crashed and was reloaded at once is offered its report at a later start. A freeze report leaves out the engine crashes the desktop app logged (they are the process's, not one window's); a freeze the browser or the system caused (a hidden or frozen tab, the computer asleep, a clock change) is not offered, so a real freeze that begins while the page is hidden, or overlaps sleep, is not offered either; and a web page restored from the browser's back-forward cache no longer marks its run, so a browser crash after that goes unnoticed. Both for a later release. App Check is not on yet (it needs a reCAPTCHA key and a privacy-text change; see App Check on the web above): until it is, the daily limits per install and per network address are the only protection of submitFeedback against abuse. The on-device log of the web app is one per browser profile: two tabs writing at once may drop the other's latest entries.
- A ball screw on its shaft (`screws[].mounting`) loads it with the nut at one place (`mounting.position`), not along its stroke; its end fixity is checked against the bearings' places and axial roles, not their tilt stiffness, and its unsupported length is not compared with the bearing span. On the safe side, as the screw's states rate the shaft: a load step is taken as driving (the larger torque), and the screw shaft's fatigue (and that of every shaft joined to it) takes its whole torque and axial force as fully reversed over two or more cases even where they never turn round. Refused rather than combined: a mounted screw with a motion profile (its ramps' bearing damage needs the life integrated over the speed), one with steps of its own in a project with load cases or steering strokes, and a second screw with steps of its own beside one whose steps are the cases (a later release).

- A ball screw whose nut turns (`turns: "nut"`) is mounted by its nut's shaft; the screw, which travels, is not one of the project's members, so what it pushes takes its thrust only in the user's own loads. A step the load drives back on a mounted screw whose shaft is joined to another (coupling, drive, gear pair) is refused: a load case cannot turn the train's power round (a later release).
- A mounted ball screw that does not derive the load cases (one step, or linked to them) is the shaft and turns at its speed: entered at another speed (0 too) it is refused, not rated at its own, whose critical speed, d·n and life would be at a speed it does not turn at; rating it at the shaft's speed is for a later release.
- A project saved before the driving side (`drivingSide`) reads as left-hand drive whatever its gear: the field is the side the geometry is said to be, and switching mirrors from it. Copies of the Steering rack, Column EPS and Dual-pinion EPS examples saved before had the right-hand drive gear's mirror image (their column ran forwards, to −z), so they read left while their gear is not. Tell the side from the gear: its steering pinion towards the rack's left end (x = 0) with its column rising rearwards (+z) is left-hand drive. The switch also changes the inputs checker remarks are about, which then read as outdated, as after any edit.
- A column-EPS gear's input switches from the column's torque to a rack force (`setGearInput rackForce`, the Steering module's Column EPS gear page), but not back: `torque` is refused with what to do (remove the steering system, put the torque on the pinion shaft by hand). Carrying a rack-force state back exactly (its rack friction as fixed loads, which a later switch, the static peak and the misuse states must not take again) is not done yet. The switch is refused either way in a project with design variants, whose designs would each need their own state (a later release).
- The same switch is refused where the rack mesh takes its torque from the train and the rack back-drives a pinion mounted to drive it (the train's power flows from the rack): the state would take the mounting's sense. Set the steering system by hand with its state's rack force and rack speed signed as the rack is loaded and moves (a later release).
- A column torque of 0 on a Column EPS gear's pinion shaft hides the Column torque field in the Steering module (its load is still there): enter another torque in the Shaft module, or undo (a later release).
- A bearing against a shaft shoulder, or one whose retaining ring groove adjoins its face, gets no seat tolerance on the shaft drawing: the model holds no bearing chamfer r_s to clear the fillet against, and counts the groove as cutting the seat. Every shaft example shows it since their bearings abut their shoulders (a later release).
- Find a bearing and the seat's list of fitting presets keep a support's centre: a wider bearing than one against a shoulder crosses it and is listed as not fitting until the support is moved (a later release).
- The gear-seat check takes a worm's threaded length as DIN 3975's estimate ((11 + 0.06·z_2)·m_x for one or two starts, (12.5 + 0.09·z_2)·m_x for more), as the 3D view draws it: the model has no field for it yet (a later release).
- A file axomek or the app replaces (a project saved, a report or export written over an earlier one; not a PDF report, see the next point) keeps the earlier file's mode, owner, group and access control list: on Linux its POSIX ACL (not the folder's default), on Windows its ACL. On macOS its extended ACL is kept by code no released build has run yet (there is no macOS build); on other Unix systems an ACL is not looked at, so there a file in a folder with inheritable ACL entries takes the folder's. Keep such files in a folder without them.

- Reprinting a PDF report (`axomek report --pdf`, the app's PDF export) replaces the file with one that takes the folder's permissions (mode and access control list), not the old file's. A symbolic link at its path that leads to a file is replaced by the report, not followed (the file it leads to is left alone); one that leads to a folder, a device or nothing is refused, as are a folder, a device and `/dev/stdout`. Keep a report others may not read in a folder they cannot read. Keeping the old file's permissions comes in a later release.
- Ball screw nuts are end-cap nuts, each thread start one circuit running the nut (each half of a split nut). A nut with tube or internal-deflector returns (several circuits a start, each over a few turns) is not modelled as such, for a later release: enter its total loaded turns per start. Its ISO 3408 ratings and rigidity are then right (they take the loaded turns i); the ball model's load per ball and life differ by a few per cent from those of separate circuits (see "From 0.3.0").
- End-cap nuts, small points for a later release (review of ut3-endcap-nut): a format-6 file of a nut refused then for other reasons than its circuits (more than 10 turns a circuit, more than 20 circuits) opens rated, its note saying its ratings are as before; one of 0 circuits opens with a note though it is still refused. A nut of nearly 3000 balls as circuits (e.g. 20 of 9.4 turns, 16 balls a turn: 3000) can round to a few more as one circuit a start (3008) and be refused. A variant's override that never applied (its screw not in the base) is rewritten as empty when such a file opens. Only the shown design's screws get an opening note, not another variant's. A migrated nut's turns a start are c·t/n as computed, unrounded (3 circuits of 3.3 turns on 3 starts: 3.2999…), which can round its balls a start one down (147 for 150; its ISO ratings keep i). The warning of a second half's turns on a nut without halves also shows without the ball model, where the field is hidden (its fix clears it). A refusal's formula shows a bound without its unit (10 ≤ α_0 ≤ 80, meant in degrees). Switching a nut between one piece and two halves says what it did to the turns and the ratings in the command's message (the CLI, MCP), not in the app's inspector, where only the turns, C_a and C₀a fields show it; a second half's turns entered on a nut of one piece before switching it to two halves are cleared by a switch that splits the turns itself (equal halves; one with the turns given keeps it, warned while unused; halves of different turns joined and split again come back equal: halves of 3 and 2 come back as 2.5 and 2.5, the entered C_a times (2.5/3)^0.86 = 0.86 and the life about 0.6 times (more conservative), while the geometry's rating, now of 2.5 turns not 2, is about 1.2 times; undo restores them; an edit of the base's C_a or C₀a alone, not switching a nut, reaches a variant that follows it whatever that variant's nut, as any value does; turns given with a base switch reach a variant that follows only them, its own ratings not scaled to them, as a turns edit alone would, the "C_a above the geometry's rating" warning saying so); a join scales the ratings from the larger half's turns, where the engine rates halves that differ by the smaller (on the safe side); in a variant, a reset of its nut, preload or raceway that switches its nut splits or joins its own turns as the switch does, and scales its ratings. The catalogue C_a and C₀a a switch scales are ISO 3408-5's scaling of what was entered: give the maker's ratings of the new nut where they differ. A nut without the ball geometry has no turns to split, and a switch does not scale its ratings (as before): give a half's. The 3D view's tests still lay balls out as several circuits a start, which the engine no longer gives.
- A single nut preloaded by a shifted pitch between its two starts, one circuit each (ISO 3408-4 clause 5.1 c), is no nut type of its own: two starts of a preloaded single nut are rated as a lead-offset nut of two halves, each with both starts. Enter it as a lead-offset nut of one start and the same lead, whose ISO ratings and life are the same; but its nut body's rigidity (eq. 9 with the length i·Ph rather than i·Ph/2) then comes out twice the two-start nut's, too stiff, and the ball model's sharing along the nut differs: take the rigidity from the maker. For a later release.

