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ARTICULATED BAR MECHANISM DESIGN FOR MANIPULATING THE DOORS WINDOWS OF ROAD VEHICLES

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Abstract: With the development of car manufacturing industry, besides increasing their technical performances, respectively the development of engines, transmissions, car body aerodynamics, electronic systems, or elements of active and passive safety, car manufacturers insist increasingly more the driver and passenger comfort, respectively ergonomics of the interior. In this sense, this paper presents synthesis principles of plane mechanisms based on extreme positions. As case study is presented the mechanism synthesis for driving windows of the doors road vehicles, considered a double crank-slide mechanism, which were determined lengths of kinematic elements, respectively speed driving element based on design original size. At the same time were determined also the kinematic parameters of the kinematic chain for driving mechanism.
Keywords: mechanism, extreme position, trajectory

1. INTRODUCTION

In mechanisms and machines theory, synthesis aims to determining the geometrical and kinematics parameters based on quantities required by the design theme. Theme design can be extremely complex, depending on the movement that must execute the mechanism to be done. Thus a complete project involves the following steps, [3,7]:

- ✓ Establishing optimal kinematic scheme in order to achieve driven element law of motion
- ✓ Parameters determination for motion transmitting from leader element to the driving element
- ✓ Determining the mechanism geometric parameters in order to achieve some transformation form of movement from leader element to the driving element (crank - rocker, crank - rotated coulisse, crank - oscillatory coulisse, rocking double, double crank, etc.)
- ✓ Determining the extreme positions in conditions to avoid maximum pressure angles imposed by technological process
- ✓ Establishing mechanism assembly to avoid exceeding the gauge fixed by design theme.

Geometric synthesis problems from mechanisms theory can be put primarily in terms of achieving the transformation of a movement (driving element) in a imposed move by the type (motion), of driven element. From this point of view can be up to 16 types of mechanisms, which are combinations of continuous rectilinear motion, alternative straight motion, continuous circular motion, or alternative circular motion.

Another range of problem is the condition imposed one of the mechanism elements, such a portion of its plane to pass through the certain positions imposed by the technological process (2,3,4, or 5 positions). Also the technological process may require that a point driven element to travel in a given cycle time kinematic certain trajectory, [3, 7].

Depending on the initial conditions imposed by the design theme, and considering the foregoing, the designer will have to choose one of the 16 variants of mechanisms.

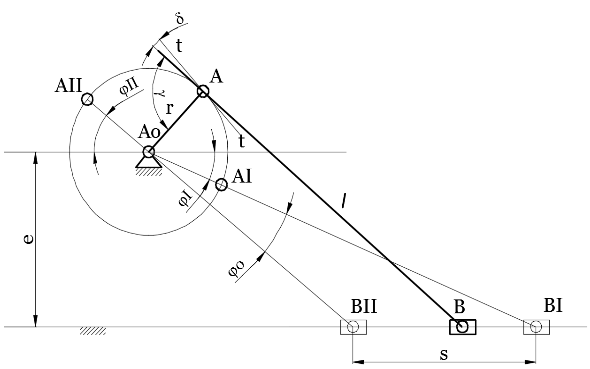


Figure 1. Mechanism geometric and kinematic quantities

2. PLAN MECHANISMS DESIGN UNDER EXTREME POSITIONS

Because the drive mechanism of windows can be considered a dual mechanism crank - slide (Figure 3), which has a limited course in car's door space, further will be presented the theoretical aspects of synthesis of these mechanisms types, based on the extreme positions. In the following will be specified in the case of eccentric slide crank mechanism (Figure 1) geometric and kinematic elements that allow choosing three of them (independent parameters), his design, [3, 7].

