

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

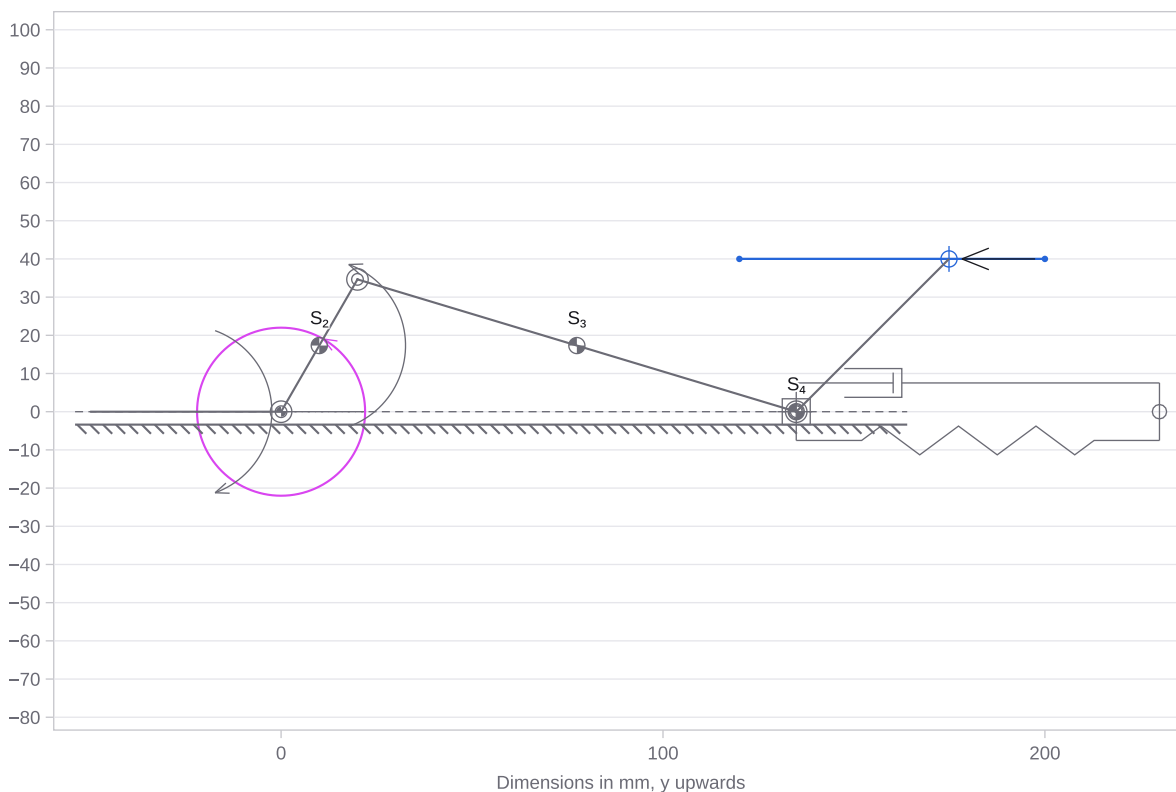
Part/Position: no real part

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.

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.

A force element is drawn as what it carries: stiffness and constant force as a spring, damping as a piston in a cylinder. If an element carries both, the two symbols stand in parallel between two connecting bars — it is ONE component between two attachment points and not an additional link. The amplitude of the spring follows its relaxed length and not the length in this position: it stretches along its axis and keeps its width. The symbols follow Hagedorn/Thonfeld/Rankers, Konstruktive Getriebelehre, plate 5.10 and VDI 2149 part 2, figure 38.

In the relaxed state the piston sits in the middle of its cylinder and travels with the change in length of the element. Its travel is limited: once the change exceeds roughly an eighth of the relaxed length it rests against its stop and from then on states only the

direction, no longer the amount. The cylinder itself is rigid and does not change size — the change in length is taken up by the straight sections, and the ends of the symbol are always the two attachment points.

