DESIGN OF A CAR DOOR WINDOW REGULATOR

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ABSTRACT

DESIGN OF A CAR DOOR WINDOW REGULATOR

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In this thesis, design of a car door window regulator is presented. This design comprises a mechanism in order that the car door window makes a specified translational motion. First, conceptual design is carried out to obtain the best suitable concept for the design and best suitable concept comes out to be a scissor mechanism. Afterwards, detailed design of the chosen concept is given. In the detail design stage, kinematic synthesis of the mechanism is performed basically using the Cardan motion. Lastly, implementation of the design on a car door is described.

Keywords: Mechanism Synthesis, Scissor Mechanism, Cardan Motion

ÖZ

ARABA KAPISI CAM KRİKOSU TASARIMI

Özsipahi, Mümin
Yüksek Lisans, Makine Mühendisliği Bölümü
Tez Yöneticisi : Prof. Dr. Eres Söylemez

Eylül 2009, 66 sayfa

Bu tezde araba kapısı cam krikosu tasarımı sunulmuştur. Bu tasarım, araba kapısı camının belirtilen öteleme hareketini yapması için bir mekanizma içermektedir. İlk olarak, tasarım için en uygun konsepti elde etmek için kavramsal tasarım gerçekleştirilmiştir ve makas mekanizması en uygun konsept olarak bulunmuştur. Daha sonra seçilen konseptin detaylı tasarımı verilmiştir. Detaylı tasarım aşamasında kinematik sentez temel olarak Cardan hareketi kullanarak gerçekleştirilmiştir. Son olarak tasarımın bir araba kapısına uygulanması anlatılmıştır.

Anahtar kelimeler: Mekanizma Sentezi, Makas Mekanizması, Cardan Hareketi

To love of my life, Hilal

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INTRODUCTION

1.1 Introduction to Window Regulators

One of the standard properties of cars is the capability to open windows of side doors. As a result, the need of a device that opens windows of car doors arose with the invention of cars. This device which is used to raise or lower the window of the car door is called a *window regulator*.

With the growth of the automotive industry, automotive suppliers spend much effort on the design of window regulators. At first, window regulators are used manually for a long time. The user generally opened and closed the window of the door by turning a handle. Later, like other man-powered mechanisms, manual operation is replaced with automatic functioning. Window regulators are powered with electric motors as a result windows are raised and lowered automatically.

No matter what type of a power source is used, window regulators require mechanisms to function. Because, as known mechanisms are basis of all mechanic devices. Hence, different types of mechanisms are utilized resulting different types of window regulators. The reason for using different mechanism types may be cost, operation, assembly, production, etc.

1.2 Literature Survey about Window Regulators

In the literature, lots of patents about different types of window regulators can be found. These window regulators can be grouped into five main headings according to their functioning. These are screw driven window regulators, window regulators utilizing rack and pinion, window regulators comprising lazy tong mechanisms, window regulators using pulleys and window regulators using arms.

Screw driven window regulators use power screws to raise and lower windows. One of the early patents belongs to Kraemer in 1928 [1]. In this design, the window is attached to the nut of the power screw. Power screw is driven with a handle with actuating gears. In Figure 1.1, a describing drawing of the patent is given.

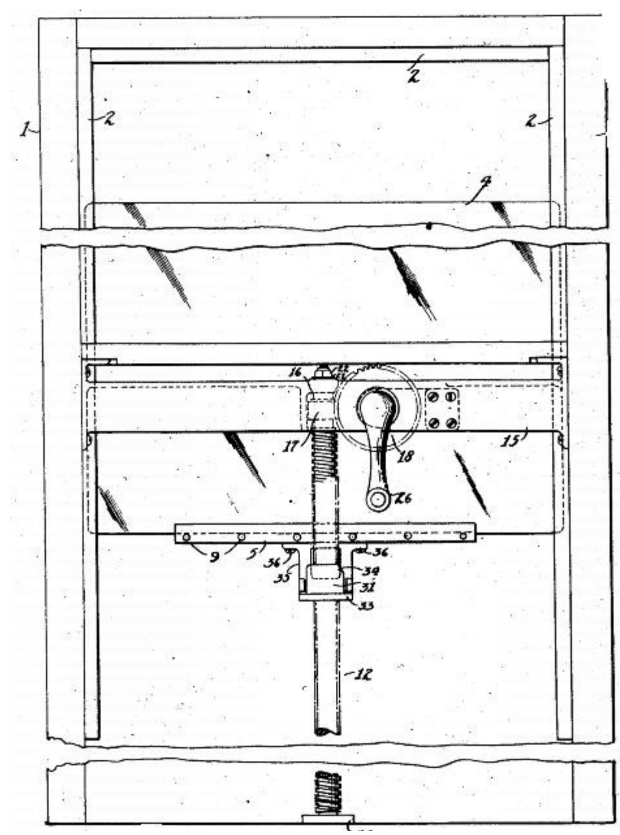


Figure 1.1 An early screw driven window regulator mechanism [1]

Another example of screw driven window regulator mechanism is patented from Szkodzinski in 2006 [2]. In this patent, power screw extends through a radius of curvature of the window. Power screw is connected to hollow axle of the electric motor like a nut. Also, the window is attached to the electric motor, in that manner the window and electric motor go together along the power screw. A drawing from the patent is shown in Figure 1.2.

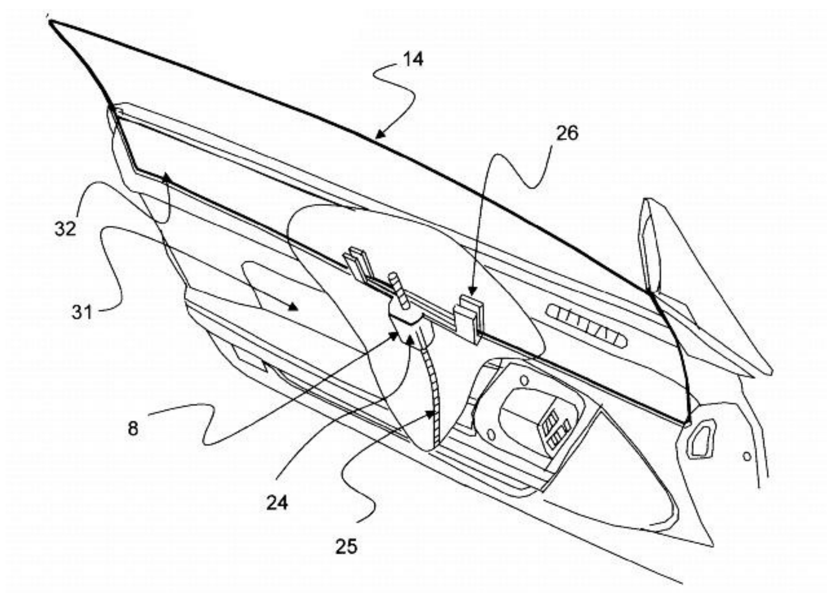


Figure 1.2 Another example of screw driven window regulator mechanism [2]

Another type of window regulator mechanisms employs racks and pinions to provide translation for windows. En early exampe belongs to Bell and Schoenleber patented in 1920 [3]. In this patent, the power from the handle is transmitted to rack and pinion pair trough gears and chains. With rack and pinion, the window is raised and lowered (Figure 1.3).