Therefore we will have the following dimensions:

- ✓ crank driver radius, r
- ✓ connecting rod length, l
- ✓ eccentricity, e
- ✓ the ratio between the crank radius and connecting rod length $\lambda = \frac{r}{l}$
- ✓ the ratio between the eccentricity and the length of the connecting rod $\nu = \frac{e}{l}$
- ✓ the angle that makes between them the two extreme positions of the crank and connecting rod that are in the same direction $\varphi_o = \varphi_{II} - \varphi_I$. From Figure 1 it follows:

$$\varphi_0 = \varphi_{II} - \varphi_I = \arcsin\left(\frac{e}{l-r}\right) - \arcsin\left(\frac{e}{l+r}\right) \tag{1}$$

- ✓ productivity coefficient determined by the ratio of average speeds of the slide on the return stroke ($B_{II}B_I$) and duction (B_IB_{II}) $k = \frac{\pi+\varphi_0}{\pi-\varphi_0}$
- ✓ slider stroke (piston) $s = B_IB_{II}$ which can be expressed by the equation:

$$s = l\cdot\sqrt{(1+\lambda)^2 - \nu^2} - l\cdot\sqrt{(1-\lambda)^2 - \nu^2} \tag{2}$$

- ✓ pressure angle, determined by the speed of a point A (Figure 1) and the force applied, in this case in the direction of the connecting rod AB, δ
- ✓ transmission angle is the complement of the pressure angle, $\gamma = \frac{\pi}{2} - \delta$

Angles δ or γ limited fields in that can operate a mechanism without danger of locking.

From the parameters specify above can choose three, independent between them, who allow the construction of a cinematic scheme of a slider crank mechanism. These parameters are imposed either by design theme, either dimension or mounting conditions. For this purpose the following design possibilities exist for the piston crank mechanism (notations in Figure 1):

- ✓ designing when are imposed: r, l, e
- ✓ designing when are imposed: s, k, λ
- ✓ designing when are imposed: s, k, r
- ✓ designing when are imposed: s, k, l
- ✓ designing when are imposed: s, k, e
- ✓ designing when are imposed: s, r, e

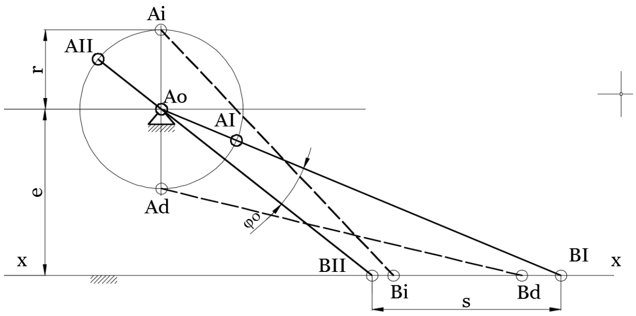


Figure 2. Building cinematic scheme

The following provides slide crank mechanism designing by imposing sizes r, l, e, the most convenient option for the synthesis of the drive mechanism of the glass. Building cinematic scheme is achieved as shown in Figure 2, in the following steps:

- ✓ xx line drawn - piston movement direction
- ✓ on distance e from xx line is chosed a fixed rotation center A0 of the crank and plot the circle with radius $A_0A = r$
- ✓ from the point A_0 as center and ray l+r and l-r is drawn circle arcs that intersect xx line in B_I, respectively B_{II} (the two extreme positions of the piston)
- ✓ then are determined all other elements of the mechanism

3. WINDOW DRIVE MECHANISM SYNTHESIS

Window drive mechanism synthesis on motor vehicles is based on the following design data:

- ✓ glass lifting stroke – h = 510 mm
- ✓ glass lifting/lowering time – t = 10s

As shown above, and considering that the driven element, glass port rail, must perform a vertical translation motion, the mechanism is a plan one, with linked bars, and rotational kinematic couplings, respectively translation. The scheme proposed for drive mechanism is shown in figure 3. The mechanism consists of two articulated bars (1.2) crossed by the rotational couple E, having equal lengths, and related on glass port rail (3) by translational coupling, to ensure its vertical displacement.

