

Kinematics report

Sample document with complete example values. It shows the report the purchased export produces for YOUR input - it is not itself a report for any part.

Project: Sample project

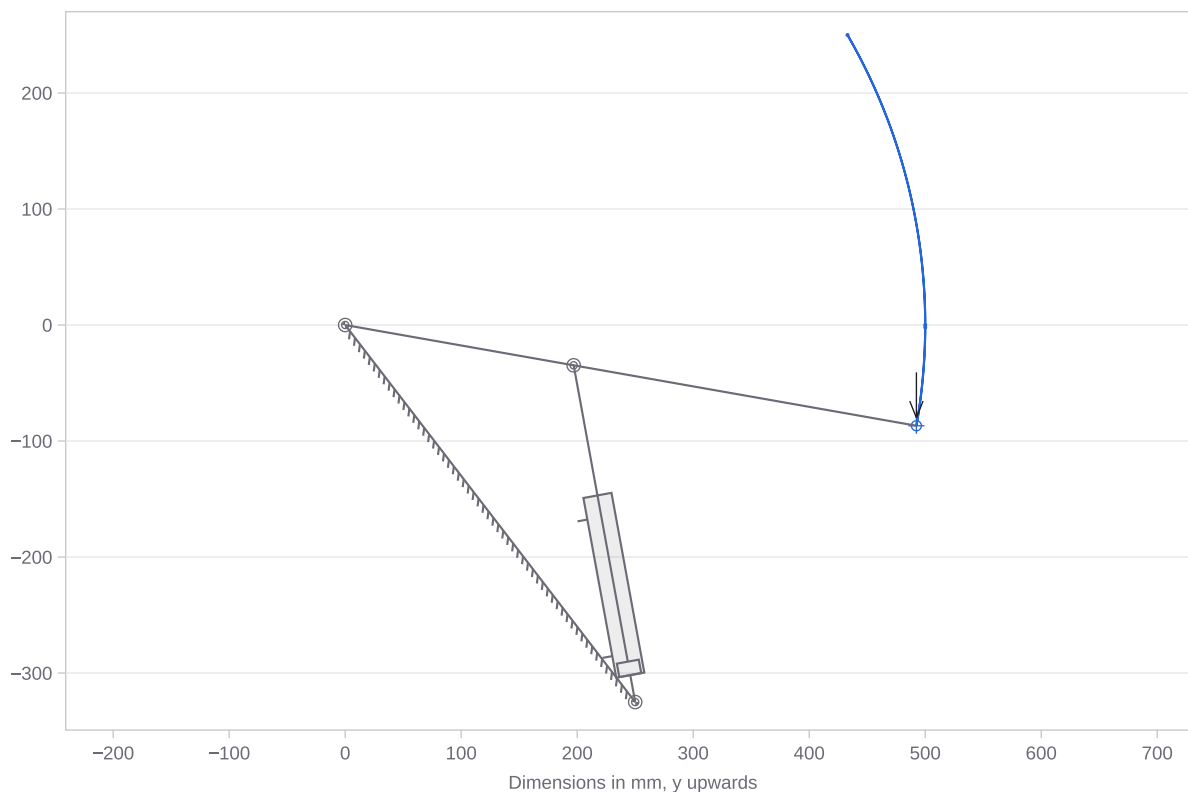
Part/Position: no real part

Prepared by: Sample, not released

Overall assessment: OK — every checked condition is met. Only the rows of the verification table were assessed; everything else is stated as a characteristic value without rating.

This document certifies the MOTION of the linkage described below: its structure, its traverse, the angles, torques and joint forces computed from it, and the accuracy of that computation. It expressly certifies NO strength, no service life, no natural frequency and no manufacturing tolerance. Of freedom from collision it certifies exactly what was checked: the position of the link centre lines relative to the drawn keep-out areas — not the links against each other, not their actual width, and nothing where no keep-out area is given.

System sketch in the initial position



Drawn is the same scene as on screen: frame, links and joints in the neutral tone.

The path of the marked point is drawn in the curve colour; its crosshair and its reversal marks carry the same tone.