- A steering's ball nut (`steering.system.assist.ballNut`) is no radial support of the rack itself (0.3.0): the rack needs a support there (a bushing with the nut bearing's stiffness, as the "Belt-driven ball-nut EPS" template has). The torque the nut passes into the rack's thread is not put on the rack (it is held against turning by the pinion's teeth and the yoke, whose ratings leave it out), and the nut's travel is not checked against the threaded length: keep the thread long enough for the nut over the rack's travel. On-centre, the feel's friction torque takes the preload's and the nut bearing's drags but not the motor's, which only the return force counts; it takes the nut's drag with the motor idle (the bearing's friction under that drag at the creep speed), so where the assist curve gives torque within the friction band the band comes out a little wide (a 1000 N preload on the template's nut, before its bearing's drag was counted: 0.601 N·m against 0.591 N·m solved at the band's edge). The steering's balance counts the friction of the bearings linked on the nut's shaft only (as a drag at the nut, see "From 0.3.0"): a bearing linked on another shaft of the steering's drive (the column, the pinion, the motor) is in the power loss but its friction is not in the driver's and the motor's torques, which come out low by it (no example links one). The nut bearing's friction in the balance is its rating's as the solve rates it (its housing's seats moved as the solve moves them); should the two differ, the solve warns at `steering.system.assist.ballNut`. A bearing linked on the nut's shaft whose outer ring sits in another shaft (a bearing between shafts) refuses the steering's state: its friction turns with the two shafts' relative speed and may drive the nut, which the balance does not hold; seat it in the housing, or unlink it to take `bearingDrag`. For a later release. Its thread is taken right-handed (a left-hand one only turns the nut the other way). Without a duty its screw is rated in the steering state alone, so its life is that state's; a steering duty with strokes rates it over the duty's states (0.3.0). The rack's misuse events leave the nut's force out: the meshes hold the event alone, on the safe side for their teeth and the bar between them and the joints, while the nut and its bearing are rated in the state only.
- Modules left out of the verdict (`report.contents.exclude`), until a later release, all on the side of saying more: a verdict whose kept checks pass close to their limits reads `partial`, not `close`; a torsion bar's "not checked" line and the first shaft's issues at `settings.*` or `rack.*` stay in the verdict when steering or the shaft is left out; a module switch's issue count is that module's alone, which can differ from the verdict's while others are off; `axomek variant compare` and the design search (`axomek optimize`) count the checks left out; a matched pair's warning (`pairs.*`) stays in the verdict when its bearing is left out; `axomek tree`'s root has no partial state (the nodes left out say `leftOut`).
- A project with parts left out of its verdict whose kept checks pass reads "Pass, parts left out" in Your projects (its chip and Status filter), its preview and Home also when it has warnings beside its checks (the report's Warnings, a problem among them or not) or a check close to its limit: the chip names that the pass is not of the whole project, not the warnings, as `axomek summary`'s `partial` does. The Overview and the report list the warnings. A chip that says both comes in a later release.
- A Hertz line contact loaded to the ends of its shorter cylinder without `edgeConcentration` passes its pressure and shear checks on the laminae's pressure, with a warning: the end faces raise the pressure there, which only the ISO 16281 edge factor (C.2, or a crown) takes. A point contact cut off by a roller's ends fails its checks (0.3.0); the line contact's edge rating comes in a later release. Turn on `edgeConcentration` or crown the roller.

- A tandem pair takes ISO 76's double-row static equivalent load (X0 1 and twice the single row's Y0) in Palmgren's load term and its thermally permissible speed, as an O or X pair does. ISO 76 takes a tandem set's two bearings each with the single-row factors at its share of the load, so the pair's P0/C0, friction and heat are overstated and its nθ understated: on the safe side.
- A plastic gear pair without a peakTorque has no peak load check (VDI 2736-2, 3.3): its peak safety is listed "not assessed" (no peakTorque given) against `limits.minPeakSafety` 1.5, so the project never passes `axomek check --strict`. Give the pair's `peakTorque`, or clear `limits.minPeakSafety` when no peak above the nominal torque occurs.
- A gear pair's mesh loss (ISO/TR 14179-2) takes the oil's viscosity at the pair's `oilTemperature`; a pair without one (every pair saved before it was an input) takes the oil at 40 °C, so in a hot sump its mesh loss comes out about 11 % low (ISO VG 220 at 95 °C): the efficiency slightly high, the heat slightly low. Give the oil's working temperature. The formula's uncertainty from the flank roughness is larger.
- A gear oil under about 6.5 mm²/s at 40 °C has no viscosity–temperature line at viscosity index 95 (ISO 2909): its scuffing check is refused at `viscosityIndex`, and with it the pair's curves, tooth contact and root FE, until the check is removed or the oil's own higher index is given. No gear oil is that thin; a later release keeps the other analyses.
- A bevel pair's entered single pitch deviation f_pt of exactly 0 gives an accuracy grade of −∞ for K_v by method C (K_v 1, with a note that the grade is extrapolated); a measured f_pt is never 0.
- The full-round limit of the basic rack's root radius is checked for cylindrical pairs, not yet for a rack mesh's own rack.
- A plastic worm wheel in intermittent duty takes the lower S_Hmin 1.25 on the checkbox alone. Above its material's continuous temperature limit (PA 66: 120 °C) it is not determined up to the hottest data curve (180 °C): run time and pause are not inputs yet.
- A plastic worm wheel's flank band rates the coolest temperatures at the whole life's load cycles, which errs on the safe side for a wheel that warms up within minutes. The data start at 20 °C, so a colder start leaves the flank not determined rather than rated (a cold-start rating below 20 °C, from moduli and flank strengths measured colder, is for a later release; the Column EPS assist gear example shows it), and with the lowest wheel temperature left empty the band starts at 20 °C: a colder start is then not looked at. A given wheel modulus E_2 holds at ϑ_M only, so its flank is not determined unless the lowest wheel temperature is given equal to ϑ_M.
- For a later release: a plastic cylindrical gear (VDI 2736-2) is not checked for a cold start. It rates its flank at its computed flank temperature only, though a cooler plastic is stiffer and takes a higher flank pressure faster than its strength rises (steel on PA 66: S_H at 20 °C about 0.73 times that at 70 °C), so a pair that passes warm can fail at ambient. The plastic worm wheel is rated down to its lowest wheel temperature; the cylindrical gear should take the same band.
- A ring gear's STEP solid is cut by its shaper cutter, so with a profile shift ($x_2 \neq 0$) its root circle differs a little from the report's nominal root diameter $d - 2 \cdot m_n \cdot (h_{fP} - x)$ (about a tenth of a millimetre on the diameter). The rating takes the nominal one.
- `axomek gear step` refuses an undercut pinion (z 16 or 17 at x 0) that the rating accepts: its root is not modelled.
- With gravity on, a shaft placed by a bearing in another shaft (seated in it) is refused when the two are solved together and its frame is turned by what places that shaft (a universal joint, a worm pair, a rack mesh): the model of shafts solved together leaves the bearings between them out, and with them that placement (0.3.0). Place the shaft by another connection or at coordinates, or switch gravity off.

