

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

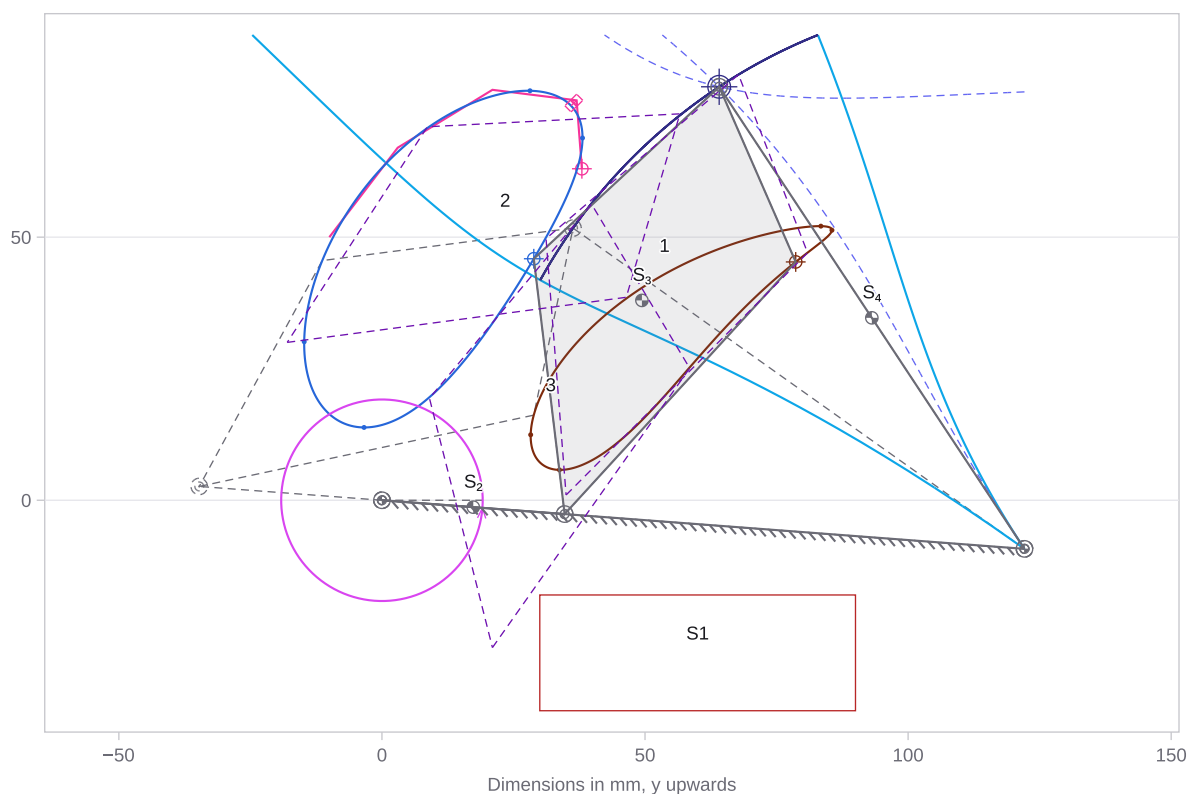
Part/Position: no real part

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.

Every marked point carries its own curve colour, and its crosshair and reversal marks have the same tone as its path. That is how to tell which curve belongs to which point, even where two of them cross.

Traced joints draw their trace in a second colour series, separate from the coupler curves. The crosshair around a joint sits where that joint stands in the position shown — the same symbol as at a marked link point.

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.

Also drawn is the given target path — the curve the author dragged on the canvas. It is a specification and not a result. The cross sits on its first vertex.

The dashed outlines with the digits are the prescribed positions of the moving plane: they show the shape of the link named below at the location and rotation the author prescribed. They too are a specification and not a result — the mechanism does not necessarily reach them. The digit is the place in the order and not a point in time; the table below states the numbers of each.

The closed outlines labelled S are the prescribed blocked areas — frame, installation space or fixture, into which no link may move. They too are a specification and not a result. Every moving link is checked over the whole cycle, including the band it sweeps between two computed positions; the frame and the links among themselves are excluded. A link is the connecting line of its attachment points and has NO WIDTH: the outline checked is the centre line of the part, not its contour. Anyone building with a 12 mm flat bar widens the blocked area beforehand by half the part width. Where exactly the corners of each area lie is not stated in this document but to scale in millimetres in the CAD file of the same export, on layer SPERRFLAECHE — a table here would carry three hundred coordinate pairs for five areas of sixty corners each, and nobody types those in.

