

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

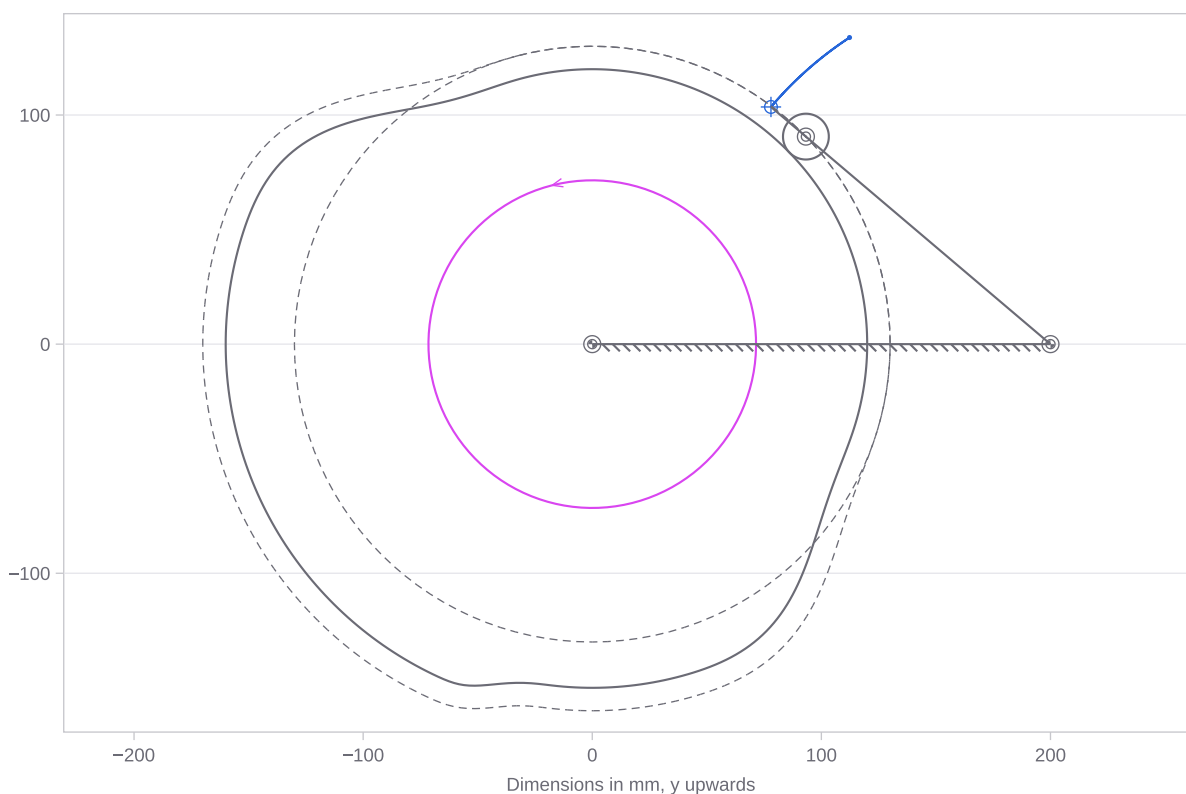
Part/Position: no real part

Prepared by: Sample, not released

Overall assessment: OK with notes — every checked condition is met, at least one of them with little margin. The row with the amber dot says which.

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.

A cam joint carries four lines: the working profile of its disc as a closed, solid curve, the pitch curve broken and half as wide beside it, the base circle r_0 about the pivot of the disc, broken as well — it coincides with the pitch curve wherever the travel is zero —, and the roller as a circle touching the profile. The pitch curve is the constraint itself — the roller centre may travel along it and nowhere else — and the ring at the centre of the roller is its joint. The point of contact carries no symbol of its own; the contact itself is the statement.

The drive stands out by the circle around its bearing. The arrowheads on it show which way it turns — one for a full revolution, two for a drive that swings between two limits. The horizontal line from the pivot to the circle is the zero direction the arc is read against: the drive angles count from it, anticlockwise, and not from the link line that starts there.