Computation

Steps per traverse	360
Positions computed	365
Duration of one traverse	0.5 s
Mean speed of the drive	120 l/min
Gravity (positive acting downwards)	9.81 m/s ²
Position shown	0
Superimposed positions in the system sketch	0
Permissible drive torque M_{perm}	50 N·m
Permissible joint force F_{perm}	1,500 N

Links

Link	x_0 [mm]	y_0 [mm]	φ_0 [°]	Link line [°]	m [kg]	s_S [mm]	J_S [kg·mm ²]
gestell	0	0	0	—	—	at the reference point	—
kurbel	0	0	60.2	60.2	0.6	20 0	80
pleuel	20	34.6	−16.6	−16.6	0.9	60 0	1,080
kolben	134.9	0	0	—	2.4	at the reference point	1,040

Joints

Joint	Type	Links	Attachment 1 [mm]	Attachment 2 [mm]	Extra
fest	Ground constraint	gestell	0 0	—	$\varphi = 0^\circ$
A0	Revolute joint	gestell – kurbel	0 0	0 0	$\mu = 0.1$; $r = 10$ mm; friction circle radius $r_R = 0.995037$ mm
A	Revolute joint	kurbel – pleuel	40 0	0 0	$\mu = 0.1$; $r = 10$ mm; friction circle radius $r_R = 0.995037$ mm
B	Revolute joint	pleuel – kolben	120 0	0 0	$\mu = 0.1$; $r = 10$ mm; friction circle radius $r_R = 0.995037$ mm
S	Prismatic joint	gestell – kolben	−50 0	0 0	axis direction 0° ; $\Delta\varphi = 0^\circ$; $\mu = 0.1$; $L = 100$ mm

Drives

Drive	Link	from φ [°]	to φ [°]	Offset to link line [°]	Segments	Transitions
1	kurbel	60	420	—	linear across the whole range	free of shock and jerk

Force elements (spring, damper, constant force)

Element	Kind	Links	Attachment 1 [mm]	Attachment 2 [mm]	Rest value g_0	Parameters
feder1	Translational element	gestell – kolben	230 0	0 0	150 mm	$F_0 = 0$ N; $k = 1.2$ N/mm; $c = 0.05$ N·s/mm
daempfer1	Rotational element	gestell – kurbel	—	—	0°	$M_0 = 0$ N·mm; $k = 0$ N·mm/rad; $c = 15$ N·mm·s/rad

External loads

Load	Kind	Link	Point of application [mm]	Direction [°]	Magnitude
last1	Force at a point	kolben	40 40	180	900 N

Load course over the cycle

Load	Segment	up to τ	up to φ [°]	Value at the start	Value at the end
last1	1	0.7	−48	0	0
last1	2	0.75	−30	1	1

Load	Segment	up to τ	up to φ [°]	Value at the start	Value at the end
last1	3	0.83	-1.2	1	0.4
last1	4	1	60	0	0

Marked link points

Point	Link	x [mm]	y [mm]	x on the link [mm]	y on the link [mm]
punkt1	kolben	174.9	40	40	40