The short line with the two diamonds is the largest deviation between target path and coupler curve: one diamond sits on the target path, the other on the path of the compared part, and the length of the line IS the value stated above. The correspondence is by nearest point — for every place on the target path the nearest location of the coupler curve, whatever crank angle reaches it. That is a pure comparison of shape; whether the curve also passes the path at the right time it does not say.

Also drawn are the two centrodes of the selected link: the fixed centrode belongs to the frame, the moving centrode belongs to the link and rolls on it. In every position the two touch exactly at the instantaneous centre. They belong to the whole cycle and are clipped to the visible frame; the centre itself often lies far outside.

No instantaneous centre is drawn for this position, and that is not an omission: the link does not move here. A jerk-free motion law starts the cycle from rest, and a link at rest has no instantaneous centre.

Computation

Steps per traverse	360
Positions computed	361
Duration of one traverse	0.5 s
Mean speed of the drive	120 1/min
Gravity (positive acting downwards)	9.81 m/s ²
Position shown	0
Superimposed positions in the system sketch	2
Instantaneous centre and centrodes of	koppel
Target path, vertices	5
Target path, course	open
Target path compared with the path of	P1
Prescribed positions, plane of	koppel
Prescribed positions, count	3
Blocked areas checked	1
Permissible drive torque M _{perm}	0.35 N·m
Smallest permissible transmission angle μ_{perm}	40 °
Permissible deviation from the target path	3 mm

Path search: the run this linkage came from

Objective function searched for	root mean square of the deviation
Quantities searched for	frame, crank, coupler, rocker, direction of the frame line, coupler point x, coupler point y
Quantities held fixed	A ₀ x, A ₀ y, scale, offset of the drive range
Smallest transmission angle μ required for the search	40 °
Crank had to rotate fully	yes
Keep-out areas were included	yes
Search method	differential evolution, then Nelder-Mead, across eight passes
Starting point of this pass	from the coupler curve atlas
Random seed of the run	1265200741
Random seed of this pass	638573432
Pass this linkage came from	4
Evaluations in this pass	25,002
Time budget of the run	20 s
End of this pass	evaluation budget spent

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	-4.3	—	at the reference point	—
kurbel	0	0	-4.3	-4.3	0.131	17.41 0	13.2
koppel	34.72	-2.63	70.1	—	0.326	43.18 0	203
schwinge	64.09	78.58	-56.5	-56.5	0.397	52.63 0	367

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	without friction
A	Revolute joint	kurbel – koppel	34.82 0	0 0	without friction
B	Revolute joint	koppel – schwinge	86.35 0	0 0	without friction
B0	Revolute joint	gestell – schwinge	122.12 -9.24	105.25 0	without friction

Drives

Drive	Link	from φ [°]	to φ [°]	Offset to link line [°]	Segments	Transitions
1	kurbel	-4.3	355.7	—	1	free of shock and jerk

Motion laws per segment

Drive	Segment	to τ	to fraction	Law	At the end
1	1	1	1	poly345	Stop

Marked link points

Point	Link	x [mm]	y [mm]	x on the link [mm]	y on the link [mm]
punkt1	koppel	28.859	45.895	43.634	22.017
punkt2	koppel	78.639	45.292	60	-25