Computation

Steps per traverse	360
Positions computed	361
Duration of one traverse	6.283 s
Gravity (positive acting downwards)	9.81 m/s ²
Position shown	0
Superimposed positions in the system sketch	0
Permissible drive force $F_{A,perm}$	4,000 N

Links

Link	x_o [mm]	y_o [mm]	φ_o [°]	Link line [°]	m [kg]	s_S [mm]	J_S [kg·mm ²]
gestell	0	0	0	—	—	at the reference point	—
ausleger	0	0	−10	—	—	at the reference point	—

Joints

Joint	Type	Links	Attachment 1 [mm]	Attachment 2 [mm]	Extra
fest	Ground constraint	gestell	0 0	—	$\varphi = 0^\circ$
A0	Revolute joint	gestell – ausleger	0 0	0 0	without friction
Z	Cylinder	gestell – ausleger	250 −325	200 0	retracted length $L_0 = 295.08$ mm; stroke 136.8 mm

Linear drives

Drive	Cylinder or prismatic joint	from s [mm]	to s [mm]	Segments	Transitions
1	Z (Cylinder)	0	136.8	6	free of shock and jerk

Motion laws per segment

Drive	Segment	to τ	to fraction	Law	At the end
1	1	0.05	0.0556	Quintic polynomial through the support points	Pass through
1	2	0.45	0.9444	Quintic polynomial through the support points	Pass through
1	3	0.5	1	Quintic polynomial through the support points	Stop
1	4	0.55	0.9444	Quintic polynomial through the support points	Pass through
1	5	0.95	0.0556	Quintic polynomial through the support points	Pass through
1	6	1	0	Quintic polynomial through the support points	Stop

External loads

Load	Kind	Link	Point of application [mm]	Direction [°]	Magnitude
last1 (Nutzlast)	Force at a point	ausleger	500 0	−90	1,000 N

Marked link points

Point	Link	x [mm]	y [mm]	x on the link [mm]	y on the link [mm]
punkt1	ausleger	492.404	−86.824	500	0

Structure and degree of freedom

Links n (including the frame)	4
Ground constraints	1
Simple joints g_1 (revolute and prismatic)	4
Higher joints g_2 (cam joints, gear pairs, rack-and-pinion pairs, Geneva drives)	0
Drives	1
Degree of freedom $F = 3 \cdot (n - 1) - 2 \cdot g_1 - g_2$ (Grübler)	1

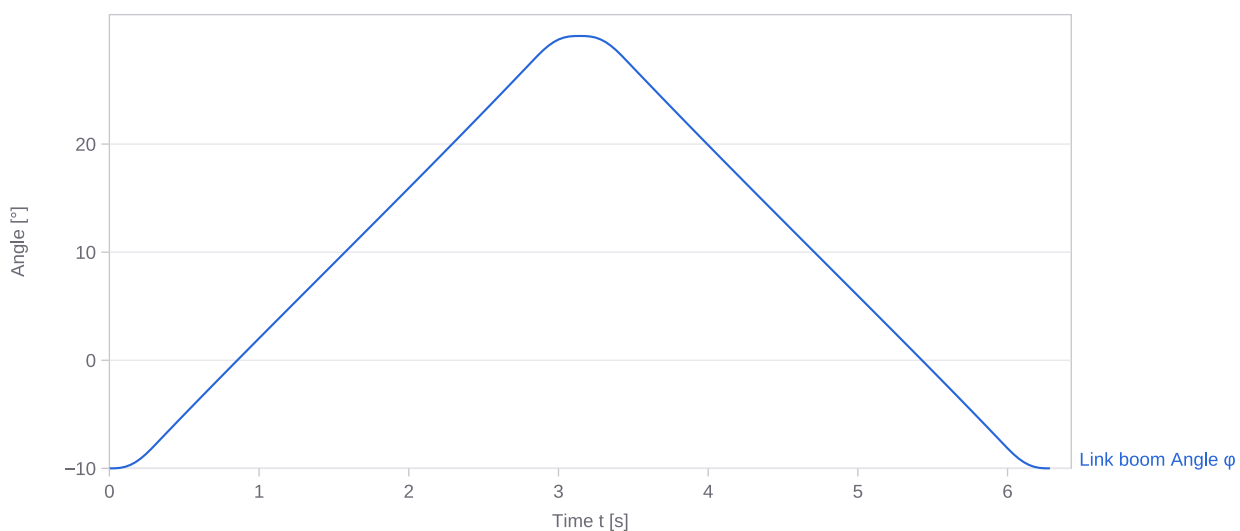
Verifications

Criterion	Value	Limit	Status
Degree of freedom F , required 1 (number of drives)	1	1	● OK
Drive range complete and without gaps			● OK
Transitions free of shock and jerk			● jerk-free
Largest drive force $F_{A,max} \leq 4,000$ N	2,626 N	4,000 N	● OK