Computation

Steps per traverse	360
Positions computed	361
Duration of one traverse	6.283 s
Mean speed of the drive	9.549 1/min
Gravity (positive acting downwards)	9.81 m/s ²
Position shown	0
Superimposed positions in the system sketch	0

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	0	—	at the reference point	—
scheibe	0	0	-255.8	—	—	at the reference point	—
hebel	200	0	139.7	—	—	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 – scheibe	0 0	0 0	without friction
B0	Revolute joint	gestell – hebel	200 0	0 0	without friction
gelenk1	Cam joint	scheibe – hebel	0 0	140 0	$r_o = 130$ mm; roller radius $r_R = 10$ mm; 6 segments

Drives

Drive	Link	from φ [°]	to φ [°]	Offset to link line [°]	Segments	Transitions
1	scheibe	-255.8	104.2	—	linear across the whole range	free of shock and jerk

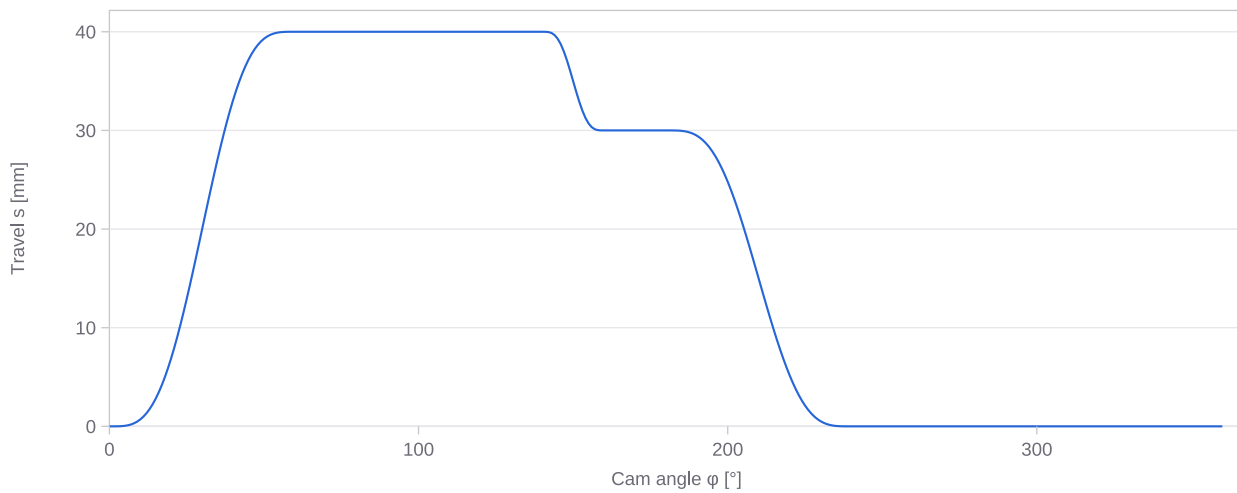
Motion plan of the cam discs

Joint	Segment	Motion type	Cam angle from [°]	Cam angle to [°]	Lift [mm]
gelenk1	1	Rise	0	60	40
gelenk1	2	Dwell	60	140	0
gelenk1	3	Return	140	160	10
gelenk1	4	Dwell	160	180	0
gelenk1	5	Return	180	240	30
gelenk1	6	Dwell	240	360	0

