ANNALS of Faculty Engineering Hunedoara
– International Journal of Engineering
Tome XV [2017] – Fascicule 3 [August]

ISSN: 1584-2665 [print; online]
ISSN: 1584-2673 [CD-Rom; online]
a free-access multidisciplinary publication
of the Faculty of Engineering Hunedoara

1.Imre Zsolt MIKLOS, 2.Cristina Carmen MIKLOS, 3.Carmen Inge ALIC

SYNTHESIS OF SEAT POSITIONING MECHANISM TO VERTICAL ON ROAD VEHICLES

1-3. University Politehnica Timişoara, Faculty of Engineering Hunedoara, ROMANIA

Abstract: Current trends in the automotive industry have as object besides increasing their performance also development aspects related to the interior ergonomy. So vehicle interior design should provide passengers the conditions that create them actually believe that in the car, have everything they need to move with maximum comfort and safety. Seats and bench seats must be comfortable to ensure passengers positions as free and restful can be. They must be provided with adjustable systems in horizontal and of the elevation plan, with a geometry adapted to the morphology of the human body and lumbar spine. In this sense the present paper presenting the geometric synthesis based on the theorem on the crank existence, the mechanism of vertical adjustment of the driver's seat in motor vehicles. After synthesis, in order to fulfill initial requirements was obtained an articulated parallelogram mechanism, for which were determined cinematic elements lengths either from graphical constructions either constructive from terms of graphics or gauging assembly. At the same time they were determined also cinematic parameters of the powertrain for driving mechanism.
Keywords: mechanism, extreme position, crank, rod, rocker

1. INTRODUCTION

In mechanisms and machines theory synthesis aims at determining geometric and kinematic parameters imposed by the design theme.

The design theme can be extremely complex, depending on the movement on mechanism must realize.

A complete project involves the following steps:

- Establishing the cinematic scheme for optimal law of motion of the driven element.
- Determining variation diagrams of the forces because it based on them to achieve constructive form of mechanism, able to perform under optimal conditions imposed process (motion).

In the two phases, the issues that the designer must solve are quite numerous also complex at the same time:

- Determining element for transmitting element leader motion to the driving element
- Determining the geometric parameters of the mechanism to achieve some motion transformation form from the element leader to the driven element (crank - rocker, crank - rotating coulisse, crank - oscillating coulisse, double rocker, double crank, etc.)
- Determining the positions of extreme in conditions of avoiding maximum pressure angles, imposed by technological process
- Establishing mechanism assembly to avoid exceeding the size set by the design theme.

Geometric synthesis problems from mechanisms theory can be put first in terms of achieving the transformation of a movement (leader element) in a motion type imposed by the driven element process (movement). From this point of view can be achieved 16 mechanisms variants, shown in Table 1 [6].

- A further range of problems is a condition imposed on one of the mechanism elements, like a portion of its plane to pass through specific positions imposed by technological process (2,3,4, or 5 positions)
- Finally technological process may require like an point from driven element to travel during cinematic cycle a specific trajectory.

Depending on the initial conditions imposed by the design theme, and considering the foregoing, the designer will have to choose one of the mechanisms options (classes) presented in Table 1.

Table 1. The mechanisms options (classes)

Class	Leader element motion	Driven element motion	Example
1	Continues rectilinear motion	Continues rectilinear	Levers and articulated bars
2		Back and forth rectilinear	Hydraulic wheel
3		Circular continues	Winch, pulley, lifting jack, gear rack
4		Back and forth circular	Pendulum
5	Back and forth rectilinear motion	Continues rectilinear	Pithead with lifts
6		Back and forth rectilinear	Levers
7		Circular continues	Piston-crank mechanism
8		Back and forth circular	Oscillating coulisse mechanisms
9	Circular continues motion	Continues rectilinear	Rolling mills
10		Back and forth rectilinear	Crank-piston mechanism
11		Circular continues	Transmissions with flexible element
12		Back and forth circular	Crank-rocker mechanism
13	Back and forth circular motion	Continues rectilinear	Pendulum
14		Back and forth rectilinear	Oscillating coulisse mechanisms
15		Circular continues	Crank-rocker mechanism
16		Back and forth circular	Double rocker mechanism

2. THEORETICAL ISSUES RELATED TO PLANE MECHANISMS SYNTHESIS

Mathematical tool showed that, for example, an imposed trajectory of a point from the articulated plan mechanism connecting rod plane equals in the most general case of a curve 6th degree (curved rod), with its many variants, solving the problem being always related to graphics. That is why the first synthetic methods have been graphs. Along with the development of computer technology (hardware, software) appeared inevitably also analytical methods, that did not do and do not make currently any other than to confirm and complete the precision of graphic calculation result.