As kinematic scheme the drive mechanism is similar with a mechanism consists of two slider crank mechanisms, articulated to one another in E couple. For the geometrical synthesis can be choosing the option where are imposed sizes e, r and l, as follows (notations are shown in Figure 3):

- ✓ d = 510/2 = 255 mm (half of the stroke up / down) - is considered joint fixed on the same horizontal with track (fixed) guide direction, respectively on glass halfway.
- ✓ a = b = 340 mm - from size and mounting conditions (available space in the vehicle door)

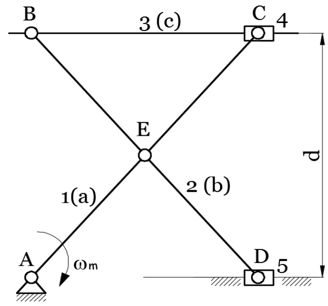


Figure 3. Mechanism kinematic scheme

Based on the chosen geometric dimensions (imposed by the design theme, respectively by the size and mounting conditions), drive mechanism kinematic scheme builds graphically as follows:

- ✓ It represents fixed guide direction, convenient (Figure 4)
- ✓ On this direction is chosen conveniently fixed joint position A
- ✓ For both sides (symmetric) is represent the extreme positions of the rail port glass (3 - as shown in Figure 3) at distances of 255 mm, Figure 4.
- ✓ At the intersection of the vertical direction by a fixed joint A and corresponding directions of chair rail port extreme positions are represented the corresponding positions of revolute pair B
- ✓ The two extreme positions of the slide C is obtained by the intersection of the circle arc, having its center at a fixed joint A and radius a = 340 mm (the length of main arm) and the two directions corresponding to the two extreme positions of the glass support rails (Figure 5)

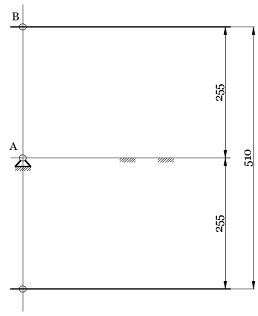
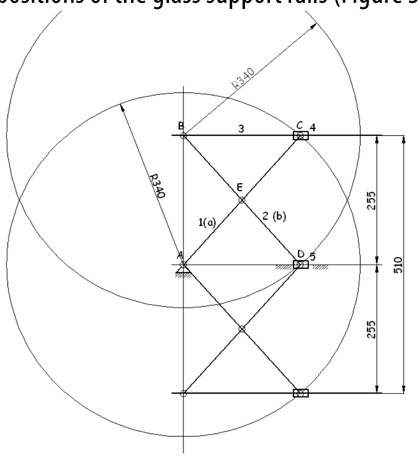


Figure 4. Fixed guide direction



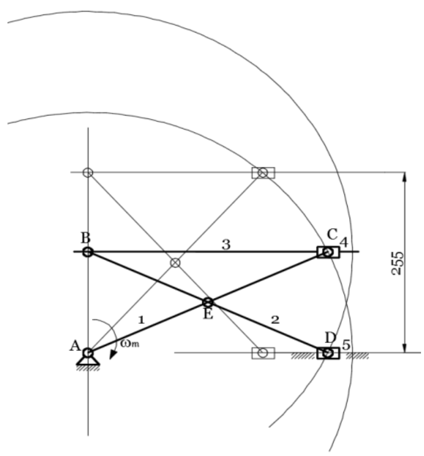


Figure 5. the two extreme positions of the glass support rails **Figure 6**. Mechanism in the intermediate position

- ✓ The two extreme positions of the slide D are obtained by the intersection of the circular arc with center in rotation kinematic couple B and radius b = 340 mm (secondary arm length) and fixed guide direction (Figure 5)
- ✓ A and C couplers positions are connected together, respectively B and D
- ✓ At the intersection of AC and BD kinematic elements is represented kinematic couple E

Kinematic scheme of the mechanism in an intermediate position, other than the ones extreme, corresponding to open and closed positions of the window, is shown in Figure 6.