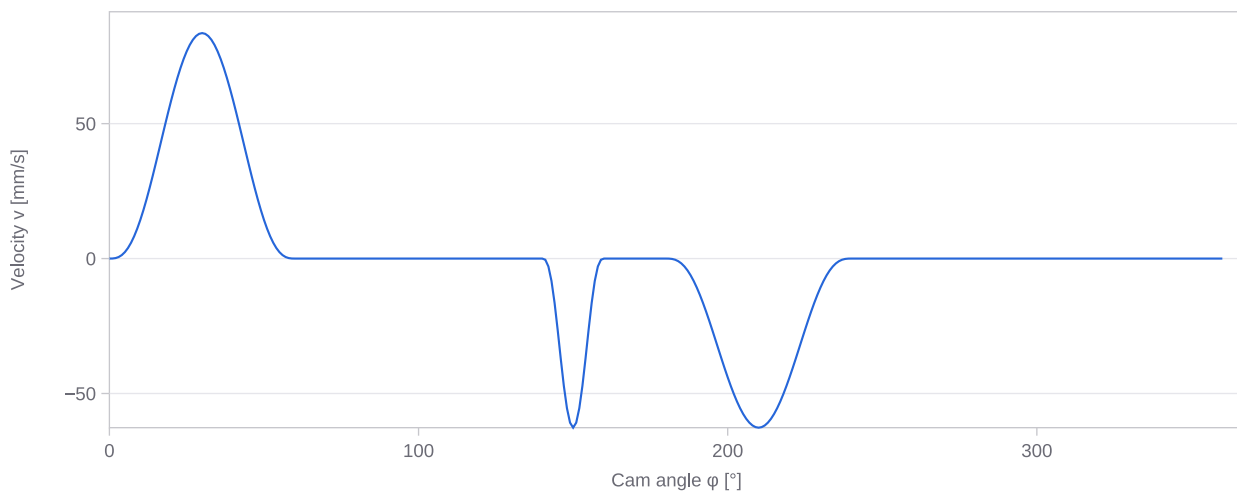
Check limits of the cam discs

Joint	Curvature safety $\rho_{\min}/r_R \geq$	Cutter radius r_F [mm]	Pressure angle limit α_{allow} [°]
gelenk1	1.43	5	30