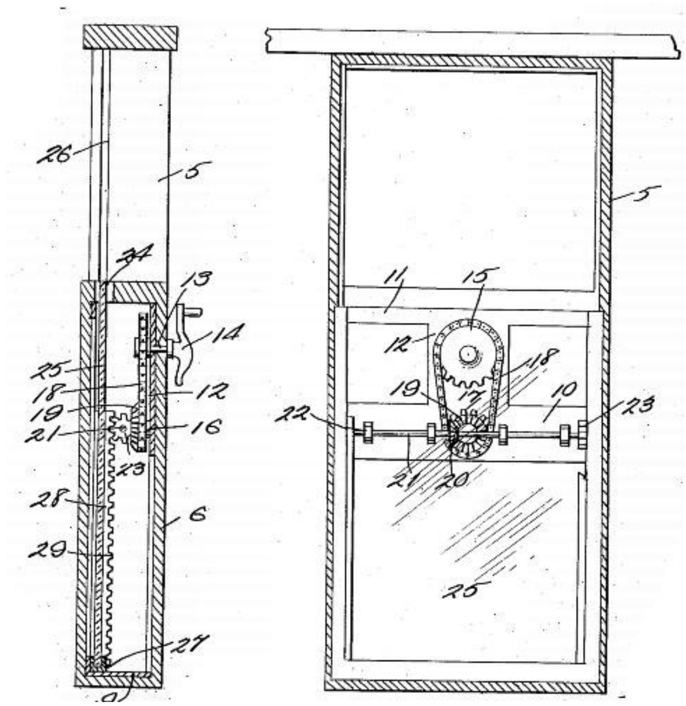


Figure 1.3 A window regulator mechanism utilizing rack and pinion [3]

A different patent belongs to Rietdijk taken in 2008 [4]. In this design, window is attached to the electric motor. Window regulator is operated by an electric motor driving rack and pinion with a gear set.

Another group of window regulator mechanisms utilizes lazy tong mechanisms to obtain translational motion of the window. An early example is patent of Eckey taken in 1914 [5]. In this patent, a handle drives the lazy tong mechanism with gears. Also, window is connected to the tip of lazy tong mechanism, by this way window is raised or lowered through the window frame. A describing drawing of this patent is given in Figure 1.4.

A further example is patent of Walters taken in 1989 [6]. In this design, the window is connected to tip of the lazy tong mechanism. The motion of the window is obtained by a lazy tong mechanism which is driven by an electric motor with a power screw.

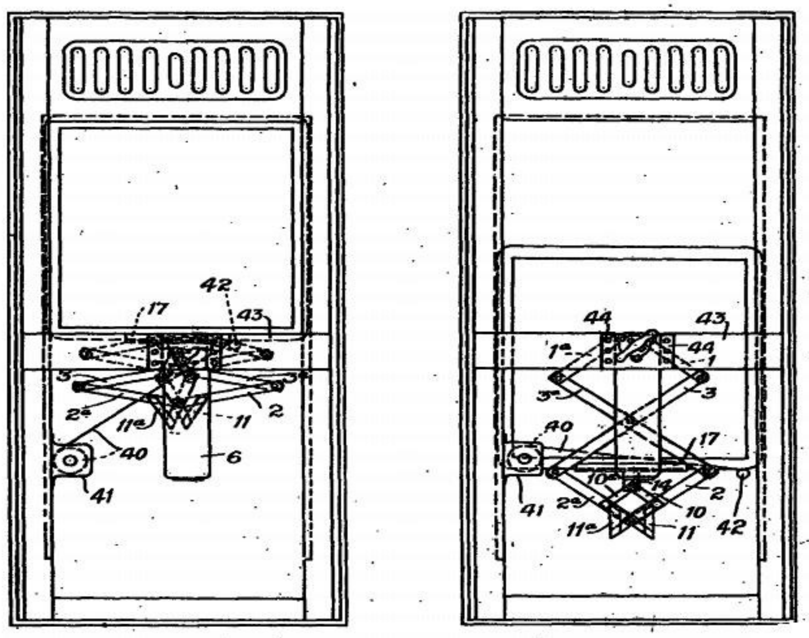


Figure 1.4 A window regulator comprising a lazy tong mechanism [5]

A different group of window regulators employs pulley and wires to obtain the translational motion of the window. One of the early patents of this type belongs to Cousinard in 1921 [7]. In this patent, window moves along the window frame by pulleys and wires. In addition, pulleys are powered by a manual handle. A drawing of this patent is given in Figure 1.5.

Another example is a patent belonging to Kuki, Isomura, Suzumura, Sakakibara and Ishihara taken in 1991 [8]. In this patent, window is attached to a bracket moving on a guide rail which is fixed to car door. The motion of this bracket on guide rail is supplied by a pulley and wire mechanism connected to an electric motor. A describing drawing of this patent is shown in Figure 1.6. In another patent belonging to Medebach, two guide rails are used with brackets [9]. Again, these brackets are attached to window and move with a different pulley and wire mechanism.