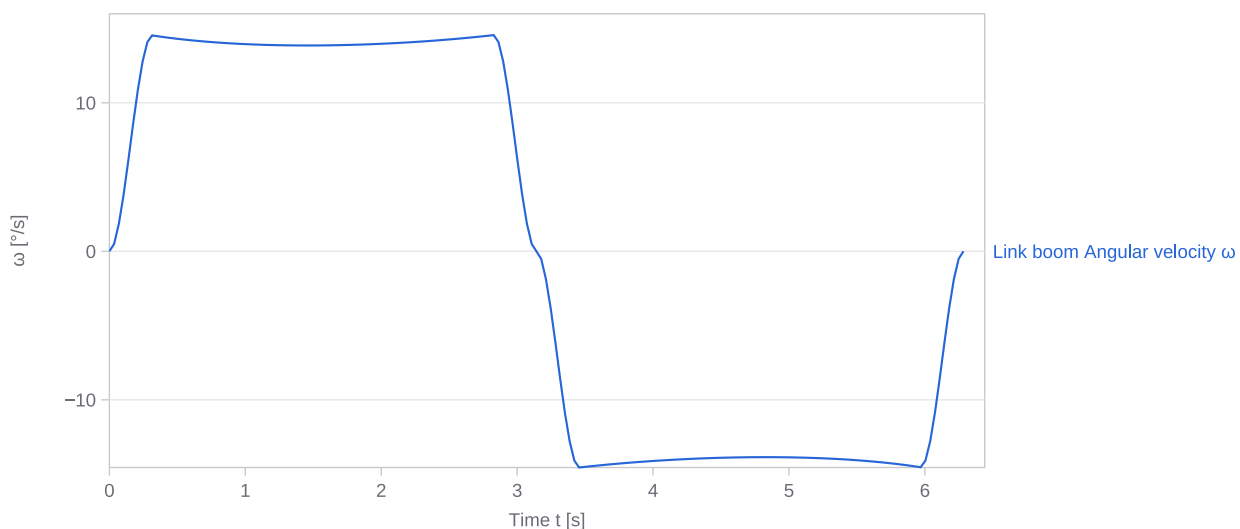
Characteristic values

Smallest transmission angle $\mu_{\min}$	not defined here
Transmission angle $\mu_{\min}$ at cylinder Z	69.6°
Largest deviation from the target path	no target path drawn
Largest penetration into a blocked area	no blocked area drawn
Smallest locking factor $r_{\min}$ (self-locking)	no μ_0 entered
Largest joint force $F_{\max}$	1,652 N
Largest drive power $P_{\max}$	125.7 W
Largest shaking force $F_{R,\max}$	no load in the model
Type according to Grashof	not a four-bar
Shortest link	not a four-bar
Grashof excess $(s + l) - (p + q)$	not a four-bar
Smallest quality index	0.710

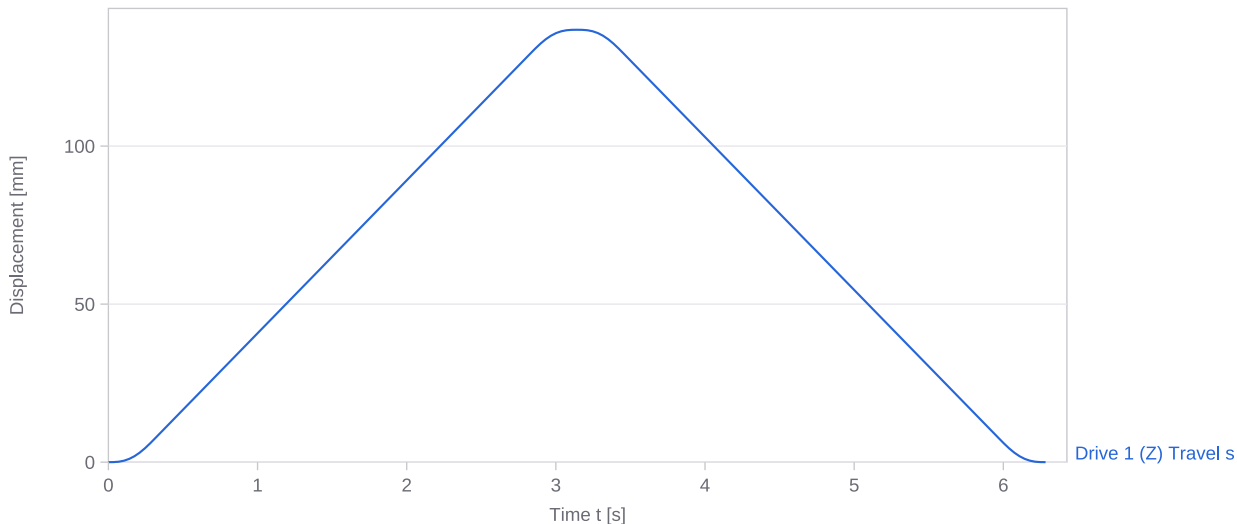
Angle over time [°]



