

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

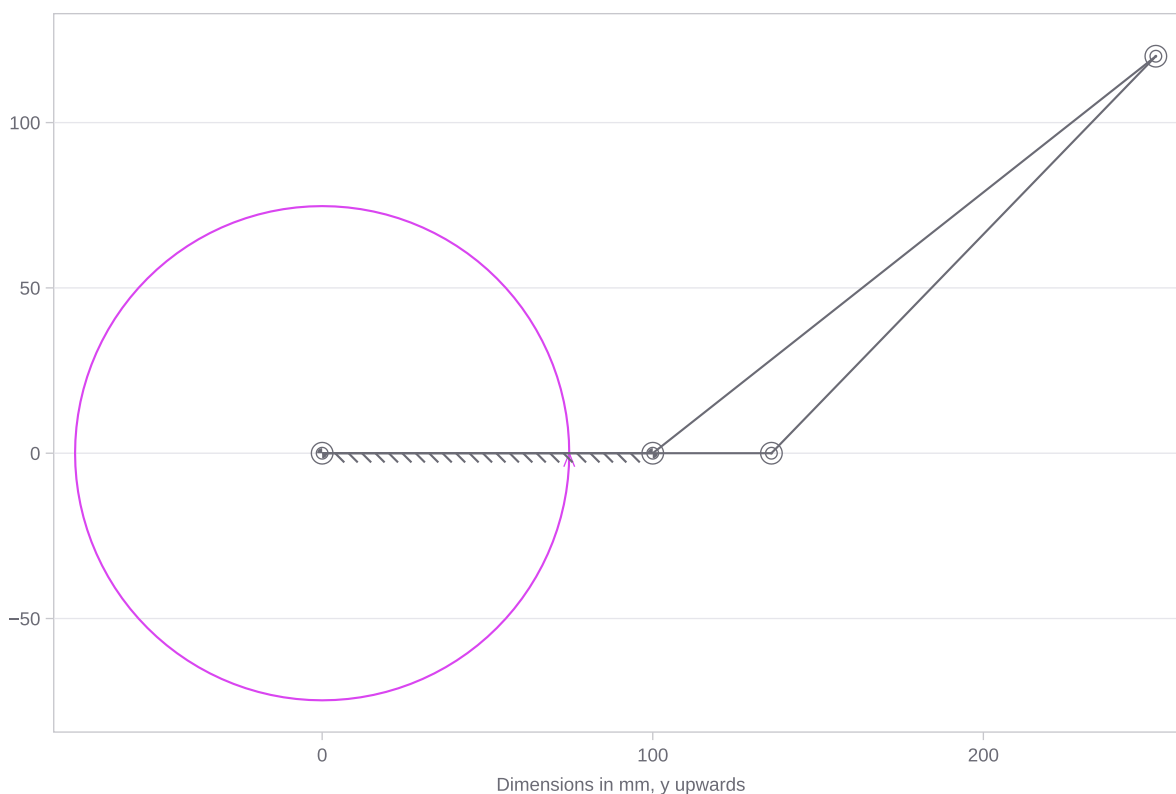
Part/Position: no real part

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 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