Prescribed positions of the moving plane

No.	Name	x [mm]	y [mm]	φ [°]
1	L1	35	1	67.7
2	L2	-18	30	30.2
3	L3	21	-28	77.4

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

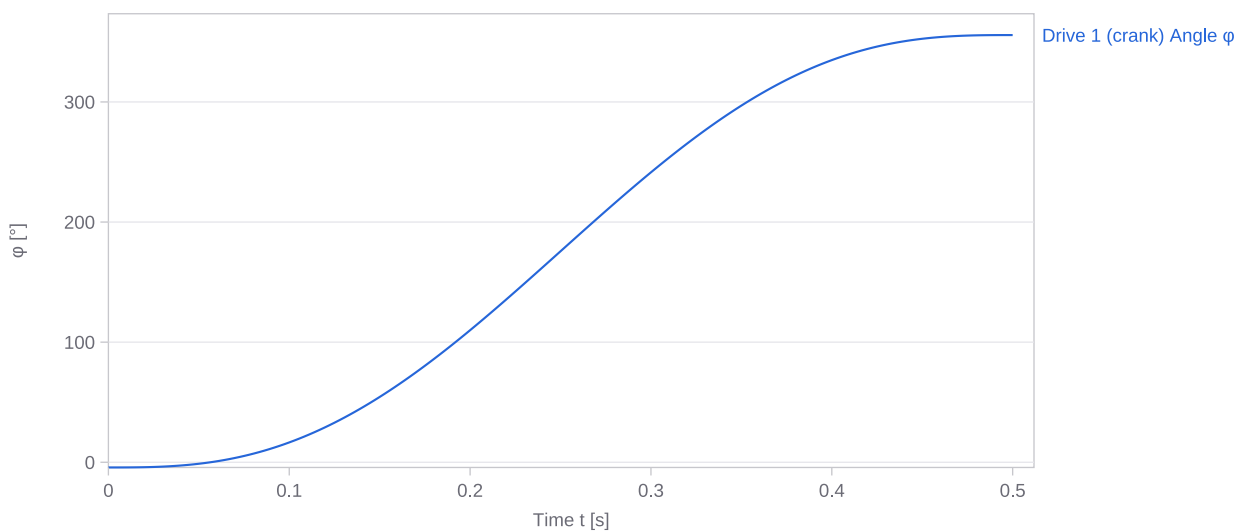
Verifications

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
Smallest transmission angle $\mu_{\min} \geq 40.0^\circ$	53.3 °	40.0 °	● within guideline
Largest deviation from the target path (path of punkt1) ≤ 3.000 mm	1.475 mm	3.000 mm	● OK
Largest penetration into a blocked area	0.000 mm		● OK
Largest drive torque $M_{\max} \leq 0.35$ N·m	0.2478 N·m	0.35 N·m	● OK