This is why the first step in the synthesis of articulated plan mechanisms is solving a geometry problem of articulated plan mechanism, respectively to establish the crank existence condition according Grashof theorem [6]. Since the positioning mechanism of the road vehicles seat is a four-bar linkage mechanism, all aspects presented below will be referring to this type of mechanism.

It puts the following problem: What should be the relationship between the lengths of the elements of a mechanism articulated quadrilateral so that one of two related at base elements perform a continuous rotation motion (crank) while the other articulated at the bottom element (rocker) make an oscillating movement.

Figure 1 is a quadrilateral crank - rocker mechanism in those two extreme positions, $A_0A_IB_IB_0$, $A_0A_{II}B_{II}B_0$.

It is known that the rod AB opposite the fixed element (base A0B0) which has both mobile joints is called the connecting rod (Figure 1) and the other two bars (kinematic elements) articulated at the base and the rod are named crank A0A when executing a rotation of 360 degrees during a kinematic cycle and rocker B0B when in the same cycle executes an oscillation between two extreme positions.

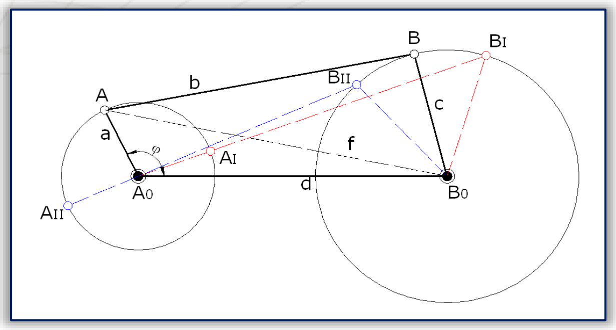

Figure 1. Extreme positions of the articulated quadrilateral mechanism

For this type of mechanism to exist is needed that in any position of the crank A0A - in this case determined by the angle that it makes with the base A0B0 - be possible position construction of joint B. As shown in Figure 1, with the notation $AB_0 = f$; $A_0A = a$; $A_0B_0 = d$; $AB = b$; $B_0B = c$, we have the following relationship:

$$AB_0^2 = f^2 = a^2 + d^2 - 2ad\cos\varphi \tag{1}$$

The existence of joint B is conditional upon inequality:

$$|b - c| \le f \le (b + c) \tag{2}$$

By squaring equation (2) becomes:

$$(b - c)^2 \le f^2 \le (b + c)^2 \tag{3}$$

Substituting the value of f in equation (1) results:

$$a^2 + d^2 - 2ad\cos\varphi \le (b + c)^2 \quad (4)$$

$$a^2 + d^2 - 2ad\cos\varphi \ge (b - c)^2 \quad (5)$$

The highest value of the first member in relation4 has place for cosφ = -1 (φ = 180⁰), and the lowest value of the first member from relation 5, for cosφ = 1 (φ = 0; 360⁰). To limit the two inequalities become:

$$\begin{aligned} a^2 + d^2 + 2ad &\le (b + c)^2 \\ a^2 + d^2 - 2ad &\ge (b - c)^2 \end{aligned} \quad (6)$$

The first inequality can be written as:

$$(a + d)^2 \le (b + c)^2 \quad \text{or} \quad a + d \le b + c \quad (7)$$

The second inequality can be written as:

$$(a - d)^2 - (b - c)^2 \ge 0 \quad \text{or} \quad (a - d - b + c)(a - d + b - c) \ge 0 \quad (8)$$

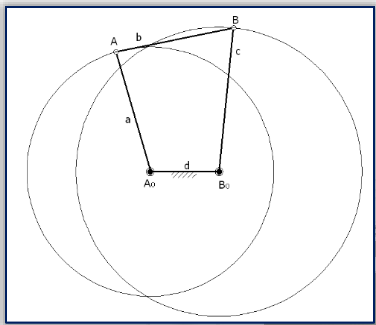

Figure 2. Double crank mechanism

Looking at equation (8) is found that there are two cases:

– **Case I:**

$$\begin{aligned} a - d - b + c &\ge 0 \\ a - d + b - c &\ge 0 \end{aligned} \quad (9)$$

From the inequalities (7) and (9) result:

$$\begin{aligned} a + d &\le b + c \\ b + d &\le a + c \\ c + d &\le a + b \end{aligned} \quad (10)$$

By summing by twos the equations of relation (10), result:

$$d \le a; \quad d \le b; \ d \le c \quad (11)$$

It follows that the length d of the base must be the shortest of all four elements. As the system (10) is symmetrical about the a and c, in case the inequality d <b <c <a is satisfied the rectangle mechanism is double crank (Figure 2).