- ✓ on C couple trajectory (circle arc with center in A and radius 340) is chosen an intermediate position
- ✓ It represents couple position B, as intersection between the trajectory and the corresponding position of chair rail port
- ✓ It represents couple position D, as intersection between the fixed guide direction and the circular arc with center in B and radius 340
- ✓ It represents couple position E, as intersection between the kinematic elements AC and BD

The construction of Figure 2.5 represents the two extreme positions of the mechanism by measuring the angle between the two extreme positions of the main arm (1) result the rotation angle of its in order to achieve the lowering / raising 510 mm course (Figure 7).

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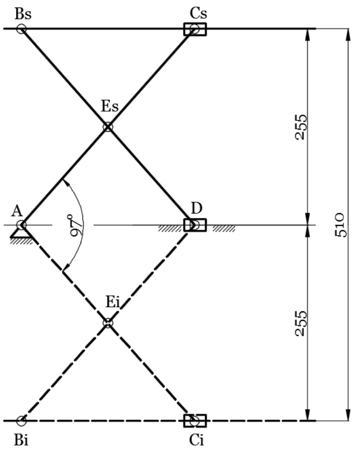

Figure 7. Mechanism extreme positions

As shown in Figure 7, is necessary main arm one rotation by an angle of 97 degrees to achieve lowering / raising 510 mm course. Knowing the time value of up / down (t = 10s imposed by theme design) may be determinated angular velocity (3), respectively main arm speed (4) (driving element).

$$\omega_m = \frac{\varphi}{t} = \frac{97 \cdot \frac{\pi}{180}}{10} = 0{,}17 \text{ rad/s} \tag{3}$$

$$n_m = \frac{30\omega}{\pi} = \frac{30 \cdot 0{,}17}{\pi} = 1{,}6 \text{ rot/min} \tag{4}$$

4. DRIVE KINEMATIC CHAIN PARAMETERS OF MECHANISM

For the case of electrical actuation of the window drive mechanism is proposed a kinematic chain consisting of an electric motor and a gear wheel transmission in two steps consisting of a worm and a spur gear, which form a gear motor assembly.

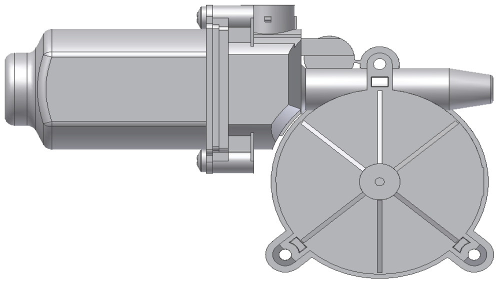

Figure 8. Drive electric motor

For the drive kinematic chain synthesis was chose a DC electric motor, 12V (Figure 8) with power P = 0,3 kW and speed $n_M = 1500$ r / min:

Through design theme the engine torque is limited to M = 2 Nm. Checking the engine power is done with relation (5)

$$M = 9550\frac{P}{n_m} \Rightarrow 9550\frac{0{,}3}{1500} = 1{,}91\text{Nm} \tag{5}$$

Kinematic chain consisting of worm - worm wheel gear (Figure 9) and spur gear (Figure 10) is to reduce the speed of the electric motor to the main arm, obtained by calculation above.