- Bearings on their own (none linked to a shaft) have no system life while one of them is "not set up yet": the set's life is not shown rather than shown without it. On the safe side.
- A bearing "not set up yet" makes no check, also when it has limits set (`minLifeHours`, `minStaticSafety`), and the verdict does not name it: it has nothing to rate until it has loads and a speed or a shaft support.
- A new bearing (the app's + Bearing, `addBearing` without loads in its patch) starts with a 3 kN radial starter load (Fy) of its preset, shown and editable in its Loads section and its summary. Without a speed it is then not rated (not "not set up yet") until a speed is entered or the load cleared; on a shaft support the shaft's loads replace it.
- A steering duty's work is limited, so that neither its derivation nor its rating can run without end (0.3.0). Bounding the linked modules over the strokes stops at a fixed amount of work (the setup that ran for hours stops in some 35 s natively; every derivation the property tests' random linked setups and the steering tests decide takes at most some 20 s; the limits allow at most some 2.5 min, for a gear pair whose true-safety search is slow throughout): past it the load cases are the strokes' own, without the linked modules' refinement, every linked module's checks over them are undecided (never a plain pass) and `steering.duty` warns why and what to change. The tooth-by-tooth rating stops at its own limit: past it the duty is refused at `steering.duty`, saying what to change, as a search cut short would understate the teeth's damage. A duty on a constant lead, or with its tooth geometry within ISO 6336's validated range, rates in a few seconds (the steering-duty template in 2.7 s natively). Only a duty with FE root assessments (the geometry outside ISO's range) over a variable lead is slow: each new lead needs its own assessment (some 130 ms), so its rating can take some 52 s natively before the limit, and with its derivation a solve of such a duty up to about 90 s natively, longer in the browser; a 0.1 mm window there takes some 22 s. To make it fast, use a constant lead or bring the tooth geometry within ISO's range.
- A bearing whose eC·Cu/P stays within about 0.01 of 5 (where aISO leaves its cap of 50) along a whole stroke keeps the duty's search for that regime change refining: one such test case needs some 168 s natively per derivation, so it stops at the work limit and the bearing's checks over the strokes are undecided rather than split at the change. The app computes the duty on request, in a worker of its own with progress and Cancel (0.3.0). The work limits stay as they were: they are counted, not timed, so the app's duty equals `axomek solve`'s, and in the browser's engine (some 1.2 to 1.5 times the native time measured) the tooth rating's limit already takes about a minute and the derivation's some 3 to 4 min, past the minute the on-request duty aims for. Keying the FE root assessment by a rounded lead is not done: it would change the results of duties that finish today, and no bound puts the rounded lead's FE/ISO factor on the safe side. Such a bearing is not decided by a direct bracket either: along a part its eC·Cu/P follows the force one way and the speed (through the viscosity ratio) the other, so it can dip below 5 and back between two samples, which only a search for its least value finds, and that value is not provably single-troughed. Both stay for a later release.
- The app's computed steering duty belongs to the project object it was computed for: showing another shaft (which orders the project anew without editing it) makes it out of date, and a run under way is cancelled (0.3.0). On the safe side: Recompute duty, or go back to the shaft shown. Keying it by the project's content comes in a later release.
- A STEP shaft import into a rack is refused: its teeth (the toothed section's flat) are not read from the file.

- A gear's STEP solid has nominal teeth: at the profile shifts, without the backlash of generating shifts (`details.rootShifts`), a protuberance tool's residual undercut or flank modifications (tip relief, crowning), which the rating takes. A double-helical gear is refused, as the project does not hold the gap between its helices: export one helix (doubleHelical off, half of `faceWidth` and of both `faceWidths` when given) and mirror it in CAD. In a later release: the gap as an input and both helices in one solid.
- Where a rack mesh is given its pinion torque and no efficiency (its own rating then takes 0.92), the steering duty rates its force tables at the default for a known rack force, 0.85: a stroke the duty card starts at the mesh's rated rack force is rated 0.92/0.85, 8 %, above the mesh's own driving load, and linked items whose loads follow the rack's actual force (a ball screw) are scaled 8 % above it too. On the safe side; exact where the mesh's efficiency is entered.
- The steering duty rates every stroke's force table as the pinion driving the rack (F/η), also a restoring stroke's and a return pass, where the road may drive the rack and η′'s direction would rate the teeth at F and the pinion at F·r·η′: on the safe side. Misuse events are rated at F.
- Over steering-derived load cases a shaft's FKM damage check (with a number of cycles) is undecided where it passes when the shaft's loads follow the speed (0.3.0): every shaft of a mounted stage (a gear or worm pair, a rack mesh, a drive), whose mesh efficiency is rated at each case's speed and enters a member's torque, and whose belt's pull rises with the square of the speed, and every shaft joined to one by a coupling or a bearing between shafts. The cases' chord split bounds that damage only where the stresses follow the force alone. In belt-drive with SH1's FKM over 800000 cycles and one-way strokes over a 126–130 mm/rev lead, the cases passed 6.681 at 6.681179 where the strokes give 6.680947.
- A steering duty whose strokes reverse the rack force (a return pass against a resisting load, a signed load below zero) is refused at `steering.duty` beside anything the load cases reach that a case cannot reverse (0.3.0): a mounted gear or worm pair or rack mesh, a drive, a coupling, a rotor, a bearing with loads of its own, a gear pair rated over the cases. The derived cases reverse only the entered shaft loads and ball screw forces, so the others would be rated at the forward loads on the reversed passes (a rotor's torque, unreversed, cancelled the reversed input in one review's case: 18.444302 where 9.222151 is right). In gearbox-single-stage with a rack mesh and resisting strokes with return passes, BR1 passed 5.5 at 5.778100 where the reversed pass gives 5.260948. One-way strokes are rated; reversing stages over the duty come in a later release.
- Over steering-derived load cases a linked gear pair is undecided where it passes (0.3.0) when its KHβ comes from its solved shafts (`misalignmentFromShafts`), or when the connections resolve the project (a shaft speed, a train-mode stage or a coupling): the solve then rates each case's bin at the torque and speed that case's train gives the pair, the duty's bounding at the pair's torque times the case's factor. A standalone pair, or one in a project with no shaft speed, train-mode stage or coupling, is bounded as before. (A project of given stages with a shaft speed is marked too, where the two ratings agree: on the safe side.)
- A gear pair linked to the load cases whose nominal rating holds but whose rating over them is refused (a case's bin it cannot rate, such as a negative factor on a given pair) is listed in the load cases' summary as rated over them, though it has no spectrum checks and its error says why (0.3.0; Sol review of R-938).