Freudenstein function synthesis

Method	Target function and range, precision points at Chebyshev spacing
Given: frame l_1 (freely chosen scale)	100 mm
Given: target function $y(x)$	$y = 1/x^2 \quad (x \neq 0)$
Given: range of the input variable x	1 ... 2
Given: input angle φ at the start of the range	10 °
Given: input angle φ at the end of the range	70 °
Given: output angle ψ at the start of the range	80 °
Given: output angle ψ at the end of the range	170 °
Coefficient $K_1 = l_1/l_2$	0.735895
Coefficient $K_2 = l_1/l_4$	0.51589
Coefficient $K_3 = (l_2^2 - l_3^2 + l_4^2 + l_1^2)/(2 \cdot l_2 \cdot l_4)$	0.723222
Result: input link $l_2 = l_1/K_1$	135.889 mm
Result: coupler $l_3 = \sqrt{(l_2^2 + l_4^2 + l_1^2 - 2 \cdot l_2 \cdot l_4 \cdot K_3)}$	167.151 mm
Result: output link $l_4 = l_1/K_2$	193.84 mm
Ratio l_2/l_1	1.358889
Ratio l_3/l_1	1.671507
Ratio l_4/l_1	1.938399
Reference direction of the input offset by	0 °
Reference direction of the output offset by	0 °
Assembly configuration	crossed
Largest residual of the equation, computed from the designed lengths	3.05E-16
Check from the dimensions: does the linkage reach its precision points?	The mechanism reaches every prescribed position, in the prescribed order.
Largest structural error between the precision points	-0.005924
It occurs at $x =$	1.2333
Range of y over the interval	0.75
Largest structural error relative to that range	0.79 %
Sampling points of the error curve at which the chain does not close	0 / 61

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	0	—	at the reference point	—
kurbel	0	0	0	0	—	at the reference point	—
koppel	135.89	0	45.9	45.9	—	at the reference point	—
schwinde	252.17	120.07	-141.7	-141.7	—	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 – kurbel	0 0	0 0	without friction
A	Revolute joint	kurbel – koppel	135.89 0	0 0	without friction
B	Revolute joint	koppel – schwinde	167.15 0	0 0	without friction
B0	Revolute joint	gestell – schwinde	100 0	193.84 0	without friction

Drives

Drive	Link	from φ [°]	to φ [°]	Offset to link line [°]	Segments	Transitions
1	kurbel	0	360	—	linear across the whole range	free of shock and jerk

Precision points of the function synthesis

No.	x	φ [°]	ψ [°]
1	1.066987	14.0192	94.5946
2	1.5	40	146.6667
3	1.933013	65.9808	167.8847

