

GEOMETRICAL SYNTHESIS OF MECHANISMS USED FOR MANOEUVRING OF WINDOWS ON CAR SIDE DOORS

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Abstract: The paper presents the method of the geometrical and kinematical synthesis of the mechanisms which are used for opening / closing of the windows the automobile side doors. For geometrical synthesis, more kinematic diagrams of four-bar linkages are considered. As special mechanisms, there are synthesized the Chebyshev's mechanism, the Evans' mechanism and parallel-crank mechanism.

Keywords: Geometrical synthesis, linkage mechanism, side door, cylindrical gear.

1. INTRODUCTION

For opening and closing the windows from the side doors of cars there are used the so-called „cranes” of side windows [1], these being usually driven with a crank, but additionally with a d. c. electrical motor. In most cases, these planar mechanisms are implemented with type bars and gears links [4], but there are also variants with cable or elastic rack [3]. These mechanisms are mounted at the lower side of motor-car doors [1, 3].

The technical name „door cranes” is used for these mechanisms that drive the windows, that is raise or lower than. The window has got a variable displacement and can move on a vertical plane, being as they are meant guided by a frame that is fixed in the metallic door's frame. The displacement of the windows by a crank is achieved. This crank is mounted inside the wall of the front and back - side doors.

We shall now present a method for geometrical synthesis [2, 4] of the main types of bar and gear links used for opening and closing the windows from the car doors.

2. THE CYLINDRICAL GEAR AND ROCKER MECHANISM

The mechanism is composed (fig. 1) of gear 1 (articulated in point O at the basis) as the driving link, the gear 2 (articulated in A_0 at the basis) which is solid with the rocker $2'$, the roller 3 articulated in A at the rocker, the slide guide in vertical direction C and C' through the translation joint with the basis O) as the driven link.

The kinematic diagram of the mechanism is drawn in the two extreme bottom and top positions of the slide 4, which defines the stroke h for the displacement of the window on a vertical direction (fig. 1). This displacement of slide 4 (together with the window) is achieved by the rotation of rocker $2'$ with the angle β , so that the vertical between points A_1 (bottom) and A_2 (top) corresponds to the window stroke h : $A_1A_2 = h_{12} = h$.

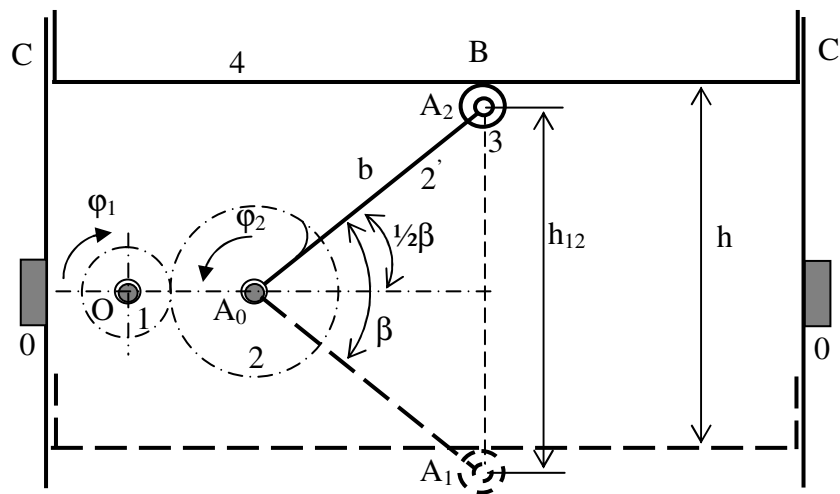


Fig. 1. Kinematic schema of cylindrical gear and rocker mechanism (limit extreme positions)

The optimum position of the rocker in the two limit situations (of the imposed maximum stroke) corresponds to the case (fig. 2.1) when the angle bisector $\angle(A_1A_0A_2) = \beta$ is horizontal (perpendicular to the fixed guide in C and C'). If the stroke h is imposed and the rocker rotation angle β is chosen, the length b of the latter is calculated by the formula:

$$b = \frac{h}{2 \sin \frac{1}{2} \beta} \quad (1)$$

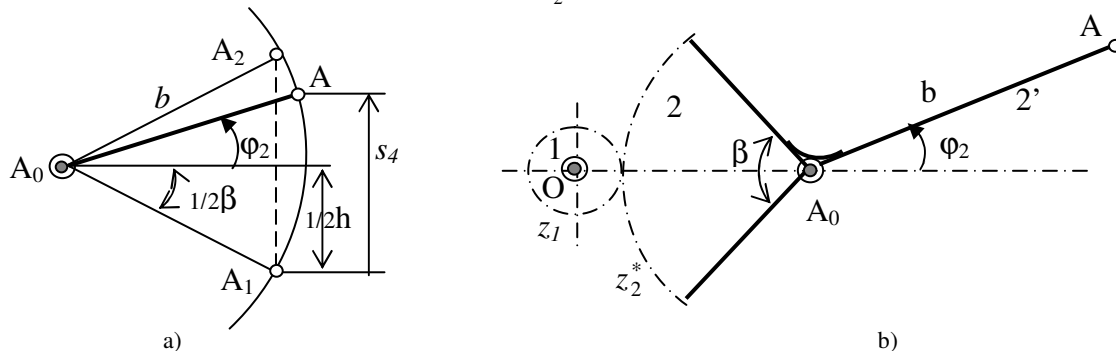


Fig. 2. Sketch for calculus of rocker length b (a)
The kinematic schema of gear with gear sector (b)

The instantaneous displacement s_4 of the slide 4 is obtained (fig. 2a) function of the rotation angle φ_2 of the rocker solid with the gear 2, from the formula below:

$$s_4 = \frac{h}{2} + b \cdot \sin \varphi_2 \quad (2)$$

The rotation angle φ_2 of the rocker (fig. 2a) is deduced function of the rotation angle of the gear 1 (fig. 1) and the transmission ratio of gearing:

$$\varphi_2 = \varphi_1 \cdot i_{21} \quad (3)$$

where $i_{21} = -z_1 / z_2$ corresponds to spur gearing (fig. 1).

The optimum value of angle β is 90° , situation in which the following relation is obtained from formula (2), for the length of rocker 2': $b = 0.7 \cdot h$.

If chosen $\beta = 90^\circ$, the gear 2 can be achieved as a gear sector (fig. 2b) which is solid with the bar 2'.

The teeth number z_2^* of gear sector 2 (fig. 2b) is calculated with formula:

$$z_2^* = z_2 \cdot \beta / 2\pi \quad (4)$$

For the chosen value $\beta = \pi/2$, $z_2^* = z_2/4$ is deduced from formula (6). The value of transmission ratio $i_{12} = z_2/z_1$ between gear 1 and gear sector 2 should be higher for the necessary force for driving to be as low as possible.

3. THE GEARING AND FOUR-BAR MECHANISMS

3.1. Chebyshev's four-bar mechanism

The kinematic diagram of the four-bar mechanism corresponds to the mechanism type λ (fig. 3). The path of one point C , which belongs to the connecting rod 2, is approximate by a straight line on section $C_I C_{II}$.

The characteristic dimensions of the crank and rocker mechanism are so such correlated that: the base length $A_0 B_0$ is the double length of crank $A_0 A$, while the connecting rod, the rocker, and the extension of the connecting rod have got the same length, being equal with two and half of the crank length.