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

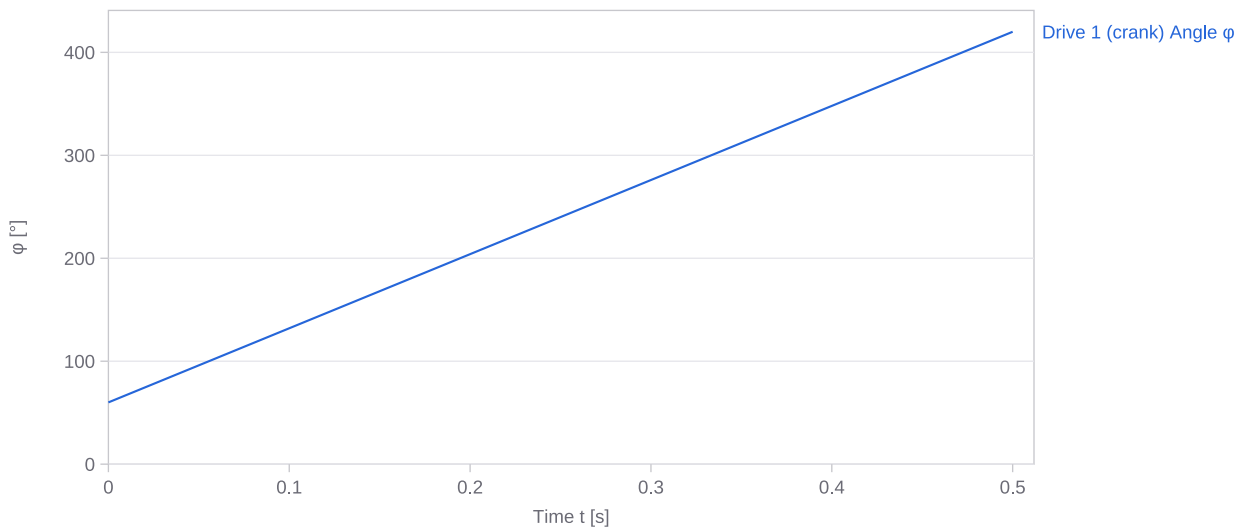
Verifications

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 torque $M_{\max} \leq 50 \text{ N}\cdot\text{m}$	41.7 N·m	50 N·m	● OK
Largest joint force $F_{\max} \leq 1,500 \text{ N}$	1,096 N	1,500 N	● OK