- A steering block (strokes or misuse events) beside a rack member or a rack mesh mounted on one, without a steering system, is refused at `steering` (0.3.0): without a system the duty rates standalone rack meshes. Its derived load cases would load the rack member, whose yokes change regime and whose spring braces hold or give way with the force along the strokes, and the cases do not bound those states between them; and it would rate a mounted mesh's teeth without the rack's lift at the contact. The `steering-rack` template with a stroke now gives that error and derives no load cases. Give the project a steering system (the duty is then rated over its balance), or rate the duty in a copy whose rack meshes stand alone (the `steering-duty` template).

- A steering duty over a steering system (0.3.0) rates the rack meshes' teeth, and with strokes the system's shafts, the bearings linked on them that take the load cases, its worm pairs and its ball nut over its states (see Steering system). Not rated over the duty (a later release): a mounted gear pair (rated at the system's `state`, its checks not rated over the duty, never a pass: its rating takes its solved shafts' KHβ, which the states do not give it yet), and likewise fits, keys and housings. A rotor's air gap is not rated over the states (its band between them is not drawn; never a pass: rate it at the duty's heaviest state). A bearing's warning over the states (a minimum load undecided at its valley) is never satisfied, and its gyroscopic slip (the high-speed model's) is the system's state's, a pass not rated over the duty. Beside such a duty a design search's metrics are refused but the shaft's mass (their values over the states have no bound between them, and some items' results are the system's state's): search with requireChecks, whose checks take the bands. The ball nut's life is the states' Miner sum (as a bearing's), its static safety the worse of the states' and the system's own state's. Left at the system's `state`: the couplings, a bearing with `useLoadCases` false, and everything at a misuse event (the events rate the pinions only; the elements keep their own `peakFactor` for a static peak, which over the duty multiplies every state's loads, as over load cases). A worm pair and a drive take the worst of their states, each rated as if it lasted their life: on the safe side, as DIN 3996 holds no load spectrum here. A shaft's states are each rated with their stresses fully reversed (as a mounted ball screw's states are), on the safe side where a stroke does not turn round. The band of a bearing's checks is drawn from its life, static safety, contact stress and thermal speed in the states, relative to each check's utilisation. Two supports in seats of one housing (whose seats push each other with the bearings' loads) leave the elements unrated over the duty (a warning at `steering.duty`, their checks never a pass): the states do not couple the seats again (a later release). The project's own load cases beside such a duty rate the shafts and bearings instead (a warning at `steering.duty`), and their checks are then not rated over the duty (never a pass): remove the load cases to rate them over it. The elements' states are some four times the duty points (284 for the Dual-pinion EPS's duty), each a state solve and a solve of every shaft: at most 1200 (`system_cases::CASE_LIMIT`) and at most 12,000 solves (`WORK_LIMIT`: each state one, and one for each shaft and each bearing on a shaft; some 25 s natively and 40 s in the browser), past which the elements are not rated over the duty (a warning at `steering.duty`; their checks never pass) while the teeth are. Measured under load on the build machine (one process, natively): the Dual-pinion EPS's duty 9.3 s for the teeth and 19 s in all; the Belt-driven ball-nut and Column EPS examples given its duty 2.4 s and 2.8 s for the teeth, 7.0 s and 5.2 s in all (168 and 88 states); the browser's engine took about the same under that load (17.9 s, 7.6 s and 6.3 s). It refuses a stroke's `torque` table (the system splits the rack force, not a pinion's torque); a rack mesh off the system's rack is not rated over the duty (a warning; it keeps its rating at its own load). Each duty point is a state solve (some 15 to 30 ms natively), at most 4000 per duty. Rating the teeth of two pinions is the slow part: the Dual-pinion EPS template's duty (a parking, a manoeuvring and an on-centre stroke, both ways) takes some 4.7 s natively and 7.3 s in the browser's engine on the build machine (some 20 s natively on a laptop, 4 s of it the state solves); the app computes it on request, with progress and Cancel (0.3.0), and solves each edit without it. Positions past the rack's travel are refused (each point is a state of the system, within its stops). The lift is rated within 1e-4 mm of the states' between points (the contact ratios within some 4e-5), not raised; the forces are raised to bound theirs. Misuse events are rated at standstill (vehicle speed 0, forward efficiencies, no friction).

- The steering duty rates only the bearings linked to the system's shafts' supports. A support that names a bearing by its designation only links none: nothing rates it, at the system's `state` or over the duty; the duty lists it (`elements.unlinked`) and the summary names it as not assessed. The steering examples draw their bearings so: the Dual-pinion EPS's 6001 and HK2816 on the steering pinion and 6001 and 6203 on the assist pinion's shaft, the Column EPS's and the Steering rack's likewise, every other support of theirs (the column, the intermediate and input shafts, the motors) plain. Only the Belt-driven ball-nut EPS links a bearing, its nut bearing BR1 (see "From 0.3.0"). So the Dual-pinion EPS's duty rates no bearing. The designations are drawn, not sized: linked as drawn (6001 preset, default grease), the Dual-pinion EPS's assist-pinion 6001 at S51 would not carry its parking strokes (static safety some 0.6). Sizing and linking the examples' bearings is for a later release; until then link bearings at those supports to rate them over the duty (a needle bearing such as the HK2816 as a bearing of your own).
- Over a steering system's duty states (0.3.0), all on the side of saying more or of reporting less, for a later release: a rack buckling span that exists in some of a span's states and not in others (the bar moves under the supports) draws no marginal band over that change; a shaft standing still at the system's `state` takes FKM's checks over a number of cycles at the worst state for all its cycles instead of their Miner sum; a state that gives the ball nut no thrust can leave its rating over the duty refused; the system's own state, kept among the worst-case checks, is rated with its mean and amplitude stresses where the duty's states take theirs fully reversed (a state heavier than every duty point is rated more leniently than they are); a refused duty marks every drive's checks, also those of a drive off the steering's train; a bearing's band for its checks other than life and static safety is drawn from those quantities' bands rather than its own; `atState` does not list bearings off the shafts or planet bearings (rated at their own loads), nor the couplings.
- Over steering-derived load cases a linked bearing's minimum load check does not evaluate the axial load's holding (0.3.0): a radial load below the minimum fails there even where the axial load keeps the elements loaded; a preload still holds, and the bearing's own loads keep the exception. Precise handling comes in a later release.
- Over steering-derived load cases the checks that fail at a quantity's low point (valley checks: a linked bearing's minimum load; its gyroscopic slip, which only the high-speed model gives, is undecided with it) are undecided where they pass (0.3.0): the cases bound loads and stresses from above, not their lowest values between the cases. A preload that holds in every case keeps the check, and a module at its own loads is rated as entered. The checks at a peak (stresses, lives, safeties, temperatures, speeds) are bounded. Precise valley bounds come in a later release.
- A gear pair's loaded tooth contact analysis loads the flanks with the nominal tooth force but opens them by the misalignment that ISO 6336 method C takes at Fm = Ft·KA·Kv (the shaft layout's fsh, the solved shafts' or a carrier's), so with KA·Kv above 1 its pressure map and its face load factor show more misalignment than the nominal load gives. On the safe side; the ISO rating is not affected. Both at one load come in a later release.
- A Nastran mesh is refused for a MAT1 of a density outside 500 to 25 000 kg/m³, or two MAT1 of different densities or stiffnesses, and for a GRID with its displacements in another coordinate system (CD), also when no solid element uses that MAT1 or GRID (a shell's material, a rigid link's node). On the safe side: the message names the card or says there are several materials; remove the unused cards, or export the solid part on its own. So is an SPC that moves its node (D not 0, an enforced displacement), and a file with several SPC sets and no case control choosing one: hold the node with D 0, or export the case control (SPC = n) with the mesh.
- Case-control commands outside the supported list are refused.

