

Kinematic Synthesis of a Curve-Scribing Mechanism for Prescribed Finite Motion

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1 Introduction

The main use of this mechanism is as a substitute for hinges at distant or inaccessible spots. Well known applications are the guide of:

- car hoods [2];
- doors in furniture (see Figure 1).

This paper presents for such mechanism a kinematic synthesis procedure for a prescribed motion of the output link.

In particular, after an analysis of the finite motion properties, a series of design-charts are prepared.

2 Mechanism description

The mechanism is composed of four binary links and two ternary links. In the classical arrangement, is made of two parallelogram four-bar linkages A_0B_0ED and $ABCD$. With reference to the Nomenclature introduced in Figure 2, a distinctive feature of this mechanism is that $AB = CD = DE = A_0B_0$ and $BC = AD = A_0D = B_0E$. By applying the Aronhold-Kennedy theorem and taking into account the previous equalities, it can be demonstrated that is invariant the position of pole C_{16} of the relative motion of output link 6 w.r.t. the frame.



Figure 1: The curve scribing mechanism as a door hinge

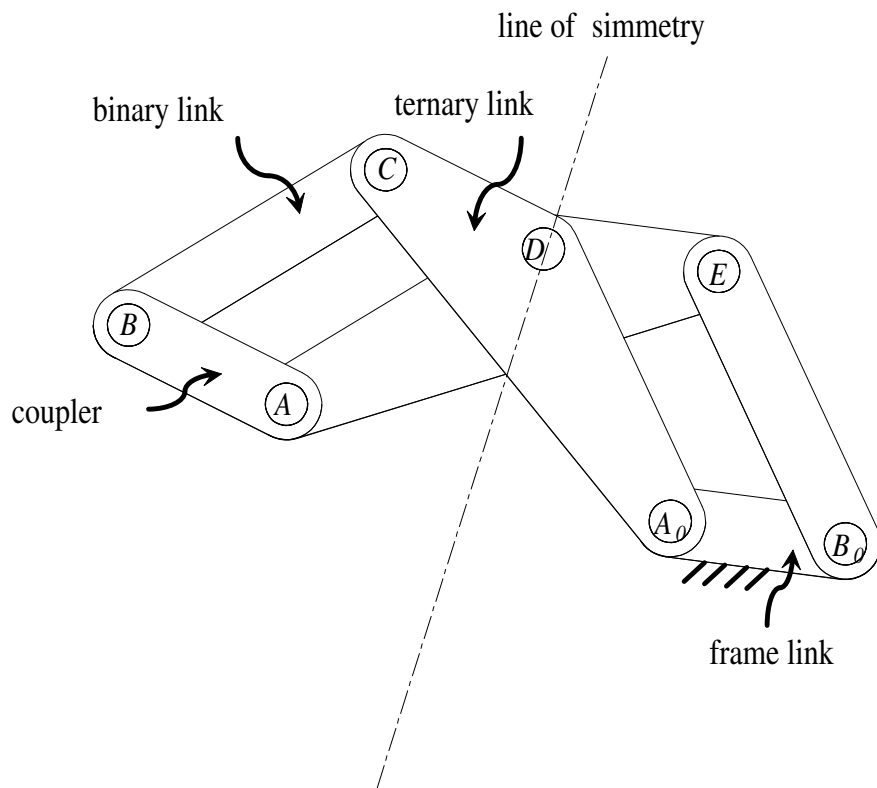


Figure 2: Curve scribing mechanism: Nomenclature

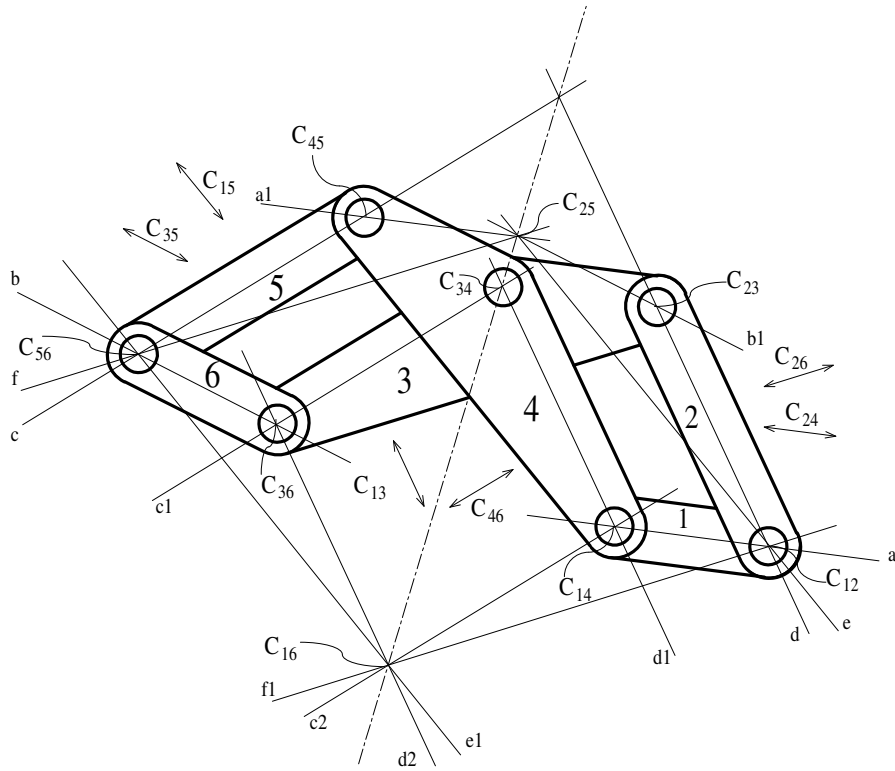


Figure 3: Relative motion poles

It can be also observed that links 3 and 5 translate only.

3 Kinematic design equations

Let us introduce, as shown in Figure 4, a fixed reference system $O-XY$, with the origin at the initial position of point A . The motion of the output link is specified by means of the following parameters: Δx , Δy and $\Delta\theta$, where:

- Δx and Δy are the coordinates of the final position of A ;
- $\Delta\theta = \theta_{in} - \theta_{fin}$ is the absolute rotation of the link 6.

In this treatment, clockwise rotations are considered with positive algebraic sign.

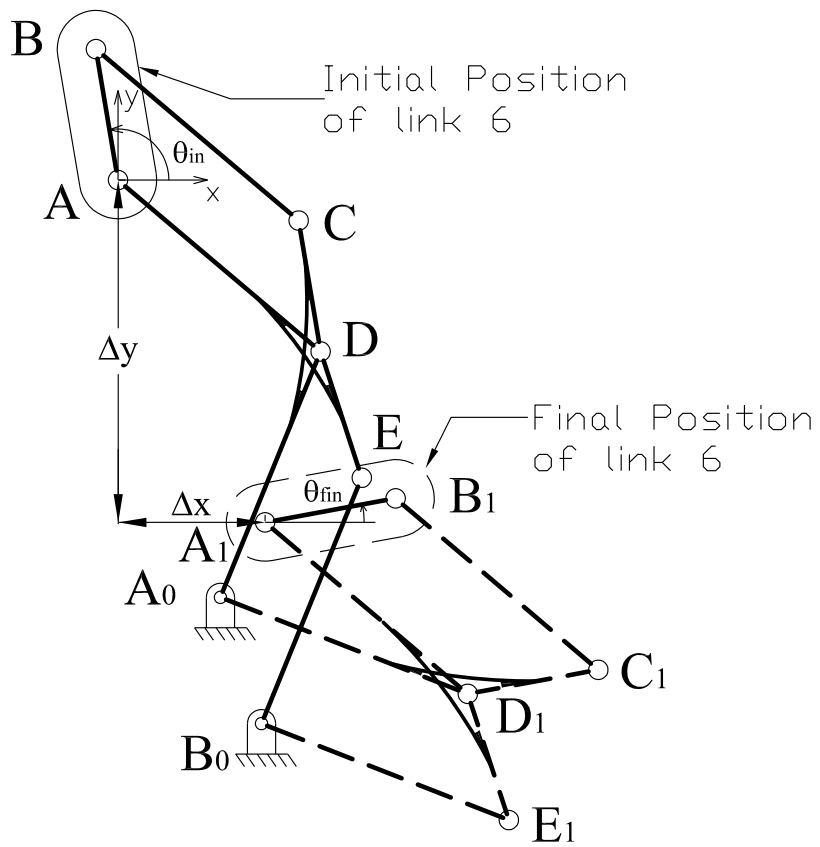


Figure 4: Initial and final configurations of the mechanism

From the geometry of Figure 4, the following equations are deduced:

$$AA_1 = \sqrt{\Delta x^2 + \Delta y^2} , \quad (1)$$

$$B_0E = \frac{AA_1}{2 \sin \frac{\Delta\theta}{2}} , \quad (2)$$

$$\widehat{ADE} = \frac{3}{2}\pi - \tan^{-1} \frac{|\Delta y|}{|\Delta x|} - \theta_{fin} - \frac{\Delta\theta}{2} . \quad (3)$$