Angular velocity over time [°/s]



Displacement over time [mm]



Basis of calculation and model assumptions

The closed constraint equations of the planar linkage are solved: for every progress τ on the common timeline the position of all links is found by Newton's method, after which velocity and acceleration are obtained from the same Jacobian. Drive torque and joint forces follow from d'Alembert's principle (kinetostatics) with the masses and inertias given. The motion laws of the segments are the normalised time laws of VDI 2143.

HOW CYLINDER AND LINEAR DRIVE ARE COMPUTED: A cylinder connects two links through one pivot each, the barrel at the first, the piston rod at the second. On its own it constrains nothing — it is a telescope, and only its drive gives it an equation. In the Grübler count it therefore counts as two links, barrel and rod, with two revolute joints and one prismatic joint between them; it contributes $3 \cdot 2 - 2 \cdot 3 = 0$ to the degree of freedom. The piston travel s is the distance between the two pivots minus the retracted length L_0 , that is, minus the distance with the piston retracted. A linear drive prescribes this travel over time — on a cylinder the piston travel, on a prismatic joint the travel of the slider along its guide —, with the same timeline and the same segments as a rotary drive, only with a travel in mm instead of an angle: $s(\tau) = s_{\text{from}} + (s_{\text{to}} - s_{\text{from}}) \cdot g(\tau)$. The force in the drive is the Lagrange multiplier of this equation, in N and positive when pushing; it appears as the largest drive force $F_{A,\text{max}}$ among the characteristic values and is rated against a limit where one is entered. Barrel and rod carry neither mass nor friction; whoever wants to include them assigns them to the attached links. If the piston travel leaves the range from zero to the stroke anywhere, a warning appears in the document.

WHAT APPLIES BETWEEN THE SUPPORT POINTS: Between two stops the motion law of the segment applies. Where one end is a pass-through or reversal point, a quintic polynomial joins travel, velocity and acceleration of the two support points instead, and the segment table names it so. At a pass-through point the drive continues at the velocity that follows from the neighbouring segments; at a reversal point it stands still for an instant and turns back. Travel, velocity and acceleration are continuous at every support point.

WHERE THE MASS ACTS: the mass of a link acts at its centre of mass s_S , which the table of links below states for every link. When a mass is entered on screen while the centre-of-mass fields are empty, the tool prefills them visibly — with the midpoint between the two revolute joints of the link, or with the mean of them where there are more than two; a prismatic joint does not count, because its attachment location along the guide is arbitrary. The suggestion can be overwritten, a centre of mass already set is left alone, and a loaded state is not touched. Where the table reads “at the reference point”, none was given, and it coincides with the build location of the link — a joint for everything this tool creates. The entered mass is then not a mass distributed over the link but a point mass in that joint. The consequence has been measured: on the four-bar linkage 10 kg on the crank thus have no effect at all on the drive torque, and on the coupler only through their weight. The moment of inertia J_S refers to the centre of mass; the parallel-axis term $m \cdot s_S^2$ is added by the computation itself.

WHEN MASSES HAVE NO EFFECT AT THE DRIVE: a link running at constant speed about a fixed axis needs no drive torque — no matter where its centre of mass sits or how large its inertia is: its centrifugal force points through the axis and has no lever arm there, and the link is not accelerated. The prefilled centre of mass changes nothing about this, it merely moves where the centrifugal force acts. Mass takes effect through gravity, through a motion law on the drive, or on a link that does not run at constant speed. Where none of this applies, all that is left of drive torque and drive power are the last bits of the computation. Both are therefore compared against the load this model can carry at all and are dropped once they lie nine orders of magnitude

below it; the row then states the reason and NO limit — a figure of that magnitude would fall below any limit and thereby pass a check that never took place. The joint force is not affected and is still reported: the centrifugal force of the crank is a real bearing force, it merely lacks a lever arm about the drive axis.