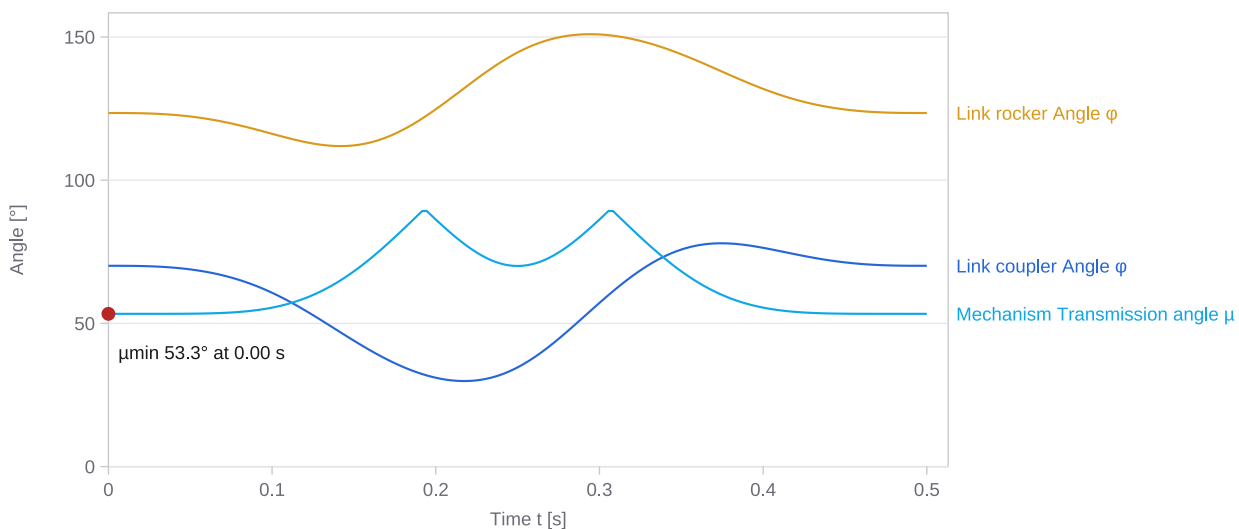
Characteristic values

Smallest locking factor $r_{\min}$ (self-locking)	no μ_0 entered
Largest joint force $F_{\max}$	15.79 N
Largest drive power $P_{\max}$	5.836 W
RMS torque M_{rms} at the drive	0.139 N·m
Largest reduced inertia $J_{\text{red,max}}$ at the drive	763.3 kg·mm ²
Smallest reduced inertia $J_{\text{red,min}}$ at the drive	182.6 kg·mm ²
Largest shaking force $F_{R,\max}$	8.781 N
Type according to Grashof	crank-rocker
Shortest link	crank
Grashof excess $(s + l) - (p + q)$	-34.3 mm
Position of $\mu_{\min}$ (drive angle)	-4.3°
Position of the largest path deviation (progress τ)	0.305
Position of the largest path deviation (drive angle)	56.6°
Correspondence used for the path comparison	nearest point
Reversal positions of the output link	2
Swing angle of the output link ψ_0	39.1°
Time ratio of forward to return stroke	2.279
Smallest quality index	0.674

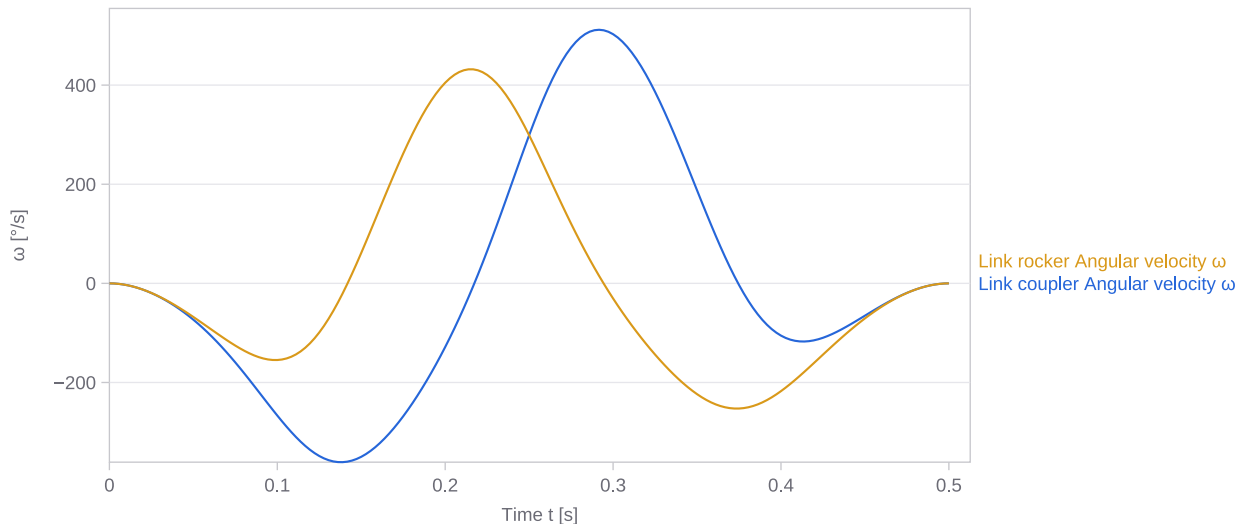
Drive angle over time [°]



Angle over time [°]



Angular velocity over time [°/s]



Roberts cognate linkages

Coupler point whose curve is compared	punkt1
Third frame pivot C_0 , x	64.060 mm
Third frame pivot C_0 , y	26.468 mm
Check: positions traversed over one full revolution	3,600
Check: largest deviation of a link length from its nominal value	5.68E-14 mm