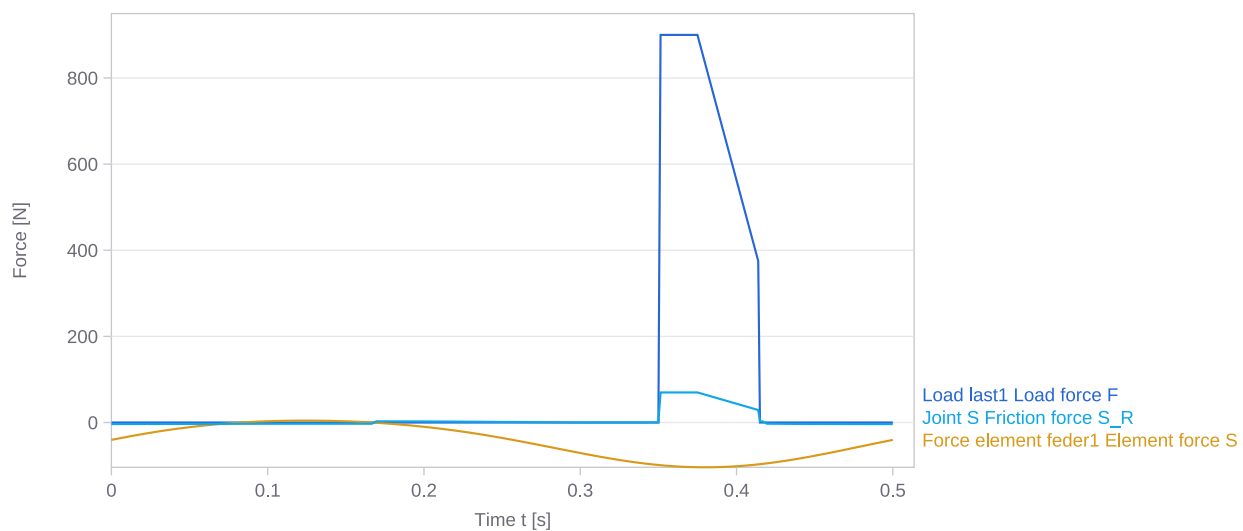
Characteristic values

Smallest transmission angle $\mu_{\min}$	70.5°
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 power $P_{\max}$	524 W
Largest force in a translational element $ S _{\max}$	104 N
Largest torque in a rotational element $ S _{\max}$	188.5 N·mm
Largest normal force in the slideway $F_{N,\max}$	698.6 N
Largest edge force of the slideway $ R _{\max}$	479.2 N
Largest friction power $P_{R,\max}$	65.8 W
RMS torque M_{rms} at the drive	9.569 N·m
Largest reduced inertia $J_{\text{red},\max}$ at the drive	6,090 kg·mm ²
Smallest reduced inertia $J_{\text{red},\min}$ at the drive	800 kg·mm ²
Largest shaking force $F_{R,\max}$	28.74 N
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 $\mu_{\min}$ (drive angle)	90.0°
Reversal positions of the output link	2
Stroke of the output link s_0	80.00 mm
Time ratio of forward to return stroke	1.000
Smallest quality index	0.709

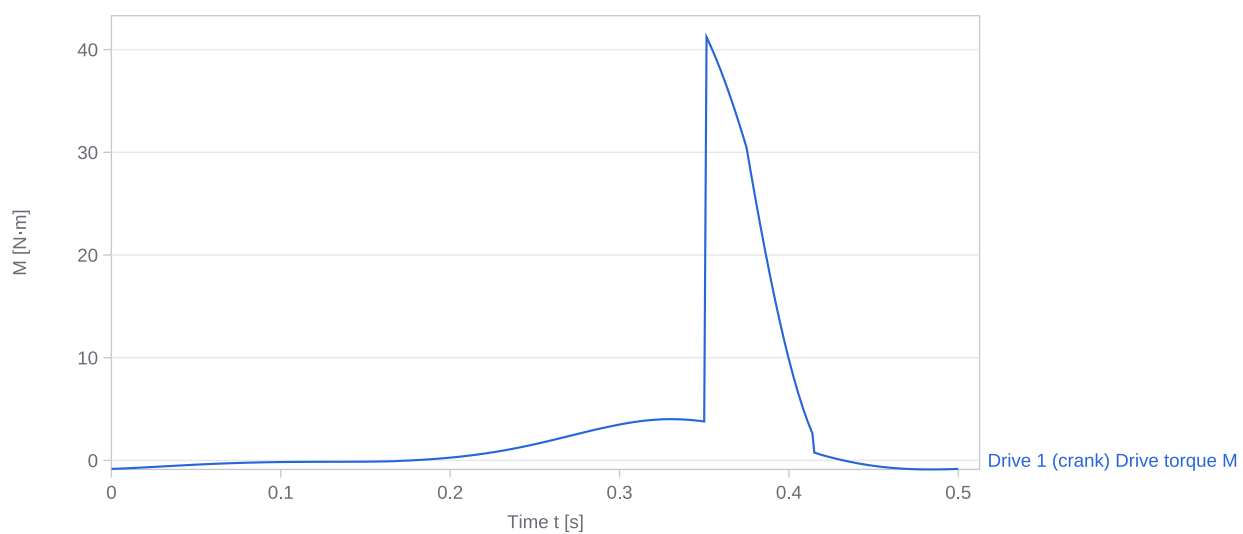
Drive angle over time [°]