HOW AN APPLIED FORCE IS COMPUTED: spring, damper, constant force and external load all enter the computation by the same route. Every force element has a scalar measure g — for a translational element the distance between its two attachment points in mm, for a rotational element the relative angle of the two links in rad — and from it follows its scalar force $S = F_0 + k \cdot (g - g_0) + c \cdot \dot{g}$ with the rest value g_0 , the stiffness k and the damping c . A positive S pulls the two attachment points together, or reduces the relative angle. The translational element measures F_0 in N, k in N/mm and c in N·s/mm and yields a force; the rotational element measures F_0 in N·mm, k in N·mm/rad and c in N·mm·s/rad and yields a torque directly. The rest value g_0 is the value at which the spring force is zero; it is taken from the distance as drawn when the element is placed and stays put afterwards — it expressly does NOT follow along when a joint is moved, because that very difference is where the force comes from. An external load has neither a measure nor a characteristic: its magnitude is exactly as entered. The attachment and application points of both tables are given, like those of the joints, in the moving frame of their link, whereas the line of action of a force load is given in the frame. The difference between the two kinds can be checked and is not a convention: a force element acts between two links and is in equilibrium within itself, whereas a load reacts against the surroundings — its counter force goes into the ground constraint.

Applied forces, not a dynamic simulation: the drive prescribes the motion and the forces follow from it. Spring, damper, constant force and external load therefore change the drive torque and the joint forces — they do not change the path, the velocities or the accelerations. Here no spring accelerates the mechanism and no load slows it down; if you release the gas spring of a flap and want to know how the flap then falls, this calculation does not give you that answer. On the real machine the separation holds exactly as long as the drive is stiff enough to enforce its motion against these forces.

WHAT IS NOT COMPUTED ABOUT THE APPLIED FORCES: the characteristic is linear. A gas spring with its friction hysteresis, a progressive spring and an end stop with cushioning are therefore covered only approximately — F_0 serves as an approximation for the nearly horizontal characteristic of a gas spring, while the friction force on extending and retracting appears in no figure. The line of action of a force load is fixed in the world and does not turn with the link; a cutting force that turns with the tool is not computed by this solver. The magnitude of a load is present over the whole traverse as long as no course is entered; the table "Load course over the cycle" appears in this document exactly where one is entered. The measure of a rotational element is the relative angle and is not folded onto one turn: on a fully rotating crank it grows without bound, and with a stiffness entered the spring force grows with it; the document carries a note about that. Without a stiffness — a pure damper, a constant force — the force measure does not depend on the measure at all, and then there is nothing to report. Friction in the joints is computed as soon as a coefficient of friction is entered at a joint; the joint table states it per joint, and where none is given, that joint is computed without friction.

HOW THE CHARACTERISTIC VALUES ARE OBTAINED: the transmission angle μ is defined for two configurations — for the four-bar chain as the angle between coupler and rocker at joint B, for the slider-crank between coupler and the normal to the guide line (Kerle/Pittschellis/Corves 2007, p. 161, eq. 6.5 and 6.6). The acute value is reported, and as $\mu_{\min}$ the smallest across all computed positions; the closed form given in the source holds for a full revolution and is therefore not applied but only used as a cross-check. For any other chain the row states that the quantity is not defined there. At a cam joint it appears as the pressure angle $\alpha = 90^\circ - \mu$ in a row of its own. If a cylinder sits on the chain, the chain is taken without it, and a row of its own reports the transmission angle at the cylinder: $\mu = 90^\circ - \alpha$ between the cylinder axis and the motion of its rod-end point relative to the base. As a guide value the same source gives $40^\circ \leq \mu \leq 50^\circ$ for slow-running mechanisms (p. 160, eq. 6.4); that is experience and not a standard — VDI 2130 devotes a section to the transmission angle and names no figure. The row therefore carries the wording of a guide value and not that of a passed verification: at 39° nothing breaks, but bearing forces, friction and sensitivity to backlash grow quickly. The output link is taken to be the non-driven link attached to the frame by a revolute joint; its reversal positions are the zero crossings of its angular velocity, interpolated linearly between two grid points. If it stands still in a dwell and moves back afterwards, the dwell is one reversal; over a closed cycle a dwell across the start of the cycle counts as well. If a reversal lies in a dwell, the time ratio is omitted, because which side the dwell time belongs to would be a convention. If it reverses exactly twice, the swing angle ψ_0 follows as its full excursion and the time ratio as the longer of the two intervals divided by the shorter, hence always ≥ 1 . The time ratio refers to TIME and not to crank angle: with a uniformly running drive the two are the same; if a segment carries a motion law they part company, and time is what counts. It is only stated for a drive that completes at least one full revolution — otherwise there is no return stroke to measure against. If the output link instead slides in a guide on the frame — the piston of the slider-crank, the follower of a cam mechanism —, its velocity along the guide counts, and the stroke $s_0 = s_{\max} - s_{\min}$ in millimetres takes the place of the swing angle, as fine as the grid. The same holds if the link at the frame rotates through a closed cycle without a reversal and a ram slides behind it, as in the Whitworth quick return. Where no unique output link can be found, the rows are absent: that applies to any mechanism with more than one drive, such as the five-bar — which of