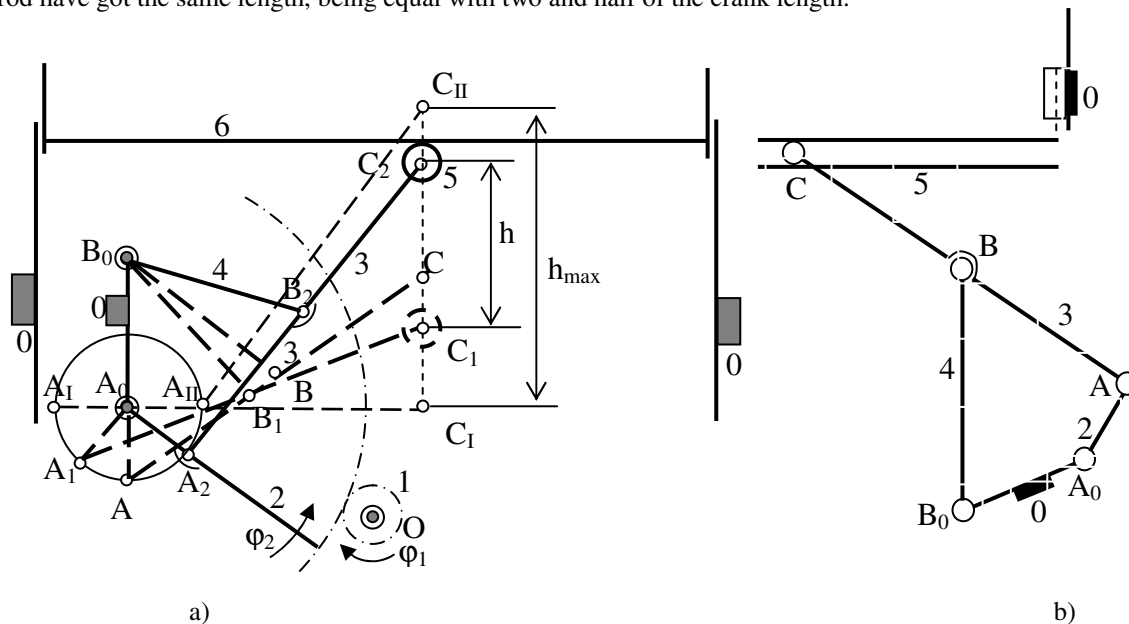


Fig. 3. Kinematic schema of Chebyshev's mechanism in five successive positions (a) and other position of this mechanism (b)

This involves the following the geometrical relation (fig. 3):

$$A_0 B_0 = 2A_0 A; \quad AB = B_0 B = BC = 2.5A_0 A \quad (5)$$

The four bar mechanism is driven through a spur cylindrical gearing, by which the driven gear 2 is achieved as a gear sector, which is solid with the crank $A_0 A$ of the four bar mechanism.

For the gear sector we can choose the angle $\beta = 90^\circ$ (fig. 3), if the stroke of point C is $C_I C_{II} = h$; or an angle, with the maximum value $\beta = 180^\circ$, when the maximum stroke of point C is achieved between points C_I and C_{II} , namely $C_I C_{II} = h_{max} = 4.A_0 A$.

The synthesis of the Chebyshev mechanism (fig. 3) is achieved, starting with stroke h of slide 6 on a vertical direction, which approximately coincides with distance C_1C_2 . In order to determine the characteristic dimensions of the four bar mechanism there are analyzed two possible cases:

- Crank rotation 2 (A_0A) with the angle $\varphi_2 = 90^\circ$ (fig. 3), between positions A_I and A_2 ;
- Crank rotation 2 (A_0A) with the angle $\varphi_2 = 180^\circ$, between positions A_I și A_{II} (fig. 3).

For the first case, the crank length of four bar mechanism is $A_0A = h/2$, and for the second case, the crank length $A_0A = h/4$ is calculated.

The other characteristic linear dimensions are calculated with relations (7). The position of the fixed side A_0B_0 is parallel to the fixed guide, thus this direction is vertical in the considered situation (fig. 3).

The first the synthesis variant is recommended, when the gear sector corresponds to one quarter of circle. The position of driving gear 1 is chosen in the right side of the rank, while the transmission ratio for gearing is chosen $i_{12} = 5...7$. The high values of this reduction ratio ensure the self-blocking of mechanism when the mechanism is drive from the window.

3.2. Evans' four-bar mechanism

The kinematic diagram of this mechanism (fig. 4) shows an articulate quadrilateral, similar with the type λ , whose characteristic point C is situated on the extension of the connecting rod, but at a greater distance than in the case of Chebyshev's mechanism (fig. 3).

The base A_0B_0 of the four bar mechanism is inclined at an angle $\alpha = 30^\circ$ (fig. 4) in relation with the direction movement of slide - frame 6, in which window will be moved.

The characteristic lengths of Evans' mechanism are selected function of the length A_0A of the crank 2:

$$A_0B_0 = 2.3 \cdot A_0A; \quad AB = 1.92 \cdot A_0A; \quad B_0B = 2.0 \cdot A_0A; \quad BC = 2.92 \cdot A_0A \quad (6)$$

For a rotation of the crank 2 with the angle $\angle(A_1A_0A_2) = 90^\circ$ in the direct sense (fig. 4), the point C moves on vertical direction with distance $C_1C_2 = h$.

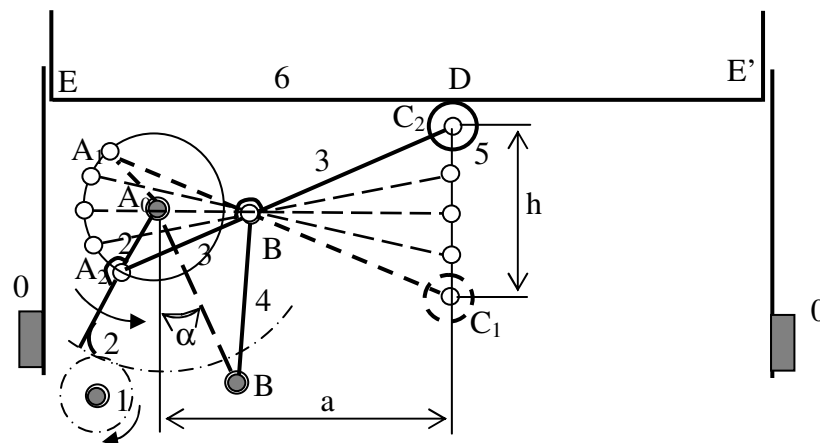


Fig. 4. Kinematic schema of Evans' four-bar mechanism
(The extremes positions of the mechanism with thick line are drawn).

By the roller 5, which is articulated at connecting rod 3 in point C, the slide 6 (window support) is moved on the direction of the vertical guide. The driving of the mechanism is achieved with a spur cylindrical gear (1, 2), on which gear 1 is a driving the pinion and gear 2 which is achieved as a gear sector solid with crank 2. The synthesis of the four bar mechanism is achieved function of the stroke h , which is imposed on slide 6, that corresponds to the maximum opening of the window. There are analyzed two cases for the geometric synthesis:

- Gear sector 2 with angle $\beta = 90^\circ$ (fig.4), for which $A_0A = h/2$;

b) Gear sector 2 with angle $\beta = 180^\circ$, in this case crank 2 has the length $A_0A = h/3.5$.

c)

In each case above the crank length A_0A function of the stroke h is deduce, after which other characteristic lengths of mechanism is calculated with the following relations:

$$\begin{aligned} a) \quad A_0B_0 &= 1.15 \cdot h; \quad AB = 0.96 \cdot h; & b) \quad A_0B_0 &= 0.657 \cdot h; \quad AB = 0.549 \cdot h; \\ B_0B &= h; \quad BC = 1.46 \cdot h; & B_0B &= 0.571 \cdot h; \quad BC = 0.834 \cdot h. \end{aligned} \quad (7,8)$$

The positioning of the fixed point B_0 is achieved on a horizontal line, in relation with line C_1C_2 , at the distance $2.87 \cdot A_0A$ or $1.435 \cdot h$ (for the first case) and $0.83 \cdot h$ (for the second case) respectively.

Point A_0 is placed on the median of segment C_1C_2 on the left side (fig. 4) at a distance $a = 2.01 \cdot h$ (first case), $a = 1.158$ (second case) respectively.

The connection in point D, between roller 5 and slide 6, should be considered in the central zone of the sliding frame between points E and E' (fig. 4). This is necessary to avoid blocking of window during the raising stroke, because of the reaction danger in the fixed side guides of E and E' .

3.3. Parallel-crank mechanism

There can be considered two variants of the parallel crank mechanism, one in which the basis is the shorter side (fig. 5a), and the other in which the basis is the longer side of the articulate parallelogram (fig. 5b). For the first variant is shown the constructive design (fig. 5c).

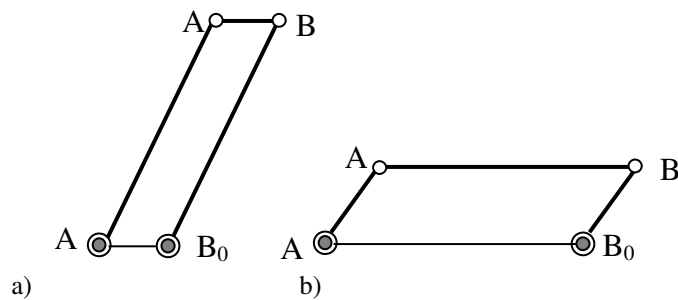


Fig. 5. Kinematic schema (a, b) of an articulated parallelogram

3.3.1. The first variant (fig. 5a) is used in two ways:

- The basis A_0B_0 perpendicular to the slide guide, in which the window is fixed (fig. 6a);
- The basis A_0B_0 parallel to the slide guide of the window (fig. 6b, 5c).

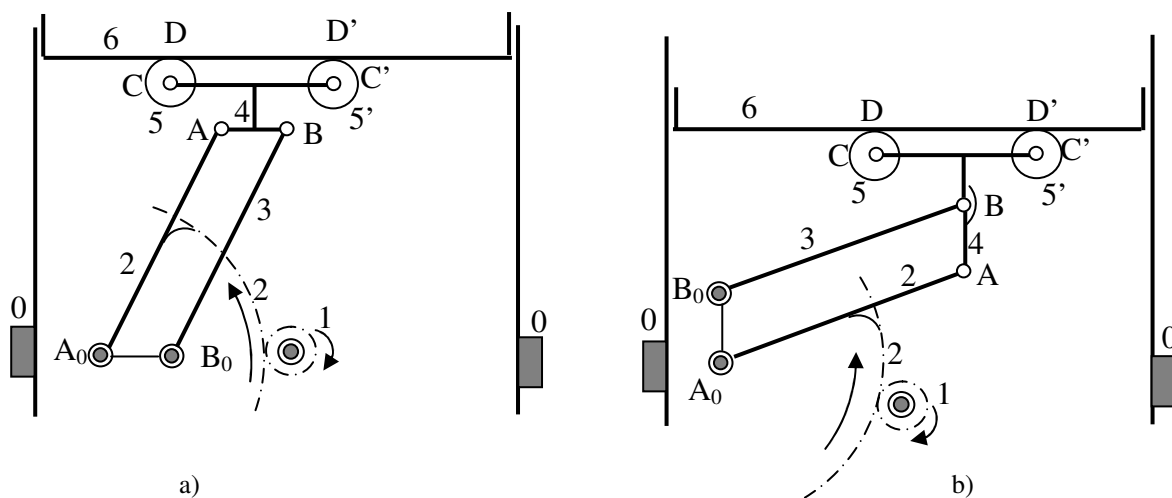


Fig. 6. Kinematic schema of the articulated parallelogram in a serial connect on the pinion – gear sector gearing

The first way (fig. 6a) uses the circle translation movement of the connecting rod 4; at which there are mounted rollers 5 and 5' in points C and C' , thus as $CC' \parallel AB$.

The second way (fig. 6b) uses the same circle translation movement of the connecting rod 4, but the points C and C' are located on each side of the connecting rod 4, thus $CC' \perp AB$.

The contact of rollers 5 and 5' with slide 6 is achieved in points D and D' , thus a better guidance is allowed for the window between these two fixed parallel guides. For parallel crank mechanism, the driving is achieved with a spur cylindrical gearing (fig. 6.6a,b) or a annulus cylindrical gear. The constructive gear 2 is achieved as a gear sector, with angle $\beta = 90^\circ \dots 120^\circ$.

In function of the imposed stroke h for slide 6 and of the chosen angle β there are calculated the length bars 2 and 3 ($A_0A = B_0B$) with formula (2). The basis and connecting rod lengths ($A_0B_0 = AB$) are chosen function of the lengths of rockers 2 and 3 with relation $A_0B_0 = AB = (0,15 \dots 0,2) \cdot A_0A$. The mechanism can be hand drive from the pinion gear 1 or mechanically with an electric motor and a reducer mechanism of a worm – worm wheel type.

3.3.2. The second variant (fig. 7) is achieved through the positioning of the parallel crank mechanism in the lower side of basis A_0B_0 and the extension of the two rockers A_0A and B_0B in the above side where they are jointed through the second connecting rod $CD \parallel AB$ (fig. 7).

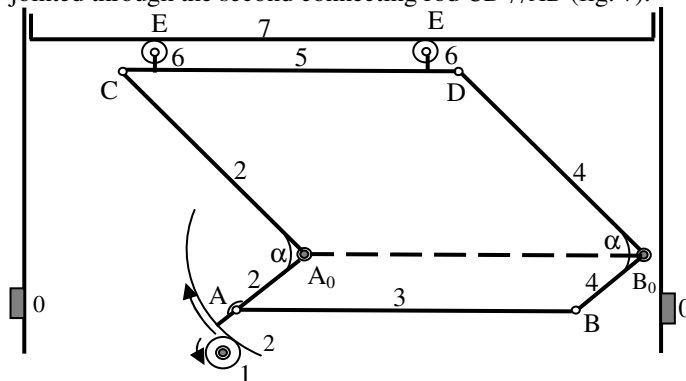


Fig. 7. Kinematic schema of double parallelogram mechanism and pinion gear - gear sector gearing.