- A solid wall's hub whose rim lies past 80 % of the way from its bore to the plate's nearest edge (or to half way to the next bore's hub) is refused, with the largest diameter that fits: the mesh's ring between hub and wall lies there at most. On the safe side; a hub nearly as wide as the wall round it comes in a later release.
- A helical pair's loaded tooth contact analysis runs the contact lines across the face the same way whatever the helix's hand. The load across the face ($K_{H\beta}$) and the transmission error do not depend on it while the pinion's and the wheel's profile reliefs are alike, but the pressure map's slant and its peak pressure, which follows the flanks' curvature along the path, can differ a little for a pair of the other hand. Taking the hand into account comes in a later release.
- The unbalance response without shear deformation, read within 1/100 of an element from a node (through the element's shape functions), omits structural damping on the in-element bending term, so it reads high there: up to about 8 % at $\zeta = 0.2$.
- DIN 732 Table A.1's friction factors are applied to angular contact bearings below 22°, a range the table does not list.
- A rack mesh's lead field (the variable-ratio table) reads a bare number in the unit shown, mm/rev or in/rev, but refuses a lead typed in the other one ("60 mm/rev" with US units shown): the field parser has no kind for a lead yet. Type it in the unit shown. The parser change comes in a later release.
- A yoke's preload is not checked against the separating force of the on-centre states (the rack may rattle on centre with a preload below it): the yoke panel checks only its mesh's contact ratios at the lift and warns when the yoke sits on its stop. Compare the preload with the mesh's separating force by hand; the check comes in a later release.
- The steering duty finds a quantity's largest value over each part of a stroke from samples: 8 even ones, 12 toward each end and seven rounds of a finer grid about the largest. A smooth peak narrower than the even grid's step (1/9 of the part) and away from the part's ends can pass between them; the jumps of a module's regimes cannot (each is located and sampled on both sides). Strokes whose force tables have such narrow peaks are best given a point at the peak.
- The scuffing flash temperature's maximum (flash temperature method, ISO/TS 6336-21) is the largest of 65 points along each piece of the path of contact between the load sharing's kinks, refined by golden section between that point's neighbours: a second peak, narrower than 1/64 of a piece and lower at every point than the largest, would be missed. The load sharing's curves have no such peaks.
- A double-walled planet carrier's wind-up from its finite elements is taken on the safe side of its mesh's error: 2.5 to 4.2 % above that of a mesh refined without end in the tests (webs of 10° to 90°, 3 to 5 planets). A carrier its mesh does not settle on within 3 refinements, or that a finer mesh stiffens (a cheek held on a small bore or none), is refused with what to give instead (`carrier.torsionalStiffness`, or the hub's bore). The model has sharp inner corners where webs meet cheeks and holds the whole hub bore of the output cheek; fillets and a hub held over part of its bore are not modelled. A long closed carrier (webs all round, a gap of 400 mm) takes several seconds.
- **Add a steering** puts the rack's meshes in the steering's train and takes the torques off the driver, the shafts coupled to it and the pinions' shafts. A gear pair, worm pair or drive between the driver and the rack still rated at its own torque is left as it is. The steering then reads "not solved", and its error offers the fix (`torqueMode` `train`). Apply that fix, or set it in the item's module.

- Weld fatigue follows EN 1993-1-9:2005, and says so on its method line, in the report and in Show calculation. EN 1993-1-9:2025 (Figure 8.2b) gives the welded details below category 71, the default fillet weld root category 36 among them, a lower curve: the constant amplitude fatigue limit at 10^7 cycles and $0.585 \cdot \Delta\sigma_C$ instead of $5 \cdot 10^6$ and $0.737 \cdot \Delta\sigma_C$, and a cut-off limit of $0.369 \cdot \Delta\sigma_C$ instead of $0.405 \cdot \Delta\sigma_C$, so ranges beyond $5 \cdot 10^6$ cycles do more damage than the app counts. Until the 2025 edition comes as an option (planned for a later release), check a weld of a category below 71 that sees more than $5 \cdot 10^6$ cycles against that edition by hand.
- Weld fatigue rates the weld throat (and a full-penetration butt line's toe when the line has its own category), not the connected plates at the weld toes (EN 1993-1-9:2005 Table 8.5 details 1 and 2): the plate thicknesses are not inputs. The results and the report say so; check the plates' nominal stress range at the toes by hand, which governs a thin plate with large fillets. A toe check with the plates' thicknesses is planned for a later release. Without them, a throat larger than its plate, or a full-penetration weld drawn as two lines at the plate's faces (the plate counted twice), is not caught; the help says how to draw it.
- A weld group does not say which kind of joint a weld is in, so EN 1993-1-8 4.11's long-joint reduction takes the group's fillet and partial-penetration welds as one lap, its length L_j the largest distance between two of their ends (gaps included, in whatever direction the force passes), wherever that is longer than $150 \cdot a$ or 1.7 m, also where the clause spares a weld (no lap, a girder's flange-to-web weld, 4.11(2)) or two welds stand far apart in one group, and the smaller of $\beta_{Lw.1}$ and $\beta_{Lw.2}$: on the safe side. From $900 \cdot a$, where $\beta_{Lw.1}$ reaches 0, such a group is refused, also a weld the clause would spare ($a = 4$ mm over 3.6 m): give a larger throat, or design separate shorter joints. A weld group stands for one joint with all the welds of its lap: a lap drawn as several groups escapes the reduction, as no group sees the lap's whole length. A switch for a weld that is no lap is planned for a later release.
- Weld fatigue counts damage on the slope-5 branch for every range down to the cut-off, also in a spectrum of one block, where EN 1993-1-9 lets a constant-amplitude range below $\Delta\sigma_D$ do none: a truly constant load is rated on the safe side.
- A parallel key's keyway edge chamfers s1 and s2 (DIN 6892 eqs. (4) and (5)) are 0 unless entered: DIN 6885-1 gives none, the drawing does, and each lowers the depth the key bears by its size. Enter them from the drawing.
- A parallel key's K_λ is read from DIN 6892's Figures 4 to 6 at 14 values of Q_A and six of l_{tr}/d (within about 0.005 of the figures), not by the more accurate method the standard cites; its Annex D.1 and D.3 print 1.07 where the figures give 1.087 and 1.097, so the app's pressures there come out 1.6 % and 2.5 % above the examples'. A thin hub (d/D from 0.6), whose own strength DIN 6892 6.2.2 then asks to be proven, is noted, not assessed. The friction factor K_R of an interference fit is not in the app (the key is rated as if the fit carried nothing, on the safe side). Nor are the load direction change factor f_W and the peak load factor f_L (6.1.3), which lower the permissible pressure for a drive that reverses or peaks often: not on the safe side, so every key's and spline's result says that reversals and the number of starts or peaks are not assessed (from 0.3.0; user test RR1). Inputs for them are planned for a later release.