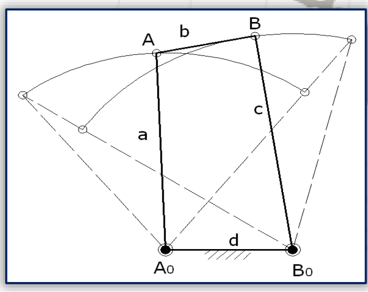

Figure 3. Double rocker mechanism

– **Case II:**

$$\begin{aligned} a - d - b + c &\le 0 \\ a - d + b - c &\le 0 \end{aligned} \quad (12)$$

From the inequalities (7) and (12) results:

$$\begin{aligned} a + d &\le b + c \\ a + c &\le b + d \\ a + b &\le d + c \end{aligned} \quad (13)$$

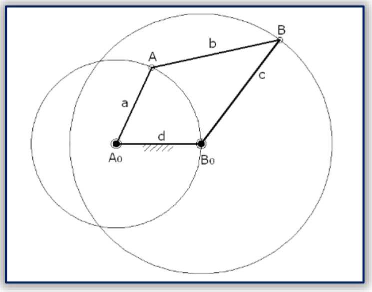

Figure 4. Gallowey mechanism

In this case the length of the element a must be the lowest. As the system is not symmetrical about the a and c, it follows that only the a element is crank, and c is rocker, so articulated quadrilateral mechanism is a crank-rocker one (Figure 1). But if the shortest element is rod, the mechanism is a double rocker (Figure 3).

The following is a particular cases of articulated quadrilateral mechanism-Gallowey quadrilateral mechanism (a = d <b = c), Figure 4.

» crank - rocker symmetrical quadrilateral mechanism (a = b < c = c), figure 5
» parallelogram mechanism (a = c ≠ b = d), figure 6.
» antiparallelogram mechanism (a = c ≠ b = d), figure 7.

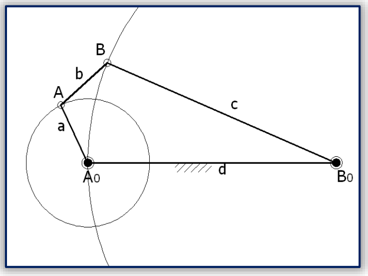

Figure 5. Crank-rocker symmetrical mechanism

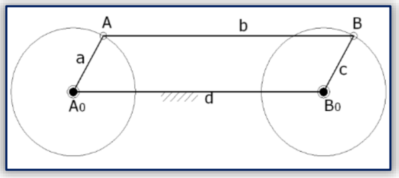

Figure 6. Parallelogram mechanism

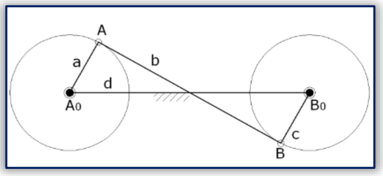

Figure 7. Antiparallelogram mechanism

3. SEAT POSITIONING MECHANISM SYNTHESIS ON ROAD VEHICLES

Vertical positioning mechanism geometric synthesis on driving seat in road vehicles was based on the following original sizes, as provided in project contract documents [9]:

- cycle to getting up the seat – h = 38 mm
- seat lifting time (from the minimum to maximum position)– t = 3,1 s
- the mass of the driver and the mechanism – m = 120 kg
- gravitational acceleration (for own weight of the mechanism) – g = 9,81 m/s2

Taking into account the movement will have to execute the seat in his go up or down motion, mechanism most appropriate to achieve this movement is articulated quadrilateral mechanism. The mechanism for positioning the driver's seat will be composed of a main mechanism (articulated quadrilateral) and a kinematic element, and two passive kinematic couplings. The passive kinematic element will be a connecting rod port seat, without kinematic role, which will be mounted effectively the seat, and that will serve to enhance (increase the rigidity) mechanism.