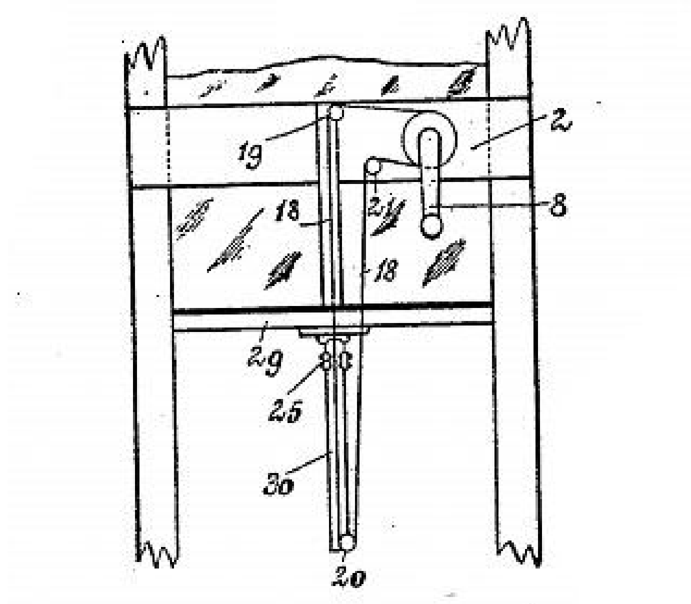


Figure 1.5 Window regulator mechanism using pulleys and wires [7]

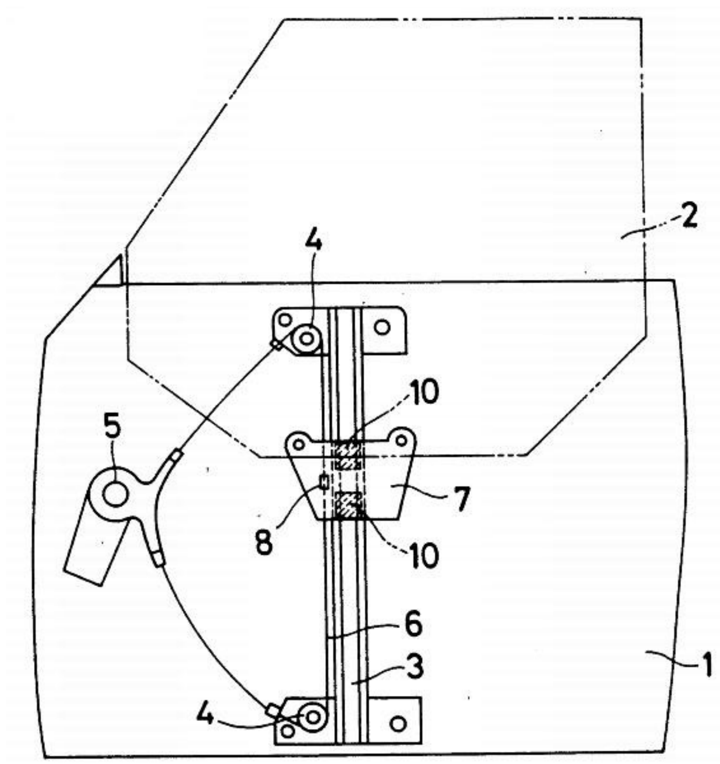


Figure 1.6 A different window regulator mechanism using pulleys and wires [8]

Last group of window regulators utilizes arms to operate the mechanism. There are lots of patents about different types of window regulators comprising arms. An early example is a patent of Seegers & Sohn taken in 1922 [10]. In this patent, window is raised or lowered using an arm with a slider at the tip. A slide attached to the window guides this slider. In addition, the arm is rotated with a gear set one of which attached to a handle. In Figure 1.7, a describing drawing of this patent is shown.

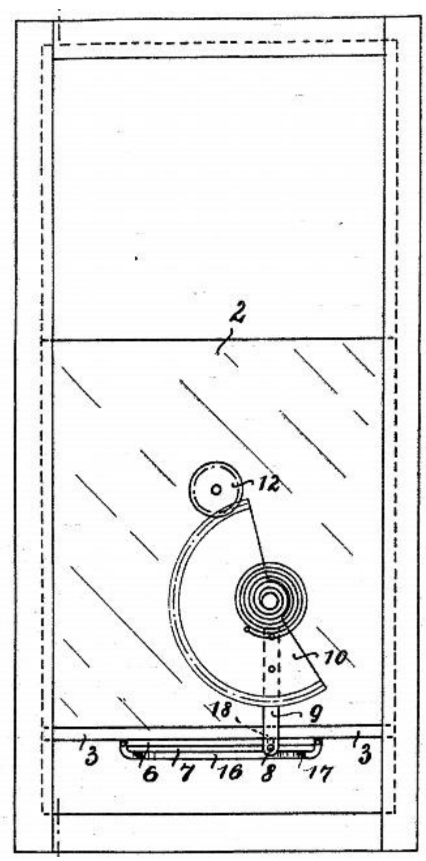


Figure 1.7 A window regulator mechanism utilizing an arm [10]

Another early patent belongs to Paul taken in 1932 [11]. This patent is similar to patent Seegers but utilizes two arms with slides. Arms are rotated with rack and pinion mechanism. The mechanism is operated with a handle attached to a gear set. A drawing of this patent is presented in Figure 1.8.

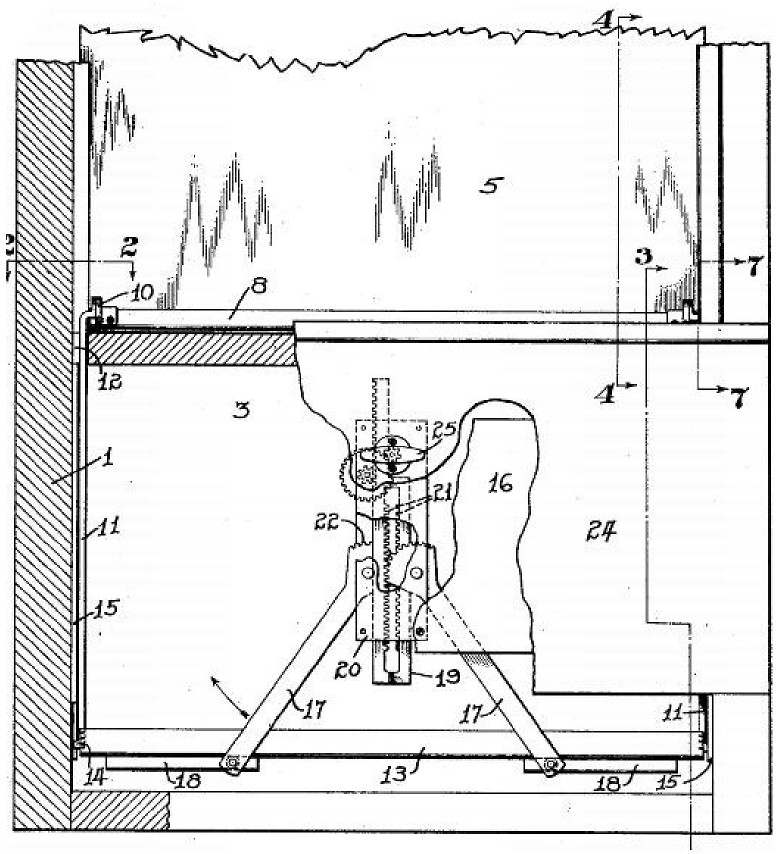


Figure 1.8 A window regulator mechanism using two arms [11]

A different example is a patent of Ternstedt Manufacturing Company taken in 1937 [12]. In this patent, a cross armed or a scissor mechanism is used. Window is attached to slide of the mechanism and moves along window frame. Mechanism is driven with a handle with a gear set. A drawing from this patent is given in Figure 1.9.