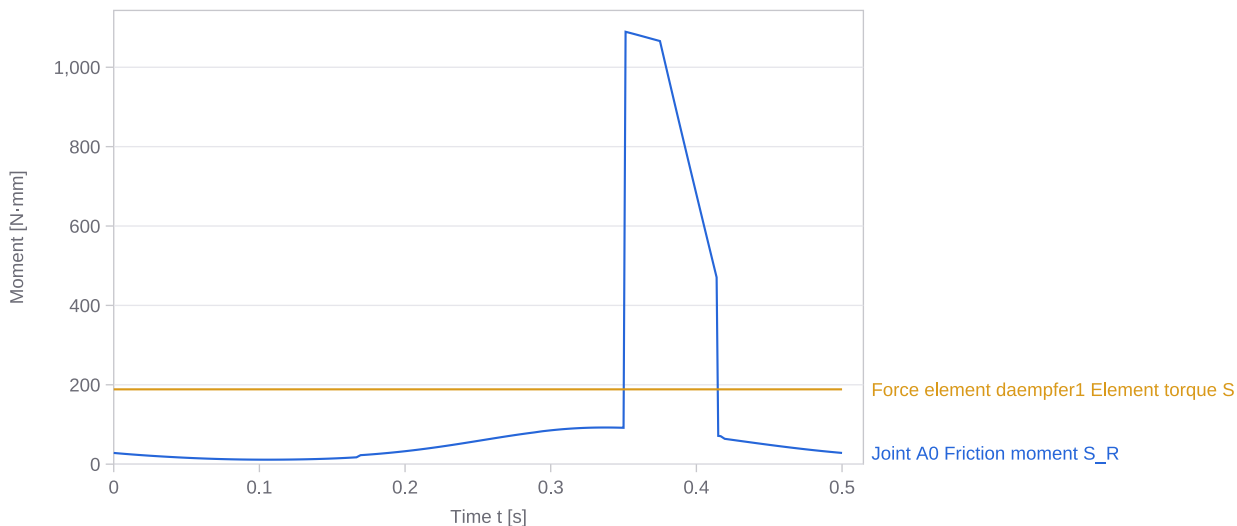
Force over time [N]



Drive torque over time [N·m]



Moment over time [N·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.

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 A LOAD COURSE IS COMPUTED: the table “Load course over the cycle” divides the common timeline into segments; each carries its right bound τ , a value at the start and one at the end, with linear interpolation between them. The values are dimensionless: what is computed is the magnitude of the load times that value — the course carries the SHAPE, the magnitude the SIZE, and the unit therefore stays in exactly one place. A value of 1 means “fully present”, 0 means “not present here”, negative means the opposite direction. It is given over the progress τ and not over the drive angle: with more than one drive there is no drive angle at all, and with a swinging drive the same angle is reached twice per traverse — a specification over it would not be a function there. With exactly one drive the table states the corresponding angle beside it; with several a dash appears there.

WHY THE DRIVE TORQUE JUMPS: an applied force may set in abruptly at a segment bound — unlike a position, whose second derivative enters the computation through the inertia and which therefore has to be continuous. The drive torque is the sum of an inertia, a weight and a load share; if the load share jumps, the sum jumps with it, and the jump is as large as the change of the load times its transmission in that position. On the real machine what remains of it is the shock as the tool touches down; how high the peak really rises depends on the stiffness of drive train and tool, and a rigid-body model does not know it. The computed jump is therefore the lower bound. So that the peak does not fall between two computed positions, every segment bound places two additional sampling points into the grid of the traverse — a window narrower than one grid step would otherwise not be hit at all, and the torque would stay smooth without anything being reported.

HOW JOINT FRICTION IS COMPUTED: at a revolute joint the joint force does not pass through the centre of the pin but is tangent to a circle around it. Its radius is $r_R = r \cdot \mu / \sqrt{1 + \mu^2}$ with the PIN radius r , and the friction moment is $M_R = r_R \cdot |G|$ with the joint force G (Kerle/Pittschellis/Corves, Einführung in die Getriebelehre, 3rd ed. 2007, eqs. 5.6 and 5.7, p. 128). It opposes the relative rotation of the two links. The friction circle always lies inside the pin, because $\mu / \sqrt{1 + \mu^2}$ is less than one; a freely entered friction radius is therefore not offered — it would have nothing to be checked against. **EXPLICITLY:** the common form $\mu \cdot r \cdot |G|$ is too large by the factor $\sqrt{1 + \mu^2}$, by 0.5 % at $\mu = 0.1$, by 4.4 % at $\mu = 0.3$ and by 11.8 % at $\mu = 0.5$. Whoever recomputes with it and finds a deviation of that size has not found an error but the other formula.