Moreover, for the definition of the initial configuration of the mechanism, the following equations are needed:

$$\widehat{BAD} = \pi - \widehat{ADE} + 2\widehat{AC}_{16}D , \quad (4)$$

where the angle $\widehat{AC}_{16}D$ is usually prescribed.

4 Mechanical advantage analysis

Moving link 2 and output link 6 are subjected to torque M and force F , respectively. This force is applied at point P (middle point of segment AB) and is always parallel to the velocity of P .

The mechanical advantage τ is defined as

$$\tau = \left| \frac{F}{M} \right| = \left| \frac{\omega}{v_P} \right| , \quad (5)$$

where v_P and ω are the velocities of point P and of the driving link, respectively. It can be seen that

$$v_P = \omega \cdot PC_{16} . \quad (6)$$

For the purpose of computing τ , it is useful to consider the following relations:

$$\vec{AC}_{16} = \vec{A_0D} = \vec{B_0E} , \quad (7a)$$

$$\vec{AB} = \vec{CD} . \quad (7b)$$

Taking into account the above equalities, it follows

$$\widehat{BAC}_{16} = \widehat{CDA_0} = \widehat{ADE} , \quad (8)$$

$$v_A = \omega \cdot AC_{16} , \quad (9)$$

where v_A denotes the velocity of point A .

From (5) and (6) one obtains

$$\tau = \frac{2}{AB \cdot \sqrt{4k^2 - 4k \cos \widehat{ADE} + 1}} , \quad (10)$$

where

$$k = \frac{B_0E}{AB} . \quad (11)$$

5 Synthesis procedure

The synthesis procedure is simplified by the use of design charts.

In particular, the following charts are available:

- Distance AA_1 .vs. final position of point A (coordinates Δx and Δy) (see Figure 5);
- Length of binary link .vs. distance AA_1 and rotation $\Delta\theta$ (see Figure 6);
- Fixed angle \widehat{ADE} .vs. $\tan^{-1} \frac{\Delta y}{\Delta x}$ (see Figure 7);
- Mechanical advantage τ .vs. adimensional ratio k (see Figure 8).

The sequence of use depends on the data and requirements available.