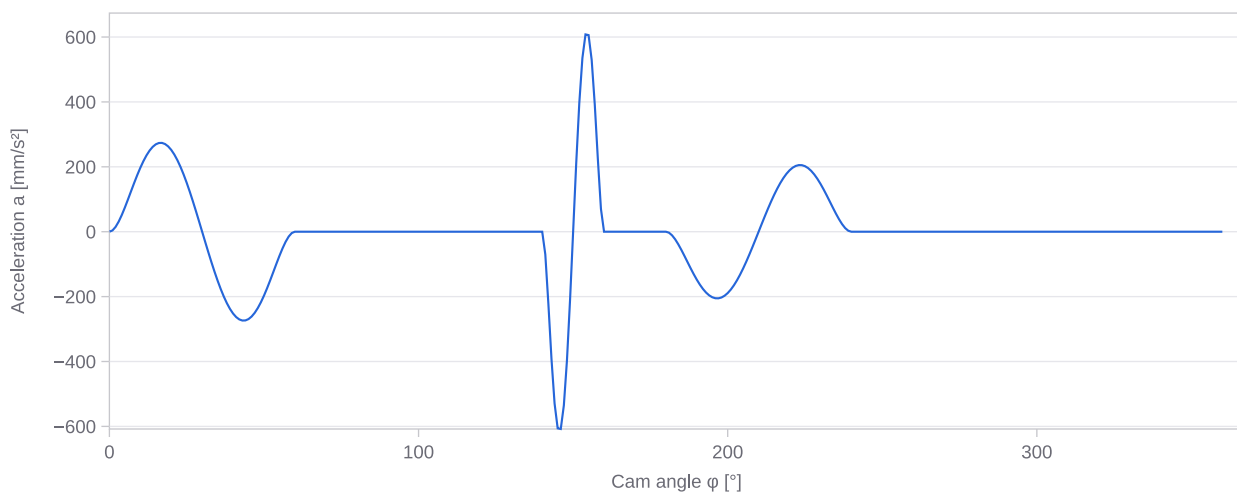
Travel s over cam angle: gelenk1



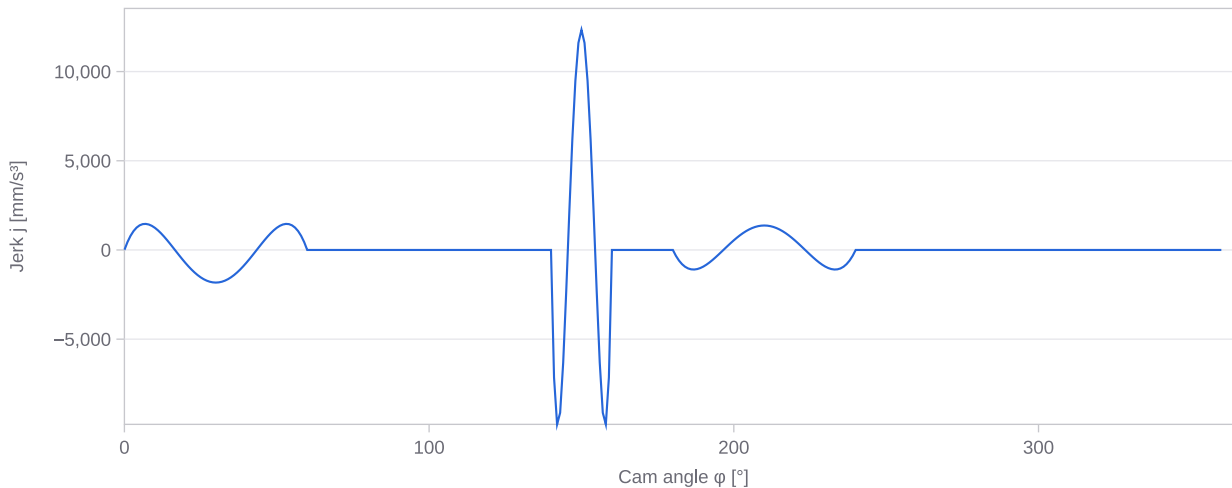
Velocity v over cam angle: gelenk1



Acceleration a over cam angle: gelenk1



Jerk j over cam angle: gelenk1



Marked link points

Point	Link	x [mm]	y [mm]	x on the link [mm]	y on the link [mm]
punkt1	hebel	78	103.518	160	0

Structure and degree of freedom

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

Verifications

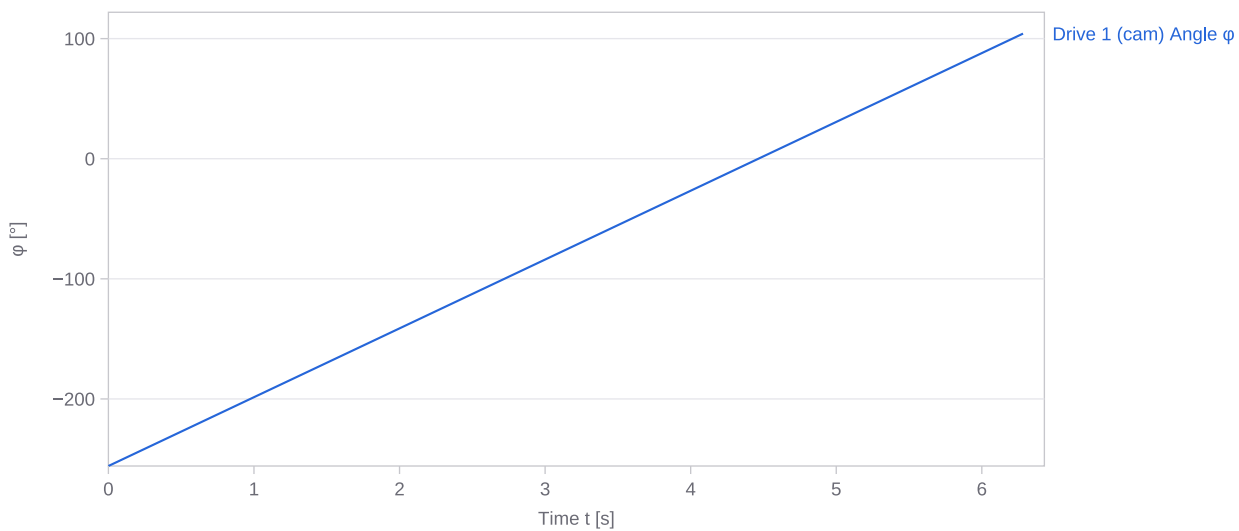
Criterion	Value	Limit	Status
Degree of freedom F, required 1 (number of drives)	1	1	● OK
Drive range complete and without gaps			● OK
Transitions free of shock and jerk			● jerk-free
Pressure angle $\alpha_{\max}$ at gelenk1 $\leq 30.0^\circ$	29.3 °	30.0 °	● just within guideline
$\rho_{\min}$ of the pitch curve at gelenk1 > 10.00 mm	37.70 mm	10.00 mm	● OK
Curvature safety $\rho_{\min}/r_R$ at gelenk1 ≥ 1.43	3.77	1.43	● OK
$\rho_{\min}$ of the working contour at gelenk1 ≥ 14.30 mm	27.70 mm	14.30 mm	● OK
Smallest concave radius of the contour at gelenk1 $\geq$ cutter radius 5.00 mm	71.36 mm	5.00 mm	● OK