- A press fit's yield strengths are sized for steels only (DIN 743-2's K1 at the hub's outside diameter and the shaft's seat diameter): the cast irons and aluminium of the list keep their listed values, and a yield strength entered by hand is taken as the part's. A shaft larger elsewhere than at its seat is weaker than its seat diameter gives (DIN 743-2 takes its largest diameter), which only the shaft strength check takes. The size classes the app holds end at 100 mm for C45 +QT, 160 mm for the alloy steels and 250 mm for S235 and S355 (EN 10025-2:2019, Table 6, p. 27; its class over 250 up to 400 mm is for flat products only, Clause 1, and is not taken for a fit's round parts), and it holds none for E335 and the case-hardening steels' cores: beyond them a passing yield check is not determined until the guaranteed yield strength at that size is entered. The class is taken at the part's finished diameter, not at the size of the bar or forging it is turned from, whose class the steel's guarantee follows and which can be lower (an S355 hub of Ø40 from Ø45 bar: 335 MPa, where Ø40 takes 345; review of 0.3.2): for a part near its yield limit, enter the yield strength of the stock's size. E335's classes (Table 7) wait for an input of the stock's size for the same reason. A fit is not linked to a shaft, so its torque is not compared with the torque its seat passes (a placed key's is): a later release.
- A shaft–bearing coupling (Couple linked bearings) that does not settle is not rated. A bearing under radial load alone (no axial load) has almost no tilt stiffness inside its clearance, so its tilt has no settled value: the paint mixer of the release-candidate retest (a11y) settles with its paddle hub at 600 N but not at 500 N, where the upper 6206 still carries some 270 N. The refusal names the bearing and says so (from 0.3.2); a coupling that settles such a bearing (its tilt held where its stiffness vanishes) is for 0.3.8. Meanwhile turn Couple linked bearings off, or give that bearing less clearance or a preload.
- The shaft strength check (DIN 743) takes DIN 743-2's K1 alone, as the standard prescribes where the strength at the part's size is not known (6.2): for an alloy steel inside a size class it can lie above the class's guaranteed minimum (42CrMo4 +QT at Ø50: 748.6 MPa against EN 10083-3's 650 MPa), as in any calculation after DIN 743. Enter the guaranteed strengths ("strengths as given for the size") for a shaft near its limit. Press fits hold it to the class (see "From 0.3.0"). Keys of form AB, C to J and lift-out bevels are not in the app, nor a keyway shorter than its key: the key's length stands for both, where DIN 6892 takes the shorter for l_tr and K_λ. A spline's K_λ is still the estimate of 1 up to a length of one diameter, + 0.4 per further diameter: DIN 6892 is for parallel keys only, and DIN 5466-1's factors are not taken.
- A key's and a spline's yield strength (shaft, hub and key) is the material's value at 16 mm, whatever the part's size, as in 0.2.7. A larger part's can be lower (the release-candidate test's keyhub: an S355 hub of Ø90, 355 → about 314 MPa by DIN 743-2's K1), and the safeties shown are then optimistic. For a part over 16 mm, enter the supplier's guaranteed yield strength at that size. The app's Materials lists hold the steels' 16 mm values and have no field for it, so set it in the file or with echo `'{"op":"updateKey","id":"K1","patch":{"hubMaterial":{"name":"S355 at 90 mm","yieldStrength":314}}}' | axomek apply <file> -` (piped, as PowerShell 5.1 drops the quotes of an argument; `shaftMaterial`, `keyMaterial` likewise), under a name of its own: a material that keeps a listed name is shown in the app with the list's Re, though the file's value is the one rated. Taking it at size, as press fits do, is for a later release.
- An admin console tab opened before the release that brought `adminRecordingChunks` (0.3.3) cannot open a replay after it: it still calls the one-chunk functions, which that release's deploy deletes, so its player shows an error. Reload the console (review of replayspeed-033).

- A replay in the admin console comes with its details in the same answer as its first chunks (`adminRecordingChunks`). Its Storage reads stop 30 s after the call starts and the console waits 70 s, so when the admin check itself is slowed by Auth retries for more than about 35 s and Storage then hangs (on a chunk or on the listing of the chunks' sizes), the answer comes too late: the page shows neither the replay nor its details and keeps Delete off. Nothing is shown wrong or lost; open it again, or delete it from the Replays list, which reads no chunks (review of replayspeed-033, C2 and C4).
- One answer of `adminRecordingChunks` carries up to 6 MiB of chunks (about 8 MB on the wire). On a slow connection (about 700 kbit/s or less) the largest such answer takes longer than the 70 s the console waits, so a large replay does not open there (the player shows an error). Open it on a faster connection (review of replayspeed-033, C7). An answer of the details alone, and about 2 MiB of chunks an answer, are planned for 0.3.4.
- The admin console's replay player counts the last 50 ms of a replay as its end: Pause pressed there is announced as "Ended", as Play would start the replay over (review of replayautoplay-033, S2). Its end-to-end check of autoplay at 8× would also pass a player that ran at 4×, or at 2× on a slow run (its timing starts after playback has begun); the unit test checks the speed the console asks for (S3, S4).
- The admin console's Analytics › Updates takes production's release from the release log, the newest release it has seen (a pause's hold feed counts as none). The log keeps only when each release was first seen, so after a rollback of the web app to an earlier release (`docs/ops/bad-release.md`) the card still names the withdrawn one until a newer release is out; choose the version in the filter then (review of small-032). A release counts from when the log first saw it (the daily digest, the Releases page, or Analytics over a range to today, which records production's release first), so a past day around a release can name the one before; a release the Releases page first saw seconds before UTC midnight, during a deploy, can be dated when the page's other reads were in, the next day (review of updates-032).
- Analytics for a range ending today takes production's release from a live read of its files. A request made in the last moment before UTC midnight, while a release rolls out just after it, can show the new release's adoption for the day before (review of updates-032).
- A key or spline added after another in 0.3.1 or earlier took that joint's hub outside diameter (and step), which a saved file cannot tell from one entered: check `hubDiameter` of keys added that way. From 0.3.2 a new joint starts without it (DIN 6892's least favourable hub until given); a seat reference that takes it from the shaft is for 0.3.7.
- In US customary units the app and axomek leave a project's own texts (names, descriptions) as written wherever a message quotes them. A name that is itself a quantity phrase, such as a project named "35 mm bore", also stays in millimetres where a message happens to contain the same words: the value is right in the unit it shows, but it is not converted (review of small-032).
- A spring of stainless wire (1.4310) takes d_max with EN 10270-3:2011 Table 5's class T12, the widest for coiled wire, as the standard names no class to assume: a wire ordered to a tighter class has a slightly shorter solid length. Its notes say so.
- The press-fit method of the Fits module (DIN 7190-1) is written from the app's method notes: the text of DIN 7190-1 itself was not at hand to check it against.
- Compare variants shows no housing results (a housing wall's stiffness, stresses or modes) for a later release (user test ut4): a variant that changes a housing is compared by its shafts', bearings' and checks' results only.
- The shaft drawing does not draw a hub seat with its tolerance class, and has no general tolerance line (ISO 2768), for a later release: note 5 names each untoleranced seat by its diameter and where it runs, with the class of a fit case calculated for that diameter, to be specified before manufacture (user test ut4).