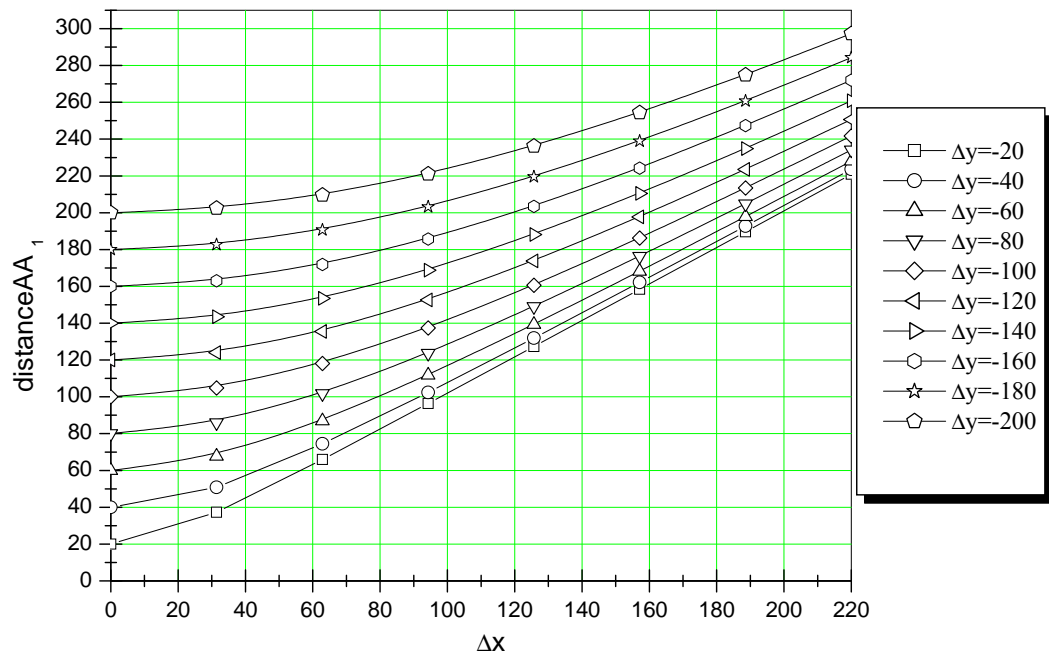


Figure 5: Design chart n.1

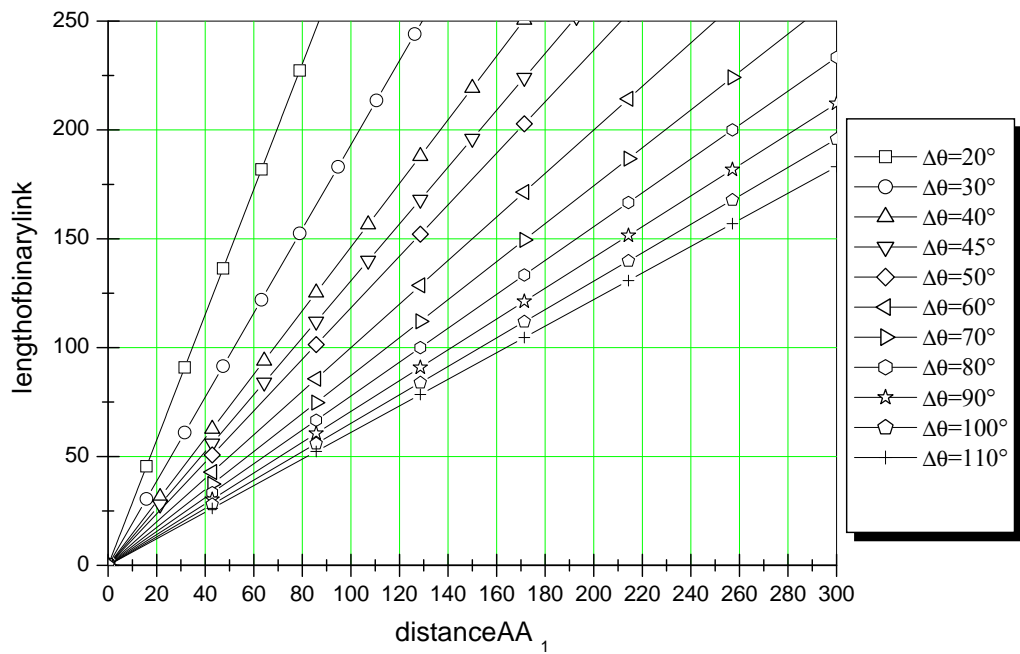


Figure 6: Design chart n.2

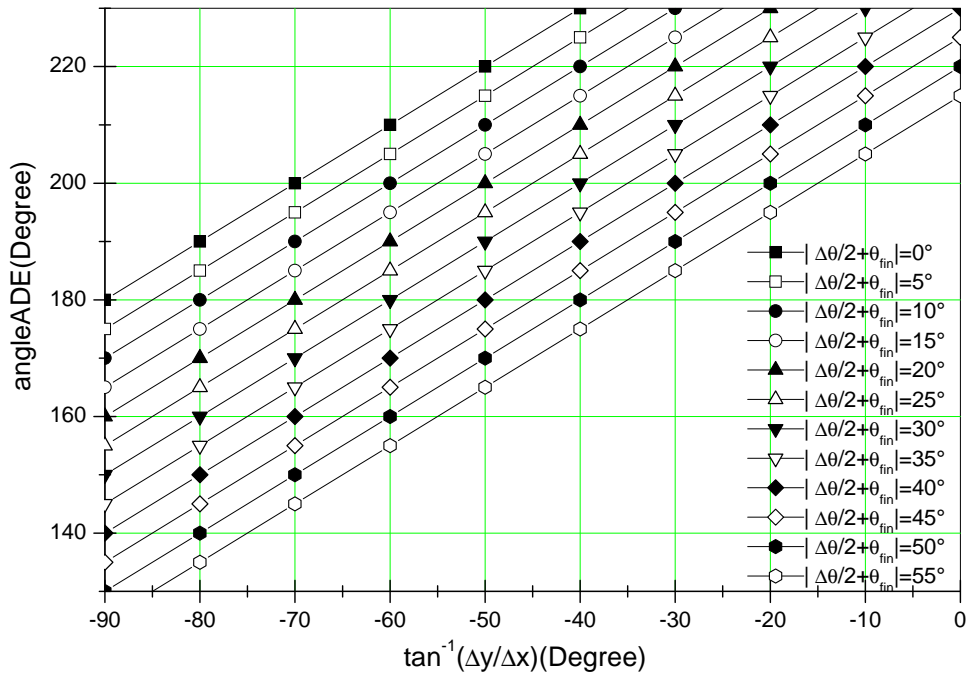


Figure 7: Design chart n.3

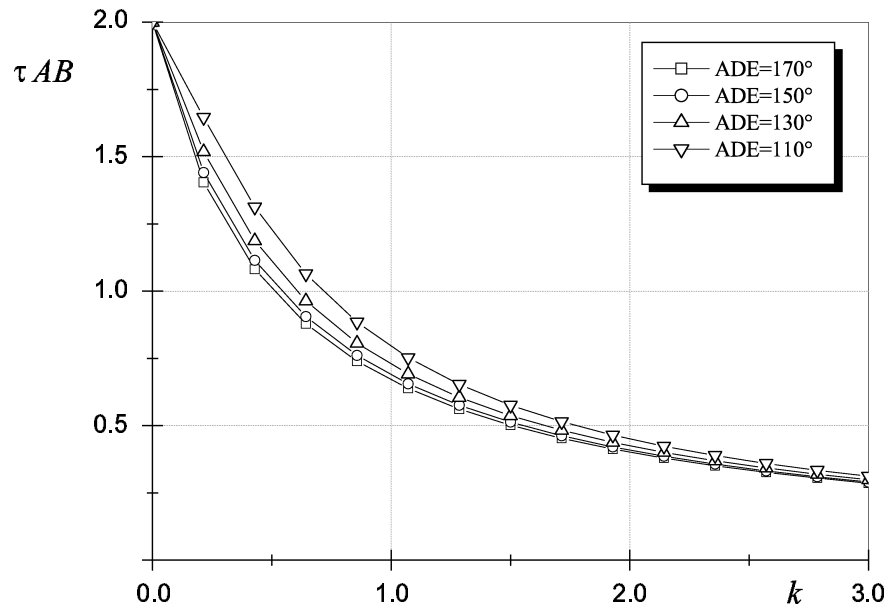


Figure 8: Design chart n.4

For instance, let us assume that Δx , Δy , $\Delta\theta$ and τ are given, *i.e.* the finite motion of link 6 and the mechanical advantage are prescribed. Then, from charts 1, 2, 3 and 4 one can define the length of the binary links, the value of angle \widehat{ADE} , and the dimension of the coupler link 6.

6 Numerical Example

Let us assume:

- $\Delta x=100$, $\Delta y=-120$, $\Delta\theta=95^\circ$, $\theta_{fin}=4^\circ$;
- $\tau \cdot AB$: See Table 1.;

From the design charts reported in figures 5, 6, 7 and 8 one respectively obtains:

- Length of the binary link = 105.9;
- $AA_1 = 156.2$;
- Angle $\widehat{ADE}=168.3^\circ$;
- k and coupler length: see Table 1

Table 1: Numerical results

$\tau \cdot AB$	k	Coupler length
1.00	0.50	211.8
0.75	0.84	126.1
0.50	1.51	70.1

7 Conclusions

A kinematic synthesis procedure for the curve scribing mechanism has been developed. With this procedure one can design this mechanism for a prescribed finite motion of the coupler and a controlled value of the mechanical advantage.

References

- [1] S. Crocesi, *Kinematics of mechanisms for the configurability of car interior* (in italian), Tesi di Laurea, Università di Roma Tor Vergata, Academic year 2000-2001.
- [2] N.P. Chironis, N. Sclater, *Mechanism and Mechanical Devices Sourcebook*, 2nd ed., 1996, McGraw Hill Book Company, p.132.