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

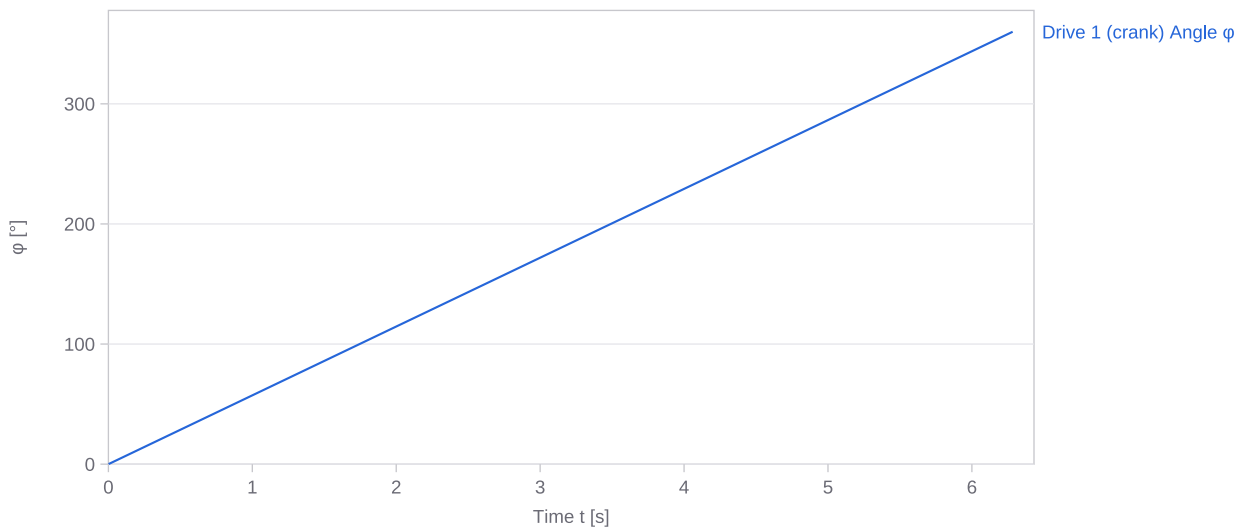
Verifications

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

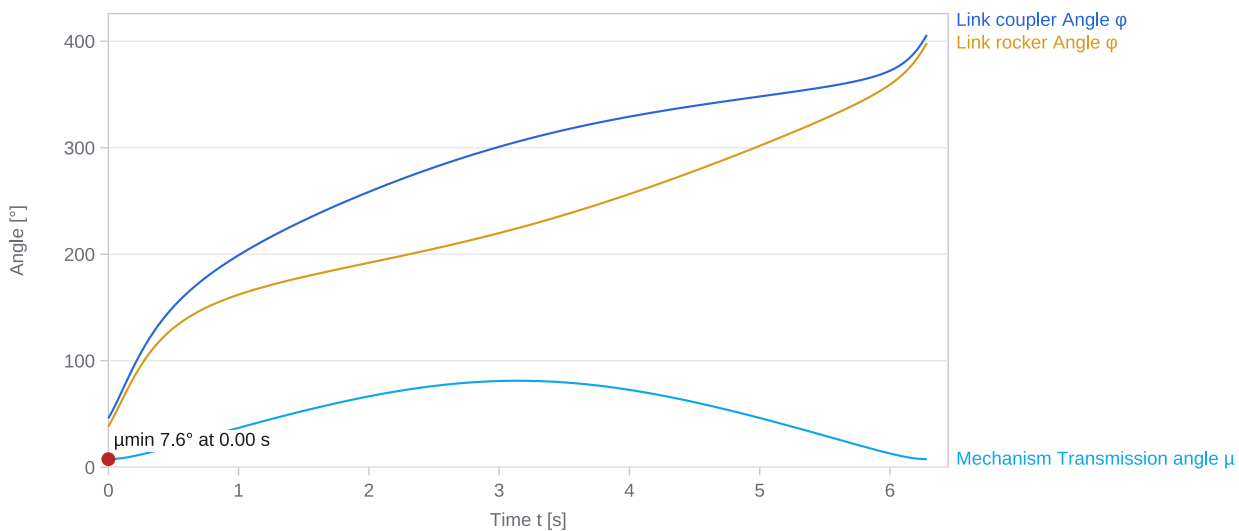
Characteristic values

Smallest transmission angle $\mu_{\min}$	7.6°
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 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 $J_{\text{red,max}}$ at the drive	no load in the model
Smallest reduced inertia $J_{\text{red,min}}$ at the drive	no load in the model
Largest shaking force $F_{R,\max}$	no load in the model
Type according to Grashof	double crank
Shortest link	frame
Grashof excess $(s + l) - (p + q)$	-9.2 mm
Position of $\mu_{\min}$ (drive angle)	0.0°
Reversal positions of the output link	0
Smallest quality index	0.582

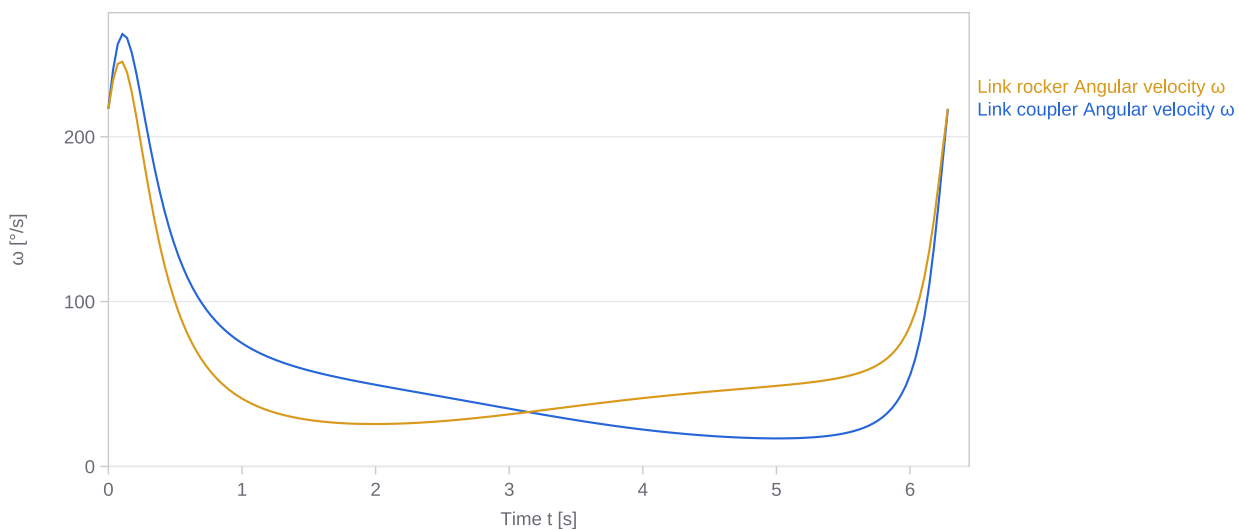
Drive angle over time [°]