- A deflection, tilt or critical speed on rigid supports (bearings linked, "Couple linked bearings" off) leaves out the bearings' give; the results, checks and "Show the calculation" say so, and the results offer "Check with the bearings' stiffness". Coupling linked bearings by default is for a later release (user test ut4).
- The app opened at a copied or reloaded Steering address (`#/steering/<type>/<section>`) opens that page once the project's first results are in, even when another module was chosen meanwhile; leaving Steering before then should let the address go, for a later release (land-t review). An example that lands on Steering no longer does this.
- Compare variants writes a steering-duty check as "not computed" (or with its value) in every column by the first column's state: a design whose steering duty is computed beside one whose duty is not shows both alike, for a later release (each column's own message; review of RR1 branch 11). The status word in each column is that column's own.
- The shaft's slope at a load is a result to read, not a check: a limit on it (a hub's or a gear's tilt) is for a later release (RR1 decision 10). Over a duty cycle it is the loads as entered, and says so.
- A reduced housing matrix from a Nastran DMIG punch, an Abaqus matrix or a CSV states no count of its entries, so a file cut off between two entries cannot be told and reads with the missing ones as 0 (a stiffer housing); check that its last entry is there. Harwell–Boeing and Matrix Market files are held to their headers (see "From 0.3.0"). A reduced matrix is read up to 600 degrees of freedom: a Craig–Bampton reduction keeping more modes is refused (RR1 audit O-12).
- Housings: a seat a support, fit or rib sits in can't be moved by an edit, and putting a seat found on a bore onto that bore's own patch (**Seat it here**) counts as a move, so it is refused while something sits there; unlink it first, or seat it on the patch before linking (housing-032 review, O3).
- Housings, for a later release (user test RR1, gap round): a housing FE solve runs on the app's own thread, so the window stops answering for the 20–30 s of a large mesh's solve and Windows may say the app stopped responding (it finishes; wait). An imported mesh without a density (a STEP part, or a Nastran deck without RHO) is taken as steel, 7 850 kg/m³, beside the grey iron's E and ν the housing keeps, while the density's tooltip names 7 200. The app does not mesh an existing housing's STEP part again at another element size, and the size is not kept with the mesh.
- An eccentric bolted joint whose interface is larger than VDI 2230's limiting size G in the bending plane (R0, Eq. 54, 55: d_W + h_min for a through bolt, 2·d_W tapped) is not rated: the checks that rest on the eccentric model are not determined, with the reason and what to do (define the interface by a recess, or check the joint numerically or by test). A model of the wider interface (its non-linear pressure) is not planned; a bolt pattern's per-bolt interface is the user's to choose (RR1 U-A12, decision 2). For a tapped thread G is taken at the upper end of VDI 2230's 1.5 … 2·d_W, so an interface between the two reads as within it; and where a working moment presses V closed in both load states while the preload alone lifts it, the opening check fails on a pressure at V below 0 that does not show the assembly state's tension (Opus review of RR1 branch 13, P3s).

- Messages and refusals, for a later release (RR1 branch 22, what it left): - Validation errors from an element's own engine are placed at the field their message starts with, but about 75 of the project's validation errors still give no hint beyond their message ("must be positive"), and a JSON value of the wrong type in `axomek apply` is refused without its field's name (RR1 TA-F53, agent F5). - A power typed into a torque field is converted at the speed field's value at that moment, said only until it is applied; the torque then stays when the speed changes (RR1 U-A28). A key placed on a shaft keeps its own diameter and torque, and a keyed section can't be swept, optimised or proposed by `fix` (U-A27). "What matters most" can rate a screw's root diameter as not moving its speed check (U-A24, not traced). The planetary refusals and fixes U-C3 lists (a nearby set the search finds but the card does not, a ring shift offered before a planet change, the default planet count beside a typed ring), the tilt note shown above a failing minimum load, a warning's reason shown only on the Overview, "Driven by" offering stages that close a loop, and a single tapered roller bearing's raw "could not reach equilibrium" in the bearing finder. - The Assistant's results give a life in whole hours, as the app shows it, so a life just under its minimum (19 999.6 h against 20 000 h) reads as the minimum there; its check, failing, keeps the exact value. Where one case of a duty cycle puts a bearing outside its rating method (κ below 0.1, balls in four-point contact), the Assistant is told to quote none of its lives, a hybrid's basic life included, not even the modified life at the loads as entered where that one is within its method (the safe side). The room the app makes to keep a field in place is kept through a mouse click on a button in the panel, not yet through a tap on a touch screen (Opus review of RR1 branch 22). - A support's own Designation is shown as its linked bearing's and can't be typed over while a bearing is linked; the value saved in the file stays and comes back when the bearing is unlinked.
- Shoulder heights and the assembly order are not checked: a shaft whose seats have no shoulder for a bearing or a hub to sit against, or with a part that cannot slide on from either end, can pass every check, and Make it pass, the design search and the Assistant's designs can lead to one (user test RR1: the Assistant's lightest shafts, "All 16 checks pass"). The shaft help says so, and the Assistant is told to keep shoulders and a mountable layout. A shoulder (abutment) check and a mountable-from-an-end check are for a later release (RR1 U-A34, decision 4).
- US customary units change the display and the entry only: the bearings are metric (no inch-bore bearings, adapter sleeves, pillow blocks or insert bearings in the presets or the bearing finder), the steels go by their EN names, the keys by DIN 6885-1, and the shaft drawing is metric on an ISO sheet, which its dialog says with US units shown (its fits table gives inches beside the mm). The help (Units, Bearings) says so. Inch drawings, ANSI sheets and US parts are not in 0.3.0 (RR1 decision 5).
- A spring's relaxation (its loss of force over time, warm or near its static limit; EN 13906-1, 10.2) is not computed or checked, and its operating temperature is not an input: the materials are at room temperature. The springs help says so; for a later release (RR1 D, user tests springs and mechanism).
- The Contact module's materials have no hardness: the default permissible pressure (ISO 76's 4 200 MPa for a point contact, 4 000 MPa for a line contact) takes both bodies as hardened to bearing hardness, and a passing pressure check does not say so; the help does. A softer body needs **Custom E and ν** with its own permissible pressure. Stating the assumption on the check is for a later release (RR1 U-A37).
- In the desktop app, a file opened while another opening still waits can get that first opening's "Opened <file>" line in its Activity list; nothing else of the first opening reaches it (Codex review of RR1 branch 18, for a later release).