Characteristic values

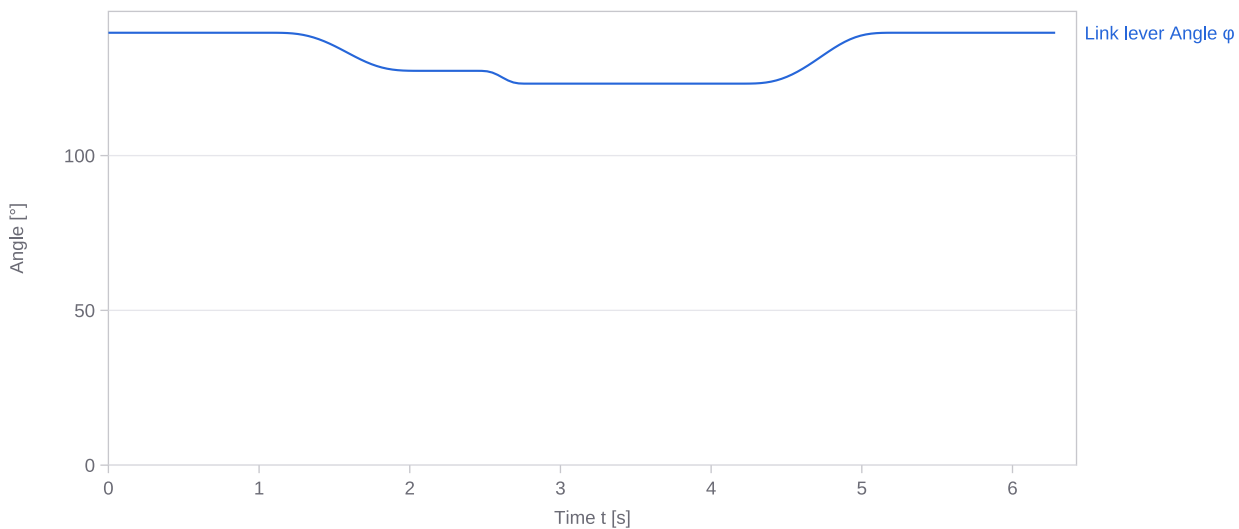
Smallest transmission angle $\mu_{\min}$	not defined here
Smallest contact force $N_{\min}$ at gelenk1	no load in the model
Largest stroke velocity $v_{\max}$ at gelenk1	83.56 mm/s
Largest stroke acceleration $a_{\max}$ at gelenk1	616.6 mm/s²
RMS jerk j_{rms} at gelenk1	1,947 mm/s³
Largest deviation 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 drive torque $M_{\max}$	no load in the model
Largest joint force $F_{\max}$	no load in the model
Largest drive power $P_{\max}$	no load in the model
RMS torque M_{rms} at the drive	no load in the model

Largest reduced inertia $J_{red,max}$ at the drive	no load in the model
Smallest reduced inertia $J_{red,min}$ at the drive	no load in the model
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
Position of the largest pressure angle at gelenk1 (drive angle)	-105.8°
Reversal positions of the output link	2
Swing angle of the output link ψ_0	16.4°
Smallest quality index	0.656