HOW THE FRICTION FORCE IN A PRISMATIC JOINT IS COMPUTED: if the guide takes a moment, the slider tilts and presses at two places whose forces are opposed. With the transverse force N , the moment M and the guide length L the edge forces are $R_a = N/2 + M/L$ and $R_b = N/2 - M/L$, and the friction force is $F_R = \mu \cdot (|R_a| + |R_b|)$ — the sum of the MAGNITUDES and not the magnitude of their sum. Multiplying the resultant transverse force by μ computes too small a value, that is, on the unsafe side. The two edge forces are drawn in Hagedorn/Thonfeld/Rankers, Konstruktive Getriebelehre, 6th ed. 2009, plate 6.1 e and f; the conversion to N and M that this solver yields is DERIVED from it and appears in none of the works reviewed — it matches the printed plate case line by line and passes into the simple case $\mu \cdot |N|$ as the overhang goes to zero. A short guide with a long overhang multiplies the edge forces by $1 + 2 \cdot l/L$ with the overhang l .

WHAT THE FRICTION MODEL DOES NOT COVER: only Coulomb sliding friction is computed, with a coefficient held fixed over the cycle, and only at those joints for which the joint table states one. **STICTION IS NOT MODELLED:** at a reversal, where the relative velocity passes through zero, the friction force is statically indeterminate and bounded only from above; there the computation sets the friction moment to zero, because no friction power flows. A breakaway torque from static friction therefore appears in no figure — the static coefficient is typically some tenth above the sliding one. Stick-slip cannot arise in this model at all: its cause is a coefficient that falls with relative velocity, and here it is constant; that is a model boundary, not an inaccuracy. The friction moment JUMPS at a reversal by twice its magnitude — the jump is real and not a numerical artefact; the friction

power does not jump, it carries the magnitude of the relative motion and goes continuously to zero. Static friction appears only in the SELF-LOCKING verification, on the stationary mechanism and with a coefficient of its own, μ_0 ; where a joint carries one, a paragraph of its own covers it. At a cam joint no friction is computed, because the model does not say whether it rolls or slides there; an invented split into rolling and sliding shares would be worse than the acknowledged gap. Joint clearance stays outside the model, and it is precisely at the zero crossings of the joint force that the computation is least reliable. Coefficients of friction are not material constants but properties of the whole system of material pairing, lubrication and contact pressure and have to be determined experimentally; for the friction torque referred to the drive shaft, VDI 2149 part 1 explicitly recommends a run-down test on the actual mechanism. Even with friction this remains a KINETOSTATIC computation and not a dynamic one: the motion is prescribed by the drive, and friction changes the drive torque and the joint forces, not the path and not the accelerations.

REDUCED INERTIA AND RMS TORQUE, with exactly one rotary drive: the influence coefficients $dq/d\phi$ follow from the same Jacobian as the velocities, with unit rate at the drive. J_{red} is then the sum over the links of $m \cdot |dr_S/d\phi|^2 + J_S \cdot (d\phi_i/d\phi)^2$, even in a dwell in which the mechanism stands still. By the principle of virtual work the drive torque splits into the inertia share M_T , the sum of $m \cdot a_S \cdot dr_S/d\phi + J_S \cdot \alpha_i \cdot d\phi_i/d\phi$, equal to $J_{red} \cdot \alpha + \frac{1}{2} \cdot (dJ_{red}/d\phi) \cdot \omega^2$, and the load share M_L from weight, force elements and loads. The RMS torque $M_{rms} = \sqrt{(1/T) \cdot \int M^2 dt}$ over the motion time T is given only for a traverse without gaps; it is integrated by the trapezoidal rule over the computed positions. If the motor sits on a link other than the driven one, torque and inertia apply at that link: $M_{link} = M/(d\phi_{link}/d\phi)$.

THE SHAKING FORCE is the force the mechanism passes to the frame from its inertia forces, $F_R = -\text{sum of } m \cdot a_S$ over the moving links (VDI 2149 part 1, section 3). Weight and external loads are not part of it: the one is static, the others come from outside.

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 μ_{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 ω_{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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