its links would be the output is not something the structure states. The swing angle is measured across the computed positions and is therefore as fine as the grid; at 360 steps it sits about 0.002° below the closed-form value. The drive power is, in every position, the sum over the drives of torque times angular velocity, and $P_{\max}$ its largest magnitude across the traverse — not $M_{\max}$ times $\omega_{\max}$, because with a motion law the two peaks do not coincide.

TYPE ACCORDING TO GRASHOF: whether a link of the four-bar chain of revolute joints rotates fully is decided by the four link lengths alone. With s the shortest, l the longest and p, q the remaining two, the condition reads $s + l < p + q$; the Grashof excess stated is the difference $(s + l) - (p + q)$ and is therefore negative wherever the condition holds. If it holds, the type follows from where the shortest link sits: at the frame a crank-rocker, as the frame itself a double crank, as the coupler a double rocker. For equality the mechanism is a change-point linkage and hence not run-safe — it passes through the aligned position, but which branch it takes afterwards is not determined. If the condition is violated, all four joints merely oscillate (triple rocker). A prismatic joint or a fifth link makes it a different chain, for which there is no Grashof type; the row then says exactly that. The type is a statement about running behaviour and not a grade — it is not assessed.

WHAT THE DRIVE ANGLE REFERS TO: the columns “from φ ” and “to φ ” of the drive table give the absolute angle φ of the driven link against the horizontal, measured about its reference point and anticlockwise. It is the same QUANTITY the link table lists as φ_0 , but not necessarily the same NUMBER: the link table gives the STORED design, that is the position in which the linkage was built, and for the solver that is merely a starting value. At $\tau = 0$ the solver imposes “from φ ” on the driven link and solves the chain from there; where φ_0 and “from φ ” differ, the drawn initial position therefore stands beside the figures of the link table by that difference, and “from φ ” governs everything computed. For a linkage computed from prescribed positions this difference is the rule and not an inaccuracy: the design is stored in the prescribed position at which the drive range begins — as a rule the first one — and the range starts one percent of the position span BEFORE it, but at most one percent of the distance to the limit position beyond which the chain no longer closes. The prescribed position thus lies safely inside the drive range and not on its edge, where rounding alone would decide whether it still belongs to it; the check table gives it as a progress τ just above zero. The drive angle is expressly NOT the direction of the drawn link line between two joints: between the two lies the constant angle that the head point has in the moving frame of the link itself. The column “Offset to link line” gives that angle per link line wherever it is not zero, and a dash stands where there is nothing to give. An offset of 180° , for instance, means that in the drawing the link line points exactly opposite to the drive angle, although both describe the same link.

The quality index states how far the linkage stays clear of a dead-centre position, not how load-bearing it is. It is, across the traverse, the smallest value of the geometric mean of the singular values of the non-dimensionalised Jacobian — known in robotics as Yoshikawa's manipulability. It is 0 in a dead-centre position and approaches 1 the more perpendicular the columns are. Where it approaches zero the linkage passes through a dead centre — joint forces and angular accelerations are unreliable there. Before that, every position is iterated until the chain closes: until the largest remaining gap of the constraint equations falls below 10^{-10} , at most 40 Newton steps. A position that does not get there is not part of this document; it is reported as a gap in the drive range — the row “Drive range complete and without gaps” says so.

Model assumptions: all links are rigid, all joints free of backlash. The computation is without backlash and without elasticity; positions follow first-order theory. Friction is computed only at those joints for which the joint table states a coefficient of friction. Only drawn keep-out areas are checked, and only where some are given; the links are NOT checked against each other. Two links may therefore pass through one another in the calculation without anything being reported here.

The drive torque and the largest joint force stated are INPUT QUANTITIES for a strength verification, not a substitute for one. For a shaft under this torque the verification to DIN 743 applies, for a plain bearing the load rating, for a pin the surface pressure and shear verification.

About this sample

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