- In the web app two tabs can write the same file: a draft reopened from Recent projects keeps the name of the file it was last saved to, and its Save offers that file while another tab has it open. Only the browser's "replace it?" prompt guards it; the web app does not hold a file for one tab, as the desktop app does for one window (Opus review of RR1 branch 19, for a later release).
- Web app save notes (savedialog-036 review, for a later release). Save in "Save changes?" and Save to file in the "only in this browser" notes (the storage warning's too) say nothing when the save dialog closes or the browser does not show it (the project stays open, its changes unsaved, and nothing is downloaded). The app cannot see a download arrive, so a download (Download a copy, Download instead, or Save where a browser has no save dialog) counts as saving the project only while this browser keeps its draft too, and its note says "Download started"; Your projects, Home, the Overview and the header's status still say "file downloaded" for a download the browser was only asked for. With storage full, a download of a project whose draft is kept already (room for the project, not for the download's note kept beside it) counts it as not kept, and leaving asks once more than it needs to. A save dialog left open after Download instead can clear a newer save's note when it closes, and the save and download notes appear with their text, which some screen readers do not read out. Check the download folder, or use Save or Download a copy again. Home's "Download started" names the project as listed (a legacy cloud file `Pump.axm.json`), not the name the browser saves it under (`Pump.axomek`). After a save conflict, an Undo to your version, a cancelled download and a Redo, your edits are only in Undo: the download note still says the project is kept in this browser without the Undo warning (leaving still asks), and Delete in Your projects is refused saying storage is full, though freeing room does not help; Undo to your edits and save or download them first.
- `axomek cloud pull` without `--force` refuses a file that is already there before it downloads, not when it writes: a file another program creates at that path during the download (a few seconds) is overwritten. A write that never replaces a file is for a later release (RR1 audit TA-F33).
- Cloud project names, for a later release (release-candidate review): - the account service refuses a name whose part before its extension is longer than 120 UTF-16 code units (61 emoji are 122), though the app and `axomek` cut a name at 111 characters, not at 120 units, and a name an earlier version saved that starts with an invisible U+FEFF: such a project can't be saved, copied, renamed or moved to the trash, nor its versions listed, opened or restored; save it under a shorter plain name; - a name in another letter case or spacing is suggested only where it matches a name this version gives; an earlier version's name that differs from it more is not (one from before the rename to Axomek, or one an earlier version kept an extension in): `axomek cloud pull pump`, with only such an old Pump listed, suggests nothing; give the name exactly as `axomek cloud list` shows it; - `axomek cloud versions` and `restore --version`, given a name that is not listed exactly (an earlier version's, or one in another letter case or spacing), look up the name this version gives it rather than suggest the listed one; give the name exactly as listed; - `axomek`'s refusal of a name says its fix twice, in the message and as its hint; - a name with control characters is quoted with Rust's escapes (`"\u{1}"`), which can't be pasted back as the name.
- A dry plastic worm pair with no friction given fails a minimum back efficiency it misses at VDI 2736-2 Table 2's 0.20, though a lower friction (the table gives no scatter) could pass it: a pass there is "not decided", a fail stays a fail, on the safe side. Give the friction measured on the pair (review of small-034, for a later release).

- The number format's Automatic (Settings › General) guesses from the browser's way of writing numbers and its time zone: a browser that writes a point (an English Chrome on a Windows set to German numbers, which writes 1234.5 as "1,234.5" where Edge writes "1.234,5": checked 2026-11-24, and the Windows nightly checks both) in a time zone of a country that writes a decimal comma refuses both "1,200" and "1.200". A computer set to a time zone that is not in the short list (`DECIMAL_COMMA_ZONES`, apps/desktop core/number-style.ts), such as one of another country than its owner's, still reads "1.200" as 1.2 there; choose "1.234,5 (comma)" in Settings, or write numbers without a thousands separator (1200), which read the same either way.
- The number format chosen in Settings › General is kept apart from the other settings: Reset to defaults leaves it as it is (choose Automatic to go back), and a Settings window already open in another tab shows the choice it had when it opened (the app uses the newer one the next time it is opened). A number typed in the desktop app while Settings is open and the computer's format is still being read is dropped with a message that can sit behind Settings; close Settings to see it. On a screen narrower than about 330 px the end of "1.234,5 (comma)" is cut off until Settings is scrolled sideways (landing review of numfmt-032, for a later release). With the comma format, an English "1,500" pasted into a field is 1.5, shown as "= 1.5" under it before it applies, not refused. The bearing duty cycle table's cells (speed, torque, temperature, share, factors) put their value back without saying why when a number reads two ways there, as for any text they do not read; the number fields say what to write (numfmt-032, for a later release).
- Development, `node scripts/ci.mjs --parallel` (accepted limit): it installs the pinned Rust once before the lanes start, so that they do not race to install it. Two local rustup settings defeat that: a `rustup override` on the checkout folder (the check before the lanes uses the override, and the lanes can still race to install the pin), and `auto-self-update check-only` with rustup's update server out of reach (the run stops before any lane). Neither is set on the CI runners or the build VMs. Remove the override, or run `rustup toolchain install` once first; with the update server out of reach, switch rustup's automatic self-update off for the run (`rustup set auto-self-update disable`, and `check-only` again after). Either way a run can fail, never pass wrongly.
- A QJ 207 linked to no support whose loads are entered by hand as a starter's (Fa 1500 N, Fy 2000 N, the earlier one's, or Fa 3000 N, Fy 1000 N, today's; the rest 0) is warned about as rated on loads nobody entered; the warning is advice only. The test that every example and Add choice solves determined (R-956) solves Add Planetary on a blank project, not as a stage driven by another; does not fail on a bearing row left not set up (no example has one); and checks that each Add choice the app sends is covered, not that a covered choice's patch matches the app's.
- Development, the tests' own thread limit (R-970): the guard that every test asks for its threads through `own_threads()` reads `use` lines one by one, so a grouped import (`parallel::{automatic, with_threads}`) slips past it; the unit test of the limit sets it directly rather than through `AXM_THREADS`; and in a parallel `cargo test` two tests that mute the panic hook can restore each other's, so a later test's panic may print no message. Under nextest (each test its own process) the last cannot happen; the first two can only let a test use more threads. The engine's own-thread helper starts one worker (with its 64 MiB stack) even for an empty list, so a computer that refuses that thread fails an empty call instead of returning nothing.
- Development, the console's shared assets script (`apps/admin/scripts/shared-assets.mjs`): its test of a refused argument checks that `chart.css` is not written, not that nothing else is (the script refuses its arguments before it touches any file).

- The public site's What's new (sitedates-033): a release without a `release <version>` commit is dated from its v`<version>` tag once the site offers that version or a later one. A version tagged but never published (its release left a draft) and followed by a later published one would read "Released" on its tag's day. When a release is abandoned, delete its tag, or fold its README notes into the next version's. A lightweight v`<version>` tag gives its commit's date, so a version tagged a day after its release commit reads "Released" on the commit's day. While the download feed is rolled back to an earlier version, a site deploy from main shows the newer version's notes without their release date until the feed offers it again. The workflow check finds the site build's checkout of this repository by its `repository:` line, not its YAML value: a folded value (`repository: >-`) or a comment on that line is read wrongly (no workflow writes either).