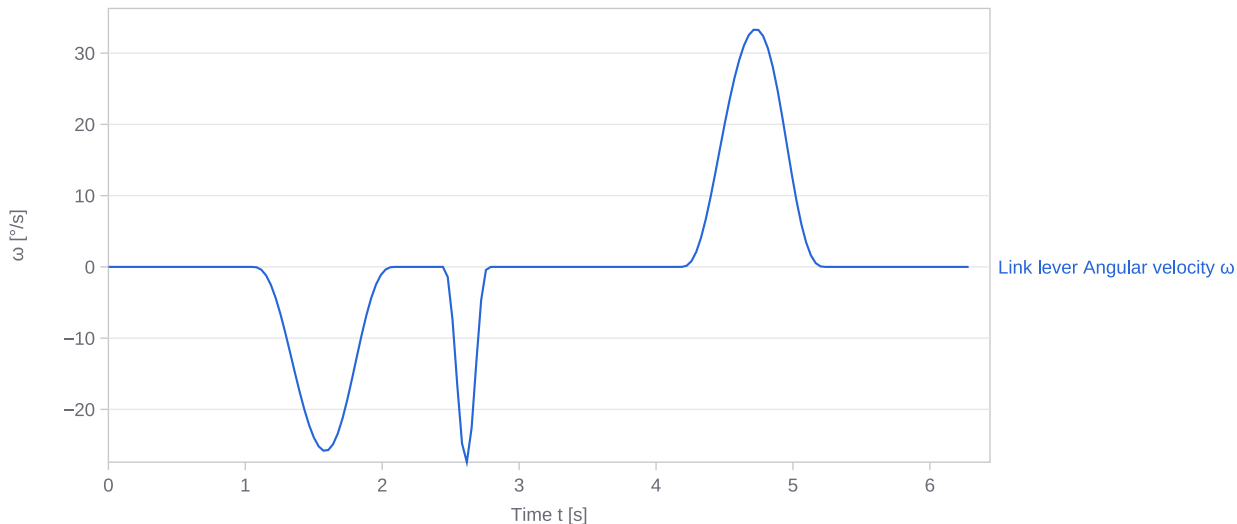
Drive angle over time [°]



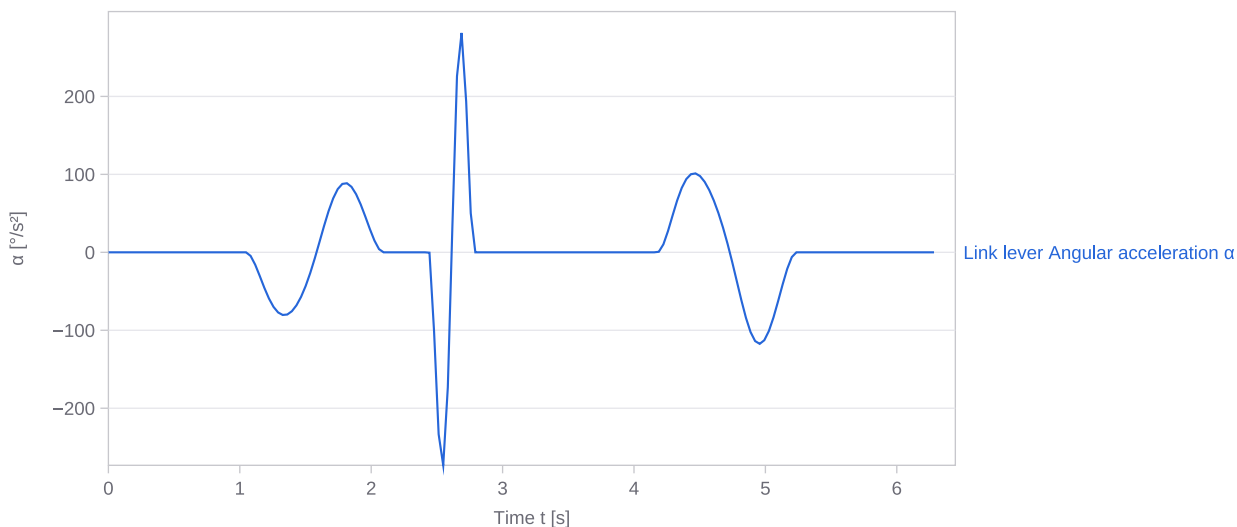
Angle over time [°]



Angular velocity over time [°/s]



Angular acceleration over time [°/s²]