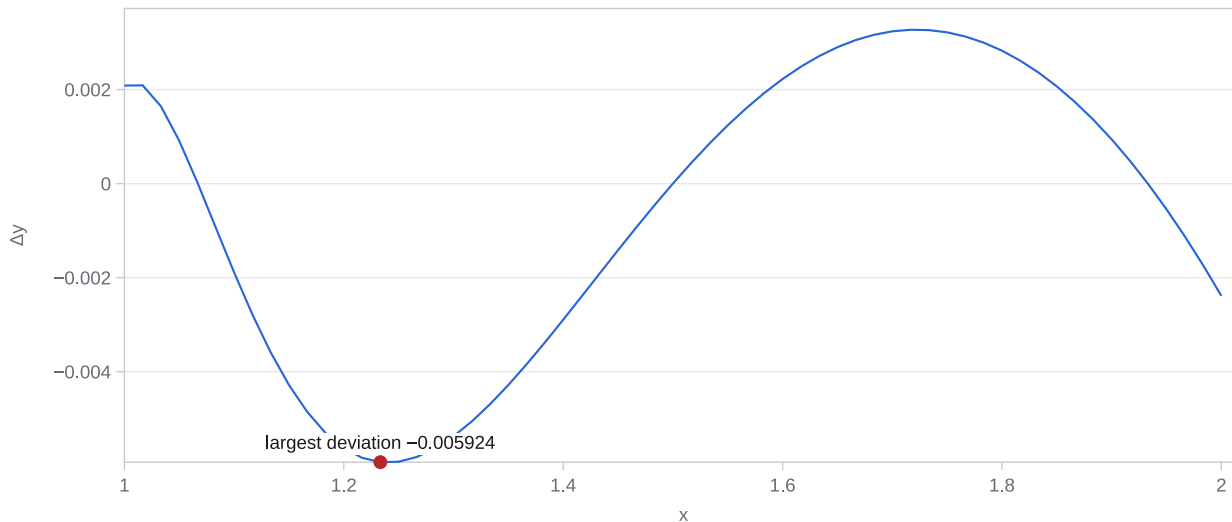
Angle over time [°]



Angular velocity over time [°/s]



Structural error Δy over the input variable x



STRUCTURAL ERROR $\Delta y(x)$: Plotted is the deviation of the generated transfer function from the desired one, over the input variable x . The three zero crossings are the precision points — there the linkage satisfies the assignment exactly, which is their definition and not a statement about quality. What the linkage is worth is decided between them: the marker names the largest deviation with its sign and its location. Where the curve is missing, the chain does not close there — the input angle lies beyond a limit position and the linkage cannot pass between two precision points; how many sampling points are affected is stated in the list of results. A straight line across the gap would claim values that do not exist and is therefore not drawn.

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 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.