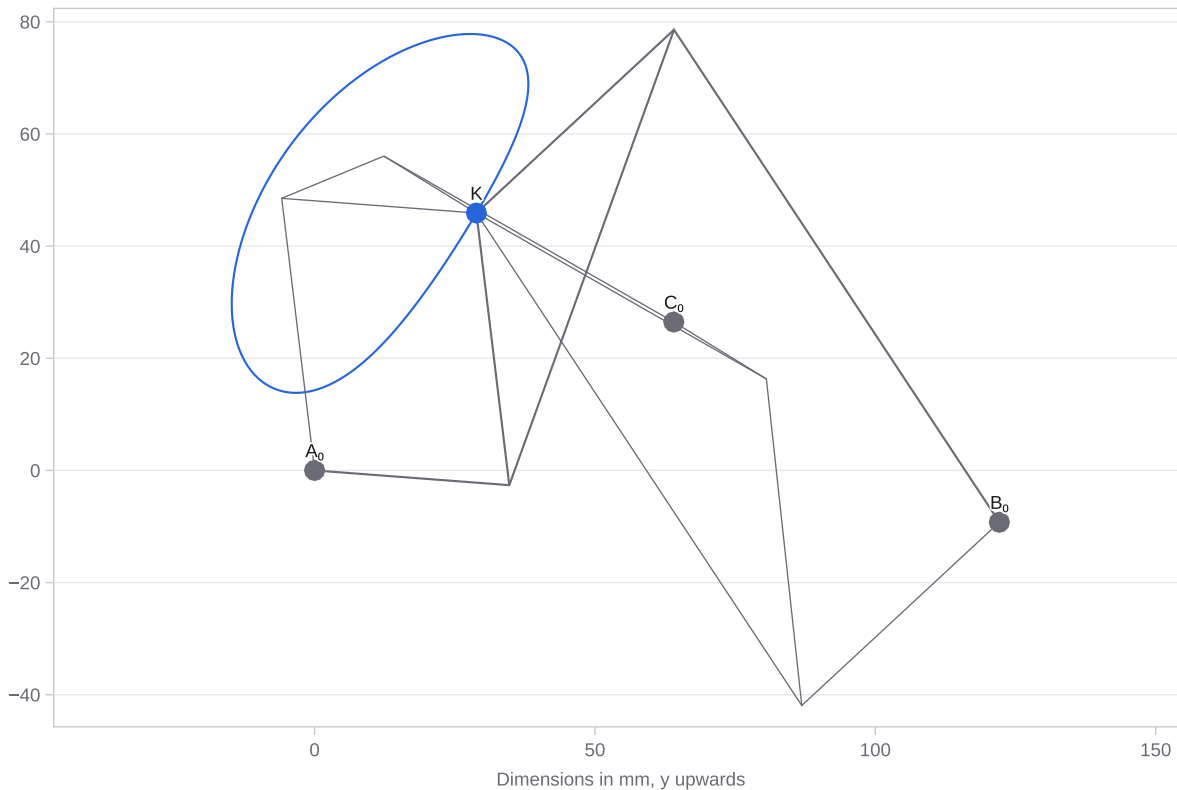
Which one fits the installation space?

Quantity	Original	Cognate 1	Cognate 2
Frame (mm)	122.466	69.313	68.157
Driven link (mm)	34.820	48.874	19.379
Coupler (mm)	86.354	19.707	58.578
Rocker (mm)	105.253	59.571	48.060
Frame pivot A_0 , x y (mm)	0.000 0.000	0.000 0.000	64.060 26.468
Frame pivot B_0 , x y (mm)	122.117 -9.237	64.060 26.468	122.117 -9.237
Coupler point from the leading coupler joint (mm)	48.874	34.820	59.571
Coupler point from the trailing coupler joint (mm)	48.060	19.379	105.253
Scale factor relative to the original mechanism	1.0000	0.5660	0.5565
Grashof type	crank-rocker	double rocker	crank-rocker
Driven link rotates fully	yes	no	yes
Installation space over the drive range, width × height (mm)	156.9 × 123.2	76.1 × 77.8 — smallest space	136.9 × 124.3
Longest link (mm)	122.5	69.3	68.2
Keep-out area	clear	clear	11.00 mm deep

The installation space is the smallest axis-parallel box the linkage stays within across a full drive range — measured on the link centre lines, without material thickness. It says something different from the longest link: a cognate with longer bars can need less room because its bearings sit elsewhere. Two rows with the same four lengths are therefore not yet the same mechanism: only the frame pivots say which one sits where.

Grashof type and driven link say different things. Grashof classifies the CHAIN, the driven-link row says whether the link at A_0 itself rotates. In a crank-rocker the shortest link is either the crank or the rocker; if it is the rocker, the output link rotates and the driven link oscillates. “Crank-rocker” next to “no” is therefore no contradiction.

All three linkages on one coupler curve



Drawn is the reference position. The thick chain is the original mechanism between A_0 and B_0 , the thin ones are the cognates: number 1 between A_0 and C_0 , number 2 between C_0 and B_0 . All three carry the same coupler point K , and the single curve drawn belongs to all three. Dimensions in mm, equal scale on both axes.

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.