Because in the up and down movement, the driver's seat mounted on rod port-chair and implicitly proposed articulated quadrilateral mechanism main rod, must not rotate from horizontal (in order to not create change position driver seat during up/down) it follows that the two rods, which obviously will have identical movements, will have alongside positions with each other in every discrete position from kinematic cycle. Such motion for the rod articulated quadrilateral mechanism is provided by a particular case of it, respectively an articulated parallelogram mechanism. Articulated parallelogram mechanism kinematic scheme is shown in Figure 8.

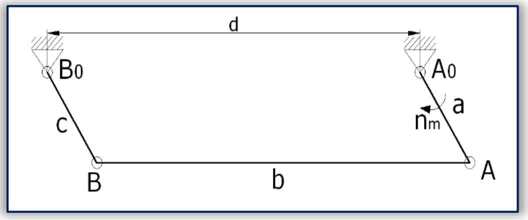

Figure 8. Articulated parallelogram mechanism kinematic scheme

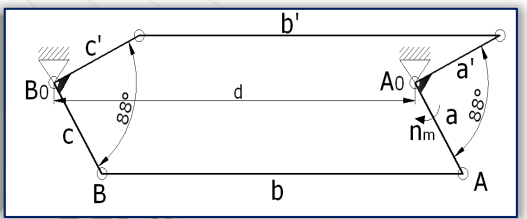

Figure 9. Positioning mechanism kinematic scheme

For reasons of weight and assembling have been adopted in a constructive manner the following geometric dimensions of the basic mechanism. Obviously, for the case a = b ≠ c = d, as shown in Figure 6.

- element length A0A- a = 35 mm
- element length B0B – c = 35 mm
- rod length AB – b = 170 mm
- distance between A0, B0 articulations - d = 170 mm

Based on the geometric dimensions above we are in Grashof's theorem case II, i.e. the crank is the shortest part of the mechanism and taking into account that a = c, the parallelogram mechanism will be a double crank type. Figure 9 shows complete mechanism kinematic scheme for positioning the driver's seat. Both kinematic elements in rotation will be constructive point of view, in the form of crank levers. Also dimensions and installation considerations were taken constructively also other geometric dimensions of positioning mechanism:

- for crank handles: a' = 50 mm; c' = 50 mm
- the length of rod port - chair: b' = b = 170 mm

For reasons of weight and space location (the mechanism must fit under the driver's seat, a height of approx. 100 mm)) the mechanism for positioning the seat has been represented, to scale, in the extreme position corresponding to the position most upper of the seat (figure 10). In this sense by geometric design were obtained following overall dimensions of the mechanism, taking into account the fact that the mechanism will be provided with a drive mechanism consisting of gear transmissions:

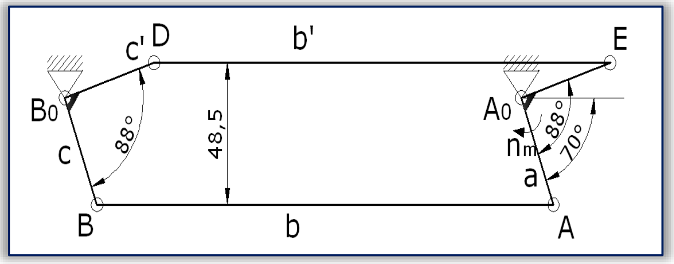

Figure 10. Positioning mechanism in the upper extreme position

- mechanism maximum height (the maximum distance between the two rods)– 48,5 mm
- the position angle of the driving element A0A, corresponding to the positions – 70 degrees

The rotation angle of the driver element (A0A) and hence of the driven element (B0B) was determined under the conditions of the mechanism with kinematic scheme proposed (Figure 10), performs a complete up cycle (or down) in accordance with the design theme, h = 38 mm, as shown in figure 11. As a point mark can be considered any point of the connecting rod port - chair, for example couple D or E.
In conclusion the synthesis of the vertical positioning mechanism for the driver's seat was achieved by the following steps:

- was shown to scale mechanism in the upper extreme position (Figure 10)
- compared to the couple E it was shown a 38 mm straight, corresponding to the minimum extreme position (position of the rod port - seat)
- point E trajectory (a circle) intersects the line at the point E1
- from the geometric conditions adopted (trajectory and geometry bent element) is determined A1 point position
- similar is determined the positions of the points B1 and D1.