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 THE CAM JOINT IS COMPUTED: The cam disc ties the roller centre to its pitch curve. In the moving frame of the disc that curve is given in polar form $r(u) = r_0 + s(u)$: u is the cam angle, r_0 the base circle, and s the travel taken from the motion plan, computed with poly4567 — the same motion law as in the cam design calculator. The constraint supplies two equations, namely "the roller centre lies on that curve", and at the same time brings an unknown of its own: the cam parameter u . It therefore takes only ONE degree of freedom from the linkage and counts as a higher joint g_2 in the count. The cam parameter is not stored but read off the drawn position — the curve is star-shaped about the pivot of the disc, so every ray from there meets it exactly once. The roller radius r_R enters no equation: the pitch curve is the path on which the centre of exactly THAT roller runs. It alone decides manufacturability — if the smallest radius of curvature of the curve is no larger than r_R , the working contour derived from it intersects itself, and the document carries a note about it and a key-value row that checks $p_{\min}$ against r_R .

PRESSURE ANGLE AT THE CAM JOINT: The disc pushes the roller along the normal of its pitch curve. The pressure angle α is the angle between that normal and the path velocity of the roller centre, computed in every position of the traverse in which the roller moves; the largest one is reported together with its position. $\mu = 90^\circ - \alpha$ is the transmission angle of the cam joint, and it is therefore checked against $90^\circ - \mu_{\min}$ — the same limit as at the four-bar, with the same traffic light —, unless a pressure angle limit α_{allow} of its own is entered at the cam joint; then that one applies. At the radial roller follower this is the closed form $\tan \alpha =$

$s'/(r_0 + s)$, at the roller lever the angle between the normal and the perpendicular to the lever; computing it from the velocity covers both and any other guidance of the roller. In a dwell the roller stands still and has no direction of motion; those positions are left out, and their angle there is the limit of the adjacent moving positions. The roller is taken as the driven member. CONTACT FORCE: The force with which the disc presses the roller normal to the path comes from the same computation as the joint forces; the smallest one over the traverse is reported together with its position. Where it is zero or less, the roller lifts off in a force-closed design, and in a grooved cam it changes flank. The motion after lift-off is not computed. Without a load in the model the force is zero everywhere, and the line stays unrated. CHECKS OF THE DISC: As in the cam calculator, the curvature safety $S = \rho_{\min}/r_R$, the smallest radius of curvature of the working contour (convex $\rho - r_R$, in a concave section $\rho + r_R$) and the smallest radius of a concave section are shown, sampled at 3000 points of the motion plan. $S \geq S_{\text{allow}}$ is checked, the contour against $S_{\text{allow}} \cdot r_R$ and the cutter radius against the smallest concave section, each only where the limit appears in the table of check limits. At the radial roller follower the base circle that just keeps the pressure angle limit is shown as well, $r_{0,\text{req}} = \max(|s'|/\tan \alpha_{\text{allow}} - s)$. If the disc rotates uniformly, the largest stroke velocity $v = s' \cdot \omega$, the largest stroke acceleration $a = s'' \cdot \omega^2$ and the RMS jerk $j = s''' \cdot \omega^3$ over the same 3000 points are shown, as in the cam calculator; otherwise ω is not fixed over the traverse, and a sentence stands in their place.

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 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.

Cam joint, two-sided constraint: The roller is tied to the pitch curve in both directions — it can neither leave the curve nor penetrate it, whichever way the contact force points. That is correct for the form-closed design: grooved cam, cam with counter cam, constant-diameter disc. For the force-closed design, in which a spring presses the roller against the cam, it is an assumption. There the roller lifts off as soon as the contact force turns into a pull. Whether and where that happens is shown by the key figure “Smallest contact force $N_{\min}$ ”. The motion after lift-off is not computed: the calculator goes on computing the position that only a form-closed disc really enforces.

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

All values on these pages come from a freely chosen example calculation and belong to no real part. The purchased export contains the same sections with your input, your header data (project, part, prepared by) and as searchable text. Pricing and scope: maschinenbaurechner.de/en/pricing

Note: The results are design and orientation aids. They do not replace independent verification, calculation and approval by a qualified person. The terms of use at maschinenbaurechner.de apply.