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 $m \cdot |dr_S/d\phi|^2 + J_{S_i} \cdot (d\phi_i/d\phi)^2$, 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 $m \cdot a_{S_i} \cdot dr_S/d\phi + J_{S_i} \cdot \alpha_i \cdot d\phi_i/d\phi$, 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.

WHY THE THREE LINKAGES TRACE THE SAME CURVE — Roberts' theorem: the coupler point K is fixed in the moving frame of the coupler; written as a complex ratio, $z = (K - A)/(B - A)$, and this number depends on no position, because numerator and denominator turn with the coupler alike. With it the third frame pivot follows from $C_0 = A_0 + z \cdot (B_0 - A_0)$; the frame triangle $A_0B_0C_0$ is thereby similar to the coupler triangle ABK . The first cognate hangs on A_0 and C_0 , and its moving joints are $A_1 = A_0 + (K - A)$ — the parallelogram A_0AKA_1 — and $B_1 = C_0 + z \cdot (B - B_0)$. Working it out gives $|A_1 - A_0| = k$, $|B_1 - A_1| = |z| \cdot a$, $|B_1 - C_0| = |z| \cdot c$ and $|C_0 - A_0| = |z| \cdot d$, together with $|K - A_1| = a$ and $|K - B_1| = |1 - z| \cdot a$. All six magnitudes are constant over the whole revolution — precisely that makes the chain a mechanism, and because K remains the same point as for the original, it is the same curve. The second cognate follows from the same rule read from the other side: with $w = 1 - z$ one has $K = B + w \cdot (A - B)$, and exchanging A_0, A, B, B_0 and z for B_0, B, A, A_0 and w maps the construction onto itself — it yields the same point C_0 and the joints $A_2 = C_0 + w \cdot (A - A_0)$ and $B_2 = B_0 + (K - B)$. THIS IS NOT ASSERTED BUT MEASURED: the two rows headed “Check” traverse one full

revolution of the original mechanism, build the cognate joints in every position by the rule above and compare five distances per cognate against the nominal values from the formulas. The largest deviation appears above in millimetres; it is of the order of machine precision and is not a tolerance. WHAT DOES CHANGE IS THE GRASHOF TYPE: the cognates carry the same lengths in different roles, and therefore a crank-rocker can turn into a Grashof double-rocker. The row "Driven link rotates fully" states it per linkage; where it reads "no", that linkage traces the same curve, but its drive covers only part of it per assembly configuration. Source: Kerle/Pittschellis/Corves, Einführung in die Getriebelehre, 3rd ed., ch. 6.3.1, pp. 185–188, table 6.2 on p. 186.

HOW THIS LINKAGE WAS SEARCHED FOR — path synthesis by optimisation: unlike the four design methods above this is NOT a closed calculation. Eleven numbers are searched for — four link lengths, the frame pivot A_0 , the direction of the frame line, the coupler point within the coupler plane, a scale and the offset of the drive range — such that the coupler curve matches the drawn target path as closely as possible. Which of them were actually searched for and which kept their value is stated above under "quantities searched for" and "quantities held fixed". The assembly branch is NOT a twelfth number: nothing lies between the two assemblies of a four-bar linkage that could be shifted by a little, and an average of the two would mean nothing — it is searched for through the passes, which alternate between them. Two methods run one after the other: differential evolution after Storn/Price sweeps the whole space, the simplex method after Nelder/Mead refines the best find, and both alternate three times. Eight passes search independently, each with its own derived random seed; at the end they are compared by the same objective value. The hard constraints — usable numbers, a closing quadrilateral, the length limits, the required full rotation of the crank — are handled by the feasibility rules of K. Deb (2000): a feasible candidate beats any infeasible one, and among two infeasible ones the smaller violation wins. The required transmission angle and the keep-out areas, by contrast, enter the objective value as a PENALTY and not as a limit; the reason is given in the assumptions below. The search runs on a coarser grid than the measurement — 180 rotation steps and 240 sampling points along the path — because hundreds of thousands of evaluations on the certificate grid are not affordable. The find is then measured ONCE on the certificate grid (at least 1800 rotation steps, about a thousand sampling points), and only that figure appears above in this document.