HOW THIS LINKAGE WAS DESIGNED — Freudenstein function synthesis: What is prescribed here is an ASSIGNMENT, namely which output angle ψ is to belong to which input angle φ . For a four-bar with A_0 at the origin and B_0 at $(l_1 \mid 0)$, the closed vector loop yields the Freudenstein equation $K_1 \cdot \cos \psi - K_2 \cdot \cos \varphi + K_3 = \cos(\varphi - \psi)$ with the three dimensionless coefficients $K_1 = l_1/l_2$, $K_2 = l_1/l_4$ and $K_3 = (l_2^2 - l_3^2 + l_4^2 + l_1^2)/(2 \cdot l_2 \cdot l_4)$. The equation is LINEAR in K_1 , K_2 and K_3 — that is the whole trick: three prescribed angle pairs give three linear equations, the 3×3 system is solved once, and the lengths follow from the coefficients as $l_2 = l_1/K_1$, $l_4 = l_1/K_2$ and $l_3 = \sqrt{(l_2^2 + l_4^2 + l_1^2) - 2 \cdot l_2 \cdot l_4 \cdot K_3}$. Nothing is searched for and nothing is iterated. Dimensional check: all three coefficients are dimensionless, and only squared lengths appear under the root. THE FRAME IS A SCALE, NOT A REQUIREMENT: the synthesis determines the linkage only up to a factor, so it yields three RATIOS; only the frame length entered turns them into millimetres, and doubling the frame doubles all four lengths. If a coefficient comes out negative that is not an error but means 180° : the link points the other way, its magnitude is in the link length and the rotation in the row “reference direction ... offset by”. WHERE THE PRECISION POINTS LIE is decided, in “target function” mode, by the Chebyshev spacing $x_k = a + h \cdot \cos((2k-1) \cdot \pi/(2n))$ with a the mid-range and h the half-range. It keeps the error between them small and is explicitly NOT an optimal design — Hartenberg and Denavit write immediately after the derivation that one must not expect an optimal design from the Chebyshev distribution; the smallest spacing scheme can only be found iteratively. THE MOST IMPORTANT RESERVATION: the assignment holds exactly in three positions and nowhere else. How far it deviates in between is what the structural error says, and it depends on the function and the range, not on the method: Freudenstein's 1958 table generates $y = x^2$ over $0 \dots 1$ with a structural error below 0.075 % of the output swing, and the same function over $-1 \dots +1$ with 4.47 % — sixty times as much, because it is symmetric there and a four-bar is not suited to symmetric functions. Sources of the method: Freudenstein, Approximate synthesis of four-bar linkages (1955) for the equation; Hartenberg/Denavit, Kinematic Synthesis of Linkages (1964), section 5-5 for the Chebyshev distribution and section 10-3 for the structural error; Norton, Design of Machinery, for the counting convention used here with A_0 at the origin. The test case $y = 1/x^2$ over $1 \leq x \leq 2$ comes from a University of New Mexico course handout, the test case $y = 1/x$ with the printed structural error $E = 0.015$ from Hartenberg/Denavit, example 2 of section 10-3.

HOW THE CHECK IS CARRIED OUT: that the coefficients satisfy the three angle pairs does NOT yet mean that the built linkage actually reaches those three positions — it only means that it COULD occupy them. Four link lengths give two assemblies for every input angle, and a linkage switches between them only in a dead centre. If a precision point lies on the other branch it is

unreachable without taking the linkage apart; the computation notices nothing, because the Freudenstein equation knows both branches. The check is therefore made on the driven motion: the linkage is assembled from its own dimensions, turned through the input range, and measured in every position for whether the prescribed position is hit, whether it lies on the same branch and within one revolution, and whether the order is right. The finding is stated as a sentence in the list of results and names the cause rather than merely a verdict. THIS IS NOT A FORMALITY: two textbooks print, for the same task — $y = \sin x$ over 0 to 90° — results whose linkages do not reach their own third precision point; there the driven branch yields 72.8° instead of the required 57.0°. Neither source carries out this check.

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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