An improved version of previous one is a patent belonging to Dupuy taken in 1993 [13]. In this patent, window is raised or lowered also with a cross armed mechanism in a window frame. However, arms in this mechanism are angled. This cross armed mechanism is driven using an electric motor with a gear set. In Figure 1.10, a drawing of this patent is given.

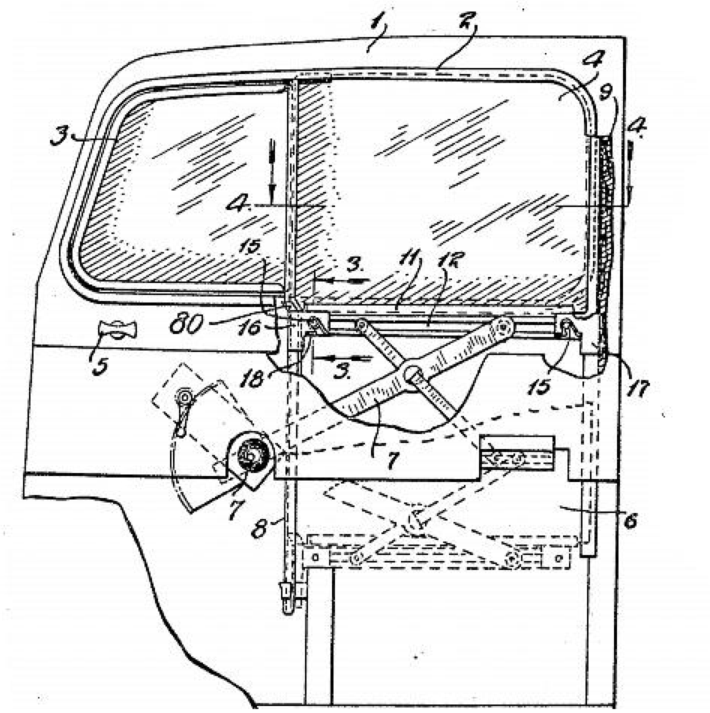


Figure 1.9 A window regulator mechanism utilizing cross armed mechanism [12]

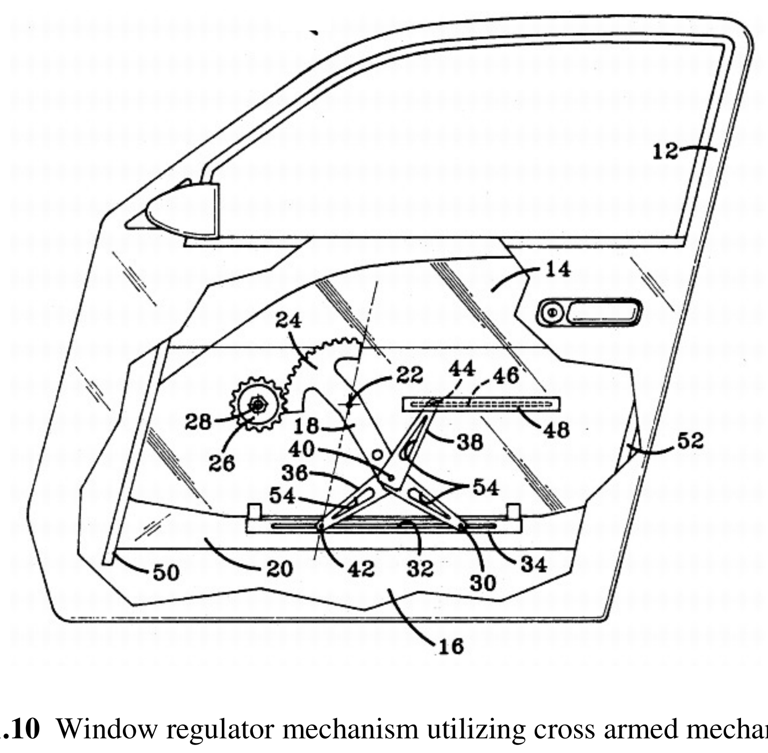


Figure 1.10 Window regulator mechanism utilizing cross armed mechanism [13]

A different example is a patent of Kriese taken in 2005 [14]. The window is moved with a different type of mechanism comprising more than two arms. The movement of the mechanism is provided with an electric motor connected to a gear set. A describing drawing of the patent is given in Figure 1.11.

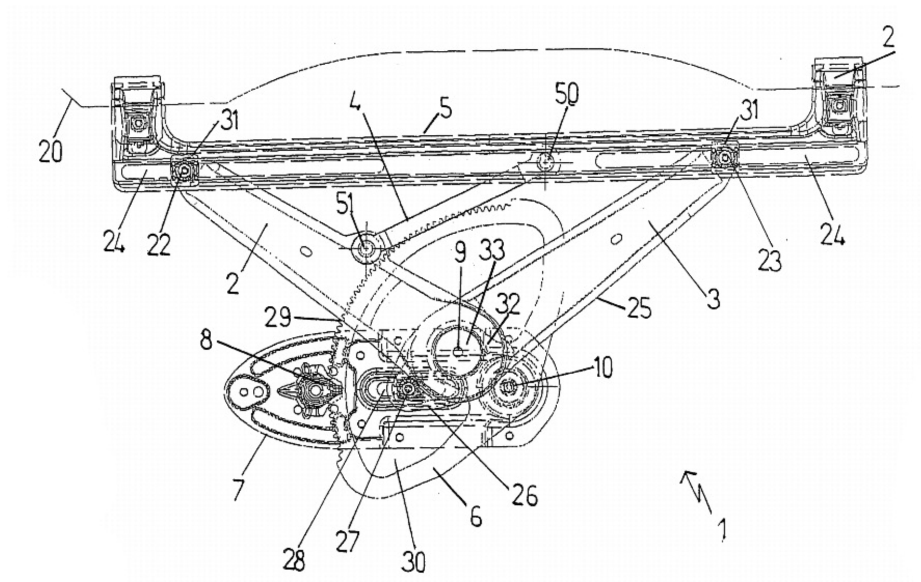


Figure 1.11 A different type of window regulator mechanism using arms [14]

1.3 Cardan Motion

The Cardan motion plays an important role in the design process, because synthesis of window regulator mechanism is carried out using Cardan motion. First, some basic definitions are presented to use the Cardan motion in synthesis.