WHERE THE STARTING POINT CAME FROM — the coupler curve atlas: before the search, a precomputed table of four-bar linkages is searched by the SHAPE of their coupler curve. For a closed target path the whole curve is compared through Fourier descriptors; for an openly drawn one a PIECE of the coupler curve is sought that resembles the drawn path. The closest matches that keep the required transmission angle come first; the first states which valley to start in and supplies the assembly branch as well. A quarter of the initial population is scattered around it, three quarters keep sweeping the whole space. What this achieves has been measured: across thirty tasks with a known solution — six four-bar linkages, five random seeds each — FIVE runs out of thirty were left in a local minimum when started from the atlas, thirteen without it; the median remaining deviation fell from 0.453 to 0.045 per cent of the path extent, at the same computing time. THE PIECEWISE SEARCH HAS A MEASUREMENT OF ITS OWN: it recognises its own entry down to three eighths of the curve, and the linkage it returns draws the requested path to within two per cent of its extent in every case measured. Below three eighths a piece becomes too smooth to tell from hundreds of others — it still serves as a starting point, not as an answer.

WHAT WAS SEARCHED FOR — the SHAPE of the path and not its timing: correspondence is by nearest point, that is, for every place on the target path the nearest location of the coupler curve, whatever crank angle reaches it. Whether the curve passes a place at the right moment does not enter the objective value. Measured across thirty tasks whose target path was generated from an existing four-bar linkage: the shape search finds the shape to within 0.045 per cent of the path extent (median) and misses the time-correct correspondence by 8.1 to 11.7 per cent — not a single one of the thirty runs comes below one per cent there. If the search is left in a local minimum it gets far worse: on one of the six linkages the time-correct deviation then lay between 286 and 426 per cent, so the curve found is traversed in an entirely different order. Whoever requires a path speed — adhesive dispensing, flame cutting, feeding — chooses the time-correct objective in the search panel.

THE MOST IMPORTANT ASSUMPTION: a search finds a GOOD linkage, not the best one. There is no proof that no better one exists — and none that a second run with a different seed would not find one. Measured across thirty tasks whose solution was known, because the target path was generated from an existing four-bar linkage, the search was left in a local minimum in five to thirteen cases (the figure depends on the starting point and is given in the paragraph above); more computing time changes nothing there, because what is missing is not budget but a different valley. THE RUN IS REPEATABLE THROUGH THE SEED: the same seed, the same settings and the same target path yield character for character the same linkage — with one exception, stated above under "end of this pass": a run that ends at the TIME BUDGET ends at a different place on a fast machine than on a slow one and is therefore not repeatable. TRANSMISSION ANGLE AND KEEP-OUT AREAS ARE NOT LIMITS but penalties: a linkage that misses them is rated worse, not discarded — otherwise the search would be torn apart exactly where the good path generators lie, and a link penetrating slightly would lose its path value even though a few millimetres of frame offset would resolve it. The winner may therefore miss both; WHETHER it does is not stated here but above, in the rows on the transmission angle and the penetration depth.

HOW THE DEVIATION FROM THE TARGET PATH IS MEASURED: the given path is compared with the path of the part stated above — the coupler curve of a marked link point, or the track of a joint, both only on a moving link. For every place on the target path the nearest location of the coupler curve is sought, and the largest of these distances is reported — the classical path error. The correspondence is therefore a pure comparison of SHAPE: which point of the path is reached at which crank angle does not enter, and a timing error stays invisible in this figure. Measurement is FROM the target path and not the other way round; what the coupler curve does beyond the target path — with a closed curve, the return stroke — therefore does not enter. Two refinements belong to the statement: the target path is sampled BETWEEN its vertices (about a thousand places along its length, plus the vertices themselves), because between two support points the deviation may grow arbitrarily large — measured only at the vertices, a path whose vertices lie on the curve yields exactly zero, while one and a half millimetres lie in between (recomputed in ``bahnvergleich.test.ts``). And the coupler curve is recomputed for the measurement with at least 1800 steps, never with fewer than the display draws: at 360 steps the value so measured lies 0.0009 mm below the limit of a 36,000-step computation, at 36 steps already 0.15 mm, and for a display from whose raster the path itself was taken it would come out as zero. Whoever has the display drawn with more than 1800 steps gets the same number of steps for the measurement — otherwise the figure would be computed against a coarser polygon than the one on screen. A rating is given only if a permissible path deviation is stated; without one the figure stands as a characteristic value, because an approximation to a drawn curve is neither a passed nor a failed verification.

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.

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