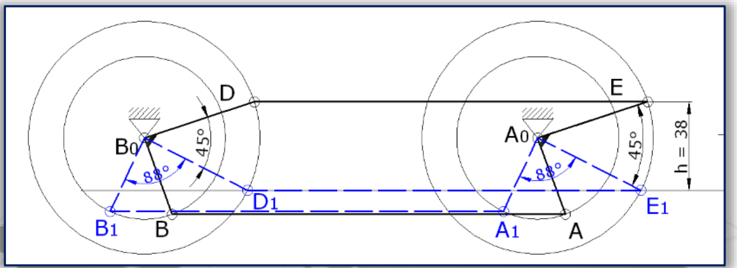

Figure 11. The rotation angle of the driver element AA_0

The mechanism shown in E1A0A1B1B0D1 position (figure 11), is its minimum extreme position. By measuring (dimensioning) the angle between the two extreme positions of the driving element, resulting its rotational angle, necessary to achieve mechanism cycle required by the design theme, respectively - 45 degrees. In this case, driving element will not execute a complete rotating motion, but oscillating one, so the mechanism became a double rocker.
For the design of the drive mechanism, consisting of the transmission gear it is necessary to determine the angular speed or driving element speed, AA0. This was done knowing the following sizes:

- Rotational angle φ = 45 degrees (resulting from the geometrical synthesis)
- Kinematic cycle time - t =3,1 s (imposed by the design theme)
- Angular speed:

$$\omega_m = \frac{\varphi}{t} = \frac{45 \cdot \frac{\pi}{180}}{3,1} = 0,253 \ \text{rad/s} \tag{14}$$

- Speed:

$$n_m = \frac{30\omega}{\pi} = \frac{30 \cdot 0,25}{\pi} = 2,4 \ \text{rot/min} \tag{15}$$

4. CONCLUSION

Synthesis of driver's seat positioning mechanism to vertical on road vehicles has been achieved by graphic - analytical methods.
By geometric synthesis was aimed the establishing type of the mechanism - articulated parallelogram and determining its kinematic elements size, respectively the driver element angular travel in order to achieve imposed race by the design theme. Finally was determined driving element speed necessary drive mechanism design. Graphic constructions were made in AutoCAD, which provides high accuracy results.

References

[1.] Artobolevski, I: Teoria mecanismelor şi a maşinilor, Editura Tehnică, Bucureşti, 1955
[2.] Freudenstein, F: Approximate synthesis of four-bar linkages, Resonance, Volume 15, Issue 8, pp 740–767, 2010
[3.] Guj, G. Z. Dong, Z.Y and Di Giacinto, M: Dimensional synthesis of four bar linkage for function generation with velocity and acceleration constraints, Meccanica, Volume 16, Issue 4, pp 210–219, 1981
[4.] Gupta, K.C: A general theory for synthesizing crank-type four-bar function generators with transmission angle control, Journal of Applied Mechanics Volume 45, Issue 2, pp 415-421, 1978
[5.] Kovacs, F; Perju, D and Savii G: Metode noi în sinteza mecanismelor, Editura Facla, Timișoara, 1076

[6.] Manolescu, N; Kovacs F and Orănescu A: Teoria mecanismelor şi a maşinilor, Editura Didactică şi Pedagogică, Bucureşti, 1972
[7.] Miklos, I.Zs; Cioată, V.G and Miklos C: Grafică tehnică asistată de calculator, Editura Pim Iași, 2011.
[8.] Miklos, I.Zs: Mecanisme. Analiza mecanismelor, Editura Mirton, Timişoara, 2005
[9.] Miklos, I.Zs; Alic, C.I and Miklos C.C: Analysis of seat positioning mechanism in road vehicles, Acta Technica Corvinensis Bulletin of Engineering, Tome III, Issue, 3, pp 27-30, 2010
[10.] Norton, T.W; Midha, A and Howell L.L: Graphical Synthesis for Limit Positions of a Four-Bar Mechanism Using the Triangle Inequality Concept, Journal of Mechanical Design, Volume 116, Issue 4 , pp 1132-1140, 1994
[11.] Todorov, T.S: Synthesis of four-bar mechanisms by Freudenstein–Chebyshev, Mechanism and Machine Theory, Volume 37, Issue 12, pp. 1505–1512, 2002

ANNALS of Faculty Engineering Hunedoara
– International Journal of Engineering