During a planar motion of a moving reference frame in which angular acceleration is not equal to zero, instantaneously there exists a point on the plane having zero velocity relative to a fixed reference frame. This point is called *instantaneous center*. During this motion, instantaneous center defines two curves, one in the fixed reference frame and the other in the moving reference frame. The curve defined in the fixed reference frame is called *fixed centrode* and the curve defined in the moving reference frame is called *moving centrode*. These two centrodes are tangent to each other for all conditions. In addition, for special motions these two centrodes take shape of a circle. For this case, the motion for which fixed centrode radius is twice as the moving centrode radius is called *Cardan motion*, and these circles are called *Cardan circles* [15]. As a general case, any point on moving frame describes an ellipse. As a special case of Cardan motion, any point on the moving centrode describes a line passing through center of the fixed centrode.

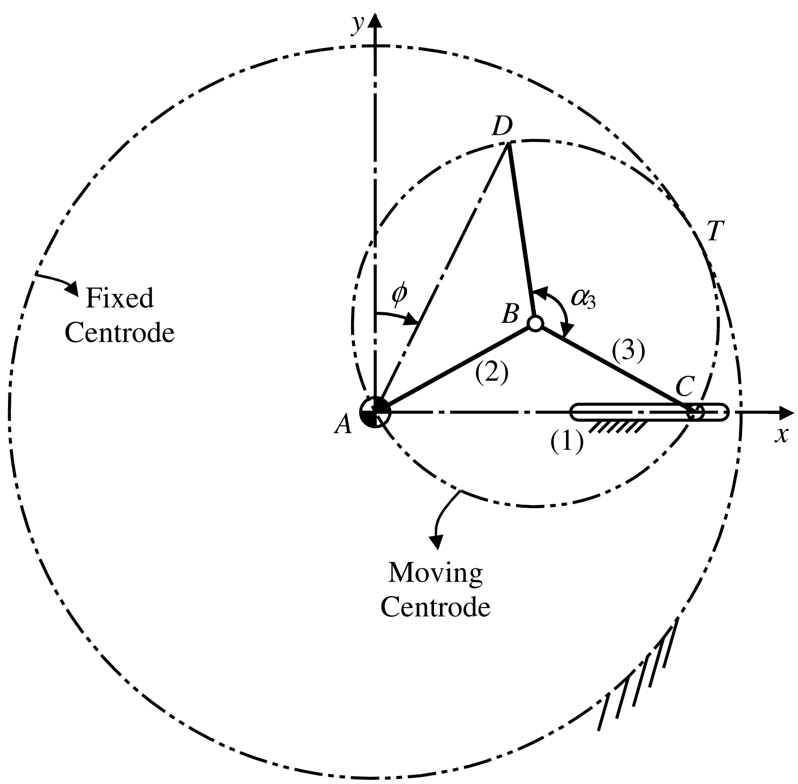


Figure 1.12 Cardan motion for a centric slider-crank mechanism

One of the known special mechanisms for which Cardan motion occurs is a centric slider-crank mechanism having equal crank and coupler lengths (Figure 1.12). Cardan motion occurs for the coupler link of this mechanism. In this case, Point *A* is the center of fixed centrode and point B is the centre of the moving centrode. These two circles are tangent at point T. Point D taken on the moving centrode, describes an exact line passing through the center of the fixed centrode (Point A). Let ϕ be the clockwise angle from y-axis to line AD and α_3 be the angle between lines DB and BC, an equation can be found between these two angles as [16];

$$\phi = \frac{\pi - \alpha_3}{2} \tag{1.1}$$

The general condition to have such a line is;

$$|AB| = |BC| = |BD| \tag{1.2}$$

1.4 Aim of the Study

The aim of this thesis is to design a window regulator for a car door. The designed window regulator should provide the desired motion for the window. In addition, during the design stage all restrictions relating window regulators must be considered. Finally, the resultant design is implemented on a RENAULT car door.

1.5 Outline of the Thesis

In this thesis, window regulators are introduced and design of a window regulator for a car door is presented. The following chapters are prepared to introduce and indicate the design process.

In Chapter 2, conceptual design of the window regulator is presented. Firstly, concepts are defined, then concept evaluation variants are defined, lastly concepts are evaluated according to concept evaluation variants to obtain the most suitable mechanism.

In Chapter 3, detailed design of the window regulator is described. Firstly, type synthesis is performed, then kinematic synthesis and analysis is carried out, afterwards basic restrictions on the mechanism design is completed, finally force analysis and design of the spiral spring is completed.

In Chapter 4, the implementation of the design is given. The implementation is done on a RENAULT car door. Firstly, kinematic synthesis is performed. Then, embodiment of the design is done. Afterwards, force analysis and spiral spring design is completed. Lastly, manufacturing of the design is mentioned.

In Chapter 5, the obtained results are discussed and the thesis is concluded by giving some recommendations for future work and further improvements.

CHAPTER 2

CONCEPTUAL DESIGN

2.1 Introduction

In the conceptual design stage, different concepts should be generated and then the best concept should be selected to satisfy design needs. At the selection of best concept, the evaluation of created concepts is performed subjective by its nature. Therefore, conceptual design is another critical stage of the study.

In this chapter, firstly the created concepts are presented in concept development and presentation. Afterwards, concept evaluation criteria are defined. Lastly, the best concept is selected among created concepts using concept evaluation criteria. The procedure followed in this chapter is a simpler version of the conceptual design procedure defined by Pahl and Beitz [17].

2.2 Concept Development and Presentation

Literature survey constitutes the background for concept development. Therefore, five different concepts are created based on types of window regulator mechanisms found in the literature survey.

2.2.1 Concept I

The first developed concept is the screw-driven window regulator mechanism. In this concept, the motion of the window is provided by a power-screw mechanism. The screw is fixed to the car door. In addition; the nut is attached to the power screw with a casing and driven by an electric motor through a gear set. Electric motor and gear set are secured in a casing. The rotation of nut is free in the casing. Furthermore, the window is rigidly attached to the casing of the nut and guided through the window frame. Therefore, as the nut is driven by the motor, the window makes a translational motion through the window frame.

An illustrative sketch of Concept I is given in Figure 2.1. Nowadays, car doors are manufactured in a curved shape. Thus, the window should make a curved motion in the window frame. So, the screw is illustrated in a curved manner.

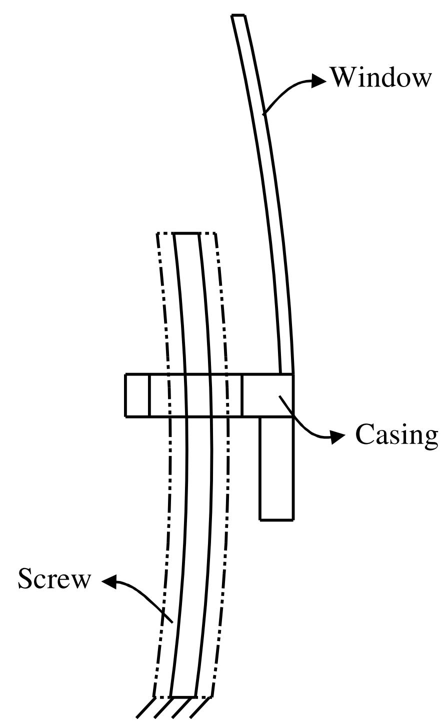


Figure 2.1 An illustrative sketch of Concept I

2.2.2 Concept II

The second concept is operated by a rack and pinion. In this concept, the rack is fixedly connected to the car door. The pinion is attached to the rack in a casing to secure the connection between them. Furthermore, the pinion is driven by a gear set powered by an electric motor which is also connected to the casing. In addition, the window is also rigidly connected to the casing. Thus; as the pinion is rotated, the window makes a translational motion along the rack.

In Figure 2.2, a descriptive sketch of the Concept II is given. Note that, the rack should be also in a curved shaped in the side view to provide the aforementioned curved motion of the window.

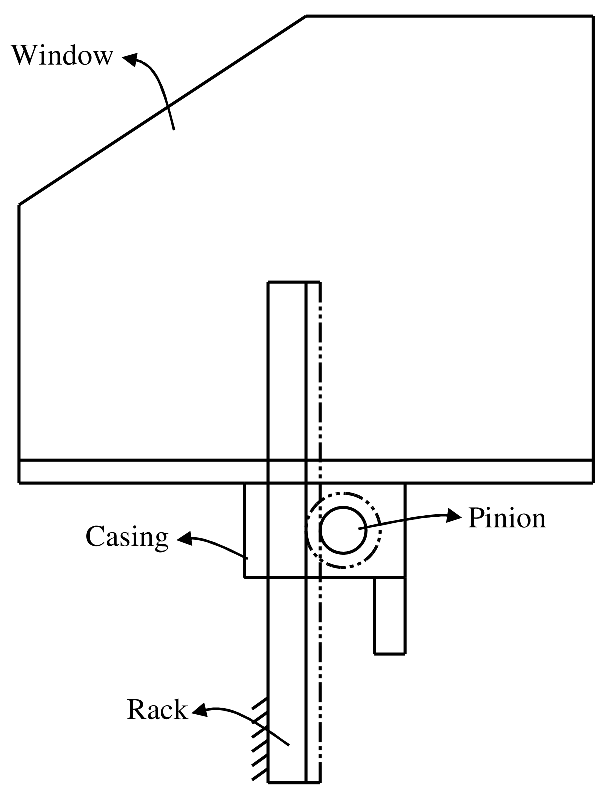


Figure 2.2 A descriptive sketch of Concept II

2.2.3 Concept III

Third concept is utilizing wire and pulleys. In this concept, a wire is attached to the main pulley which is rotated by an electric motor. This electric motor is connected to the car door chassis. In addition, the wire is also connected to a bracket moving on a carriage which is fixed to the car door chassis. Furthermore, the window is rigidly attached to the bracket. So, when the main pulley is rotated, the bracket is pulled along the carriage with the help of two additional pulleys.

An explanatory sketch of the Concept III is shown in Figure 2.3. Note that, the carriage should be also in a curved shaped in the side view to provide the aforesaid curved motion of the window.

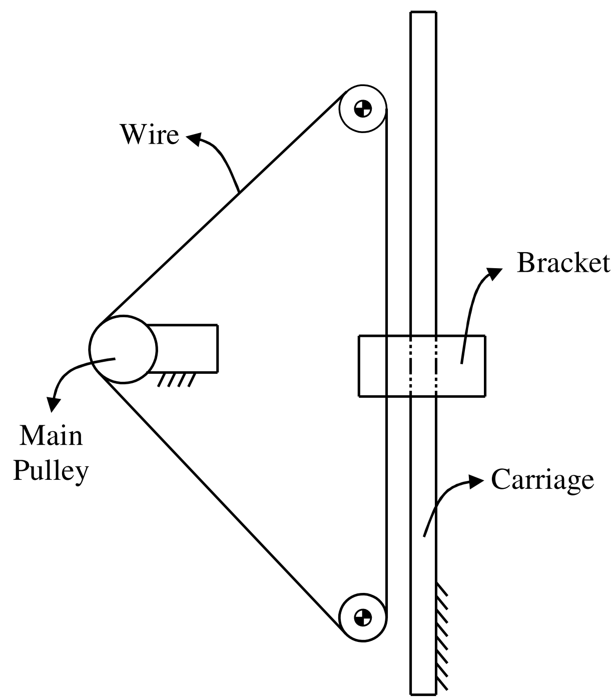


Figure 2.3 An explanatory sketch of Concept III

2.2.4 Concept IV

Fourth concept comprises an arm type mechanism. In this concept, an arm is rigidly connected to a gear which is driven by a pinion. A slider is attached to the tip of the arm and connected to a moving slide. The window which is guided through the window frame is fixed to moving slide. In addition to these, the pinion is operated by an electric motor. Therefore; as the pinion is rotated by the electric motor, the window makes a translational motion through the window frame.

An illustrative sketch of Concept IV is given in Figure 2.4. Note that, the arm can bend to provide the abovementioned curved motion of the window.

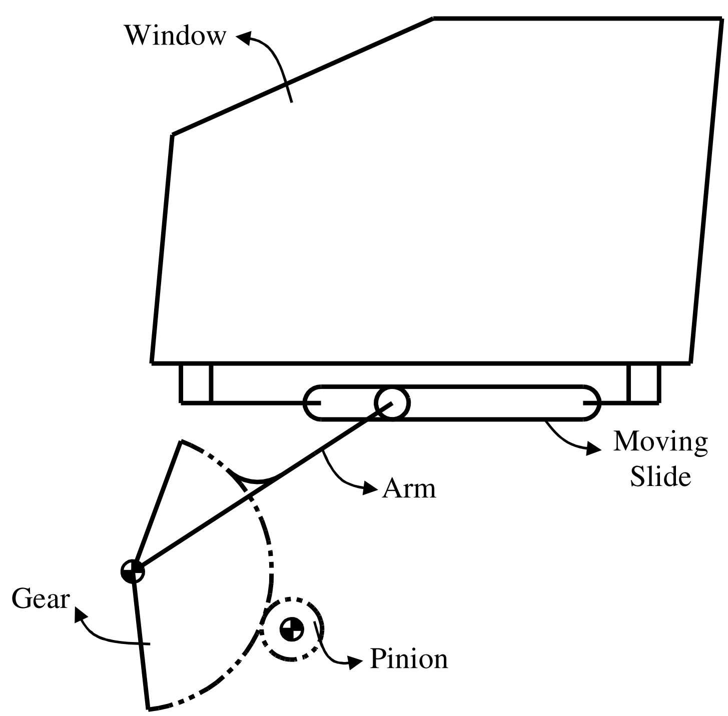


Figure 2.4 An illustrative sketch of Concept IV

2.2.4 Concept V

The last concept utilizes a cross armed or a scissor mechanism. The input link of the cross armed mechanism is fixed to a gear which is driven by a pinion. The window is rigidly attached to the moving guide of the cross armed mechanism and guided in the window frame. In addition, the pinion is driven using an electric motor. Therefore, as the input link is operated, the window makes a translational motion through the window frame.

In Figure 2.5, a descriptive sketch of the Concept II is given. Note that; as in Concept IV, arms can bend to provide the curved motion of the window.

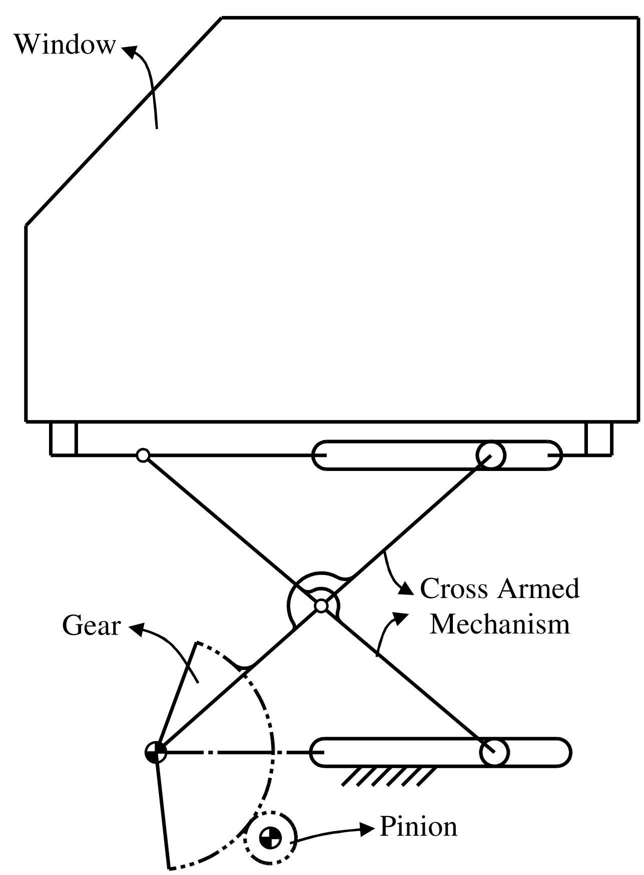


Figure 2.5 A descriptive sketch of Concept V

2.3 Concept Evaluation Criteria

After concept development is completed, the next step is clarifying concept evolution criteria. Concept evaluation criteria can be chosen among a variety of technical, economic, technical, etc. aspects. However, these criteria must be selected such that differences between concepts can be observed during the evaluation process.

Chosen evaluation criteria are small number of components, low complexity of components, low complexity of concept, long service life, regular force transmission, low noise operation, simple assembly and low space utilization.

Small number of components is mainly a cost criterion since number of components directly affects the overall cost of the product. If the product consists of higher number of components, more materials will be used in the manufacturing stage of the product.

Low complexity of components also affects cost of the product. Because, complex components necessitate using different production techniques resulting in a costly manufacturing stage. In addition, operational problems may occur because of complex components.

Low complexity of concept influences functioning and design process of the product. Because, complex designs demand complex sub-functions and assemblies which may cause a worse functioning of the design.

Long service life directly affects maintenance cost of the product.

Furthermore, regular force transmission influences functioning and maintenance of the product. Because, undesired force transmission may cause higher reaction forces resulting high wear of components.

Low noise operation affects ergonomics of the resultant product.

Lastly, simple assembly and low space utilization influences assembly stage of the product.

2.3 Evaluation of Concepts

Once the concept evaluation criteria are defined, the evaluation of concepts can be performed. To do this, firstly corresponding weights should be assigned to concept evaluation criteria.

In Table 2.1, assigned weights to concept evaluation criteria are shown. In this table, firstly weights are assigned from points zero to ten according to its importance for the design. Note that, this assignment is subjective. Then, these weights are rearranged such that they sum up to total one.

Table 2.1 Assigned weights to concept evaluation criteria

Evaluation Criterion	Weight (0-10)	Rearranged Weight
Small Number of Components	8	0,145
Low Complexity of Components	8	0,145
Low Complexity of Concept	8	0,145
Long Service Life	7	0,127
Regular Force Transmission	7	0,127
Low Noise Operation	7	0,127
Simple Assembly	5	0,091
Low Space Utilization	5	0,091
TOTAL	55	1

After assigning weights to concept evaluation criteria, the evaluation of concepts can be performed. The values that will be given to concepts should not be arbitrary. In fact, values should be given according to a value scale.

In Table 2.2, value scale used for the evaluation of concepts is given.

Table 2.2 Value scale used for the evaluation of concepts [17]

Points	Meaning
0	Absolutely useless solution
1	Very inadequate solution
2	Weak solution
3	Tolerable solution
4	Adequate solution
5	Satisfactory solution
6	Good solution with few drawbacks
7	Good solution
8	Very good solution
9	Solution exceeding the requirement
10	Ideal solution

Note that, the evaluation of concepts is subjective and the experience of the designer plays an important role during the evaluation process. However, the deviation of values because of this subjective approach can be tolerated by making the evaluation process more than once. Then, the best concept can be selected accordingly.

Finally, concepts are evaluated and the results are given in Table 2.3. As seen in the table; Concept V (Figure 2.5), which is window regulator comprising cross armed mechanism, came out to be the best concept.

Table 2.3 Evaluation of Concepts

Evaluation Criterion	**Weight**	Concept I	Concept II	Concept III	Concept IV	Concept V
Small Number of Components	0,145	6	4	6	8	6
Low Complexity of Components	0,145	2	2	6	8	8
Low Complexity of Concept	0,145	4	2	8	8	6
Long Service Life	0,127	6	6	4	2	8
Regular Force Transmission	0,127	4	4	6	2	8
Low Noise Operation	0,127	4	4	8	6	8
Simple Assembly	0,091	4	4	6	8	6
Low Space Utilization	0,091	4	4	6	8	4
TOTAL	1	**4,25**	**3,67**	**6,29**	**6,22**	**6,87**

CHAPTER 3

DETAILED DESIGN

3.1 Introduction

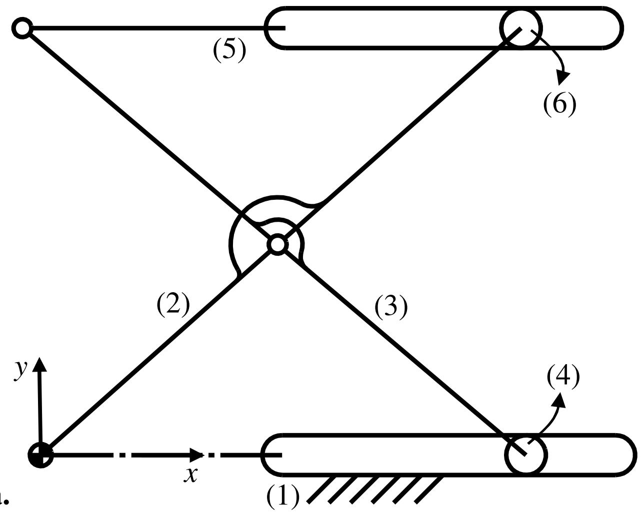
In the detailed design stage the chosen concept should be shaped in detail, in other words all parts should be formed with necessary functions and dimensions to fulfill the design criterias and restrictions. Therefore, detailed design is the most time-consuming stage of the overall design process. So, the approach to the detailed design stage should be systematic and well-planned.

The detailed design is started with the type synthesis for the selected concept. Afterwards, synthesis of the mechanism is performed to achieve the desired motion of the output link. Next, basic restrictions on the mechanism design are identified so that the resultant mechanism satisfies essential design criteria. Then, kinematic analysis is carried out to fully define the position and the motion of the mechanism. Once kinematics of the design is completed, force analysis of the mechanism is done to obtain necessary forces acting on the system. Then, design of the spiral spring is presented. Lastly, design of the gear mesh is performed.

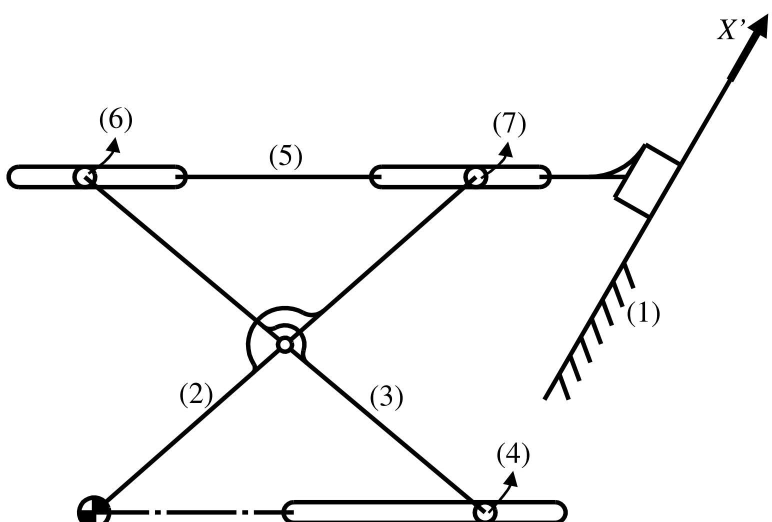
3.2 Type Synthesis

After obtaining the best concept in the conceptual design, type of the mechanism should be decided. Because there are more than one mechanism types that suit the decided concept. Thus, type synthesis is carried out for the selected concept to get the most suitable mechanism type.

There are three mechanism types that can be utilized as a window regulator for the chosen concept (Figure 3.1).



a.



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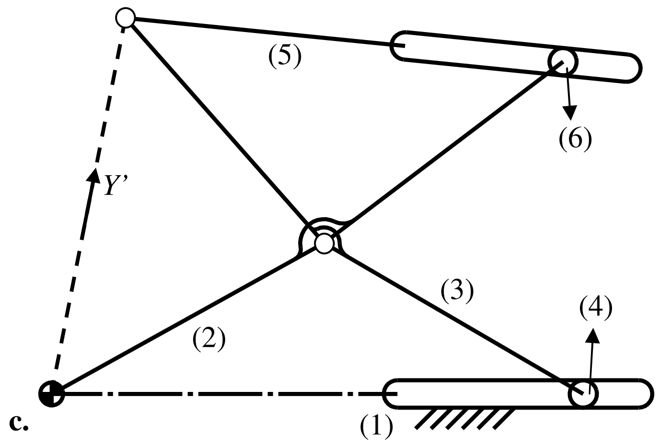


Figure 3.1 Schematic representation of different mechanism types that can be used as a window regulator; a. Type I, b. Type II, c. Type III

The window regulator in Figure 3.1.a is named as "Type I". Type I comprises a scissor-type mechanism having straightly shaped linkages. In this mechanism, link (5), which is connected to window of the car door, makes a translational motion along the *y*-axis (Figure 3.1.a).

Another type of window regulator is named as "Type II" (Figure 3.1.b). Type II also includes a scissor-type mechanism having straightly shaped links. In this case, link (5), which is attached to the window, is guided with the plastics surrounding the window. In this manner, link (5) performs a translational motion along the *X'*-axis (Figure 3.1.b).

"Type III" is the last type of window regulator mechanism (Figure 3.1.c). Type III also contains a scissor-type mechanism but this time having angulated links. For specific proportions of the mechanism, link (5) undergoes a translational motion along the *Y'*-axis (Figure 3.1.b). (Please refer to section 3.3.2 for details)

The comparison between these mechanisms is done according to two main criteria. First one is the flexibility of the synthesis that is mainly flexibility on the placements of the fixed parts while synthesizing the mechanism. The second criterion is the easiness of operation which depends mostly on the internal friction between the joints.

Considering the first criterion, Type I's flexibility is less than other two types since the direction of the translational motion (y-axis in Figure 3.1.a) must always be perpendicular to the axis of the fixed slide (x-axis in