The parallelogram mechanism A_0ABB_0 below is composed of bars 2, 3 and 4, having fixed articulations in points A_0 și B_0 . The two long sides (A_0B_0 and AB) of the articulated parallelogram (fig. 6.7a) have a horizontal position, the direction being perpendicular to the window guide. The bars 2' and 4' are solid with bars 2 and 4 and these are longer in length ($A_0C > A_0A$, $B_0D > B_0B$). Thus, the angle between bars 2 and 2', 4 and 4' respectively is a constant angle α_4 , this value is chosen $\alpha_4 \leq 90^\circ$. The ratios between the long and short sides of the two parallelograms are (fig. 7):

$$\frac{AB}{A_0A} = 3 \dots 5; \quad \frac{CD}{A_0C} = 1 \dots 1,5 \quad (9)$$

The above parallelogram A_0CDB_0 (fig. 7) has the same basis A_0B_0 as the one below, while the bars are 2', 4' and 5, the last having the rollers 6 and 6' mounted on the connecting rod.

On the two rollers 6 and 6' stays (in points E and E') slide 7, which has an imposed translation movement between the fixed vertical guides (fig. 7). The contact with slide 7 in two points E and E' is realized, while the distance EE' will be long enough to allow the quiet displacement of the window between the two guides and thus avoid blocking. The double parallelogram mechanism is driven through a spur cylindrical gearing, where the gear 2 is achieved as a gear sector with a center angle $\beta = 90^\circ$.

For the gearing pinion (1) and gear sector (2), the numbers of teeth is chosen like below:

$$z_1 = 6 \dots 12; \quad z_2^* = 35 \dots 51 \quad (10)$$

It is recommended that this gearing should have a module of 1,5 mm or 2 mm, because the thickness of teeth is small, namely 2...5 mm. Because the pinion gear 1 is achieved with a teeth number under 17, this implies a positive displacement for the profile [2]. A constructive variant of Chebishev's mechanism (fig. 3b) with a cylindrical gear is shown in fig. 8.

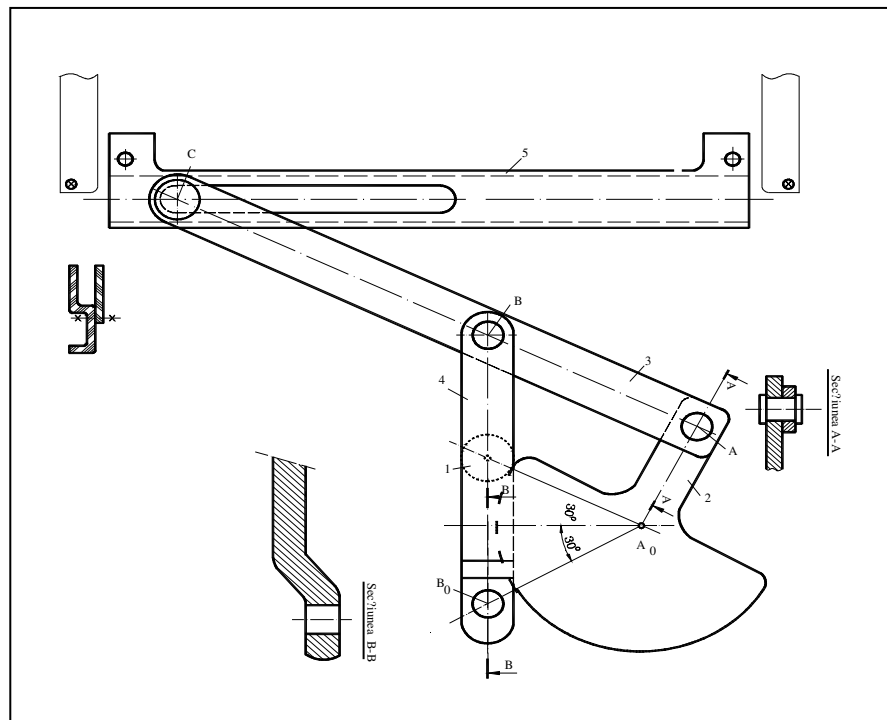


Fig. 8. Constructive design of mechanism with cylindrical gear and Chebishev linkage

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