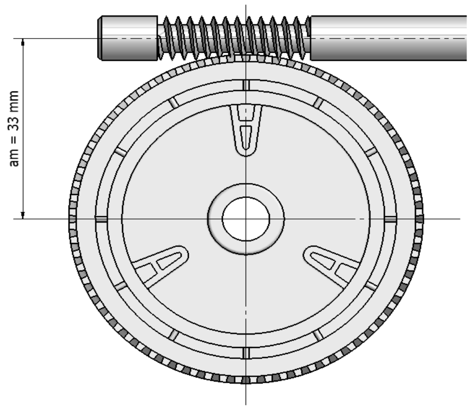

Figure 9. Worm gear

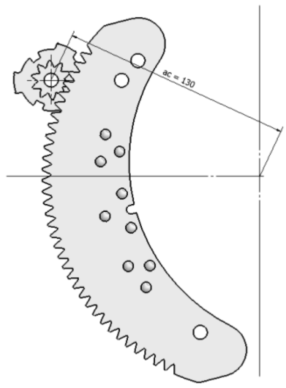

Figure 10. Spur gear

The total transmission ratio of the two steps gear is calculated with the relation (6), [9]:

$$i = \frac{n_{Motor}}{n_{manivela}} = \frac{1500}{1{,}6} = 937{,}5 \tag{6}$$

Partial transmission ratios for the two steps gear is adopted constructively as follows:

- ✓ worm gear transmission ratio: $i_m = 72$
- ✓ cylindrical gear transmission ratio: $i_c = 13$

Effectively gear ratio is calculated as a multiplication of the two partial reports (7), and the error from the real gear ratio is obtained by equation (8):

$$i_{ef} = i_m \cdot i_c = 72 \cdot 13 = 936 \tag{7}$$

$$\varepsilon = \frac{i_{ef} - i}{i_{ef}} = \frac{|936 - 937{,}5|}{936} \cdot 100 = 0{,}16\% < 3\% \tag{8}$$

The relative error of effectively ratio from the real ratio fall within the allowable limits (max 3%), so it can accept the values of partial transmission reports.

The distances between the axis of the two steps gear were imposed from dimension and mounting conditions (Figure 9, Figure 10) for the worm step: l = 33 mm, respectively cylindrical step ac = 130 mm.

5. CONCLUSION

Synthesis of the drive mechanism of the glass doors road vehicles was made by graphic - analytic methods. Graphic constructions were made in AutoCAD, which gives high precision of the results.

REFERENCES

[1.] Antonescu, P; Petrescu, R; and Antonescu, O: Synthesis of Complex Mechanisms Used for Opening/Closing of Windows on Automotive Side Doors, International Conference on Mechanisms and Mechanical Transmissions, Cluj Napoca, 15-22, 2004

[2.] Antonescu, O and Dugăeşescu, I: Mechanisms Used for Driving Windows of Automotive Side Doors, International Conference on Mechanisms and Mechanical Transmissions, Cluj Napoca, 9-14, 2004

[3.] Artobolevski, I: Teoria mecanismelor şi a maşinilor, Editura Tehnică, Bucureşti, 1955

[4.] Deliu, Gh; Cotoros, D and Popescu, R: Analysis and synthesis of a plane linkage used in industrial applications, Simpozionul național cu participare internațională PRASIC' 02, 99 - 102, 2002

[5.] Grecu, B; Adîr, G; Adîr, V and Adîr V.G: Structural design of the fundamental plane kinematic chains, Proceedings of the 10th IFToMM International Symposium SYROM, Brașov, 129 – 136, 2009

[6.] Kovacs, Fr; Perju, D and Savii G: Metode noi în sinteza mecanismelor, Editura Facla, Timișoara, 1976

[7.] Manolescu, N; Kovacs, Fr and Orănescu, A: Teoria mecanismelor şi a maşinilor, Editura Didactică şi Pedagogică, Bucureşti, 1972.

[8.] Miklos, I; Cioată, V. G; and Miklos C: Grafică tehnică asistată de calculator, Editura Pim Iași, 2011.

[9.] Miklos, Zs: Mecanisme. Analiza mecanismelor, Editura Mirton, Timişoara. 2005

[10.] Miklos, I; Miklos, C and Alic, C: Drive mechanism analysis of door windows to road vehicles, Acta Tehnica Corviniensis – Bulletin of Engineering, Tome VII, Fascicule 4, 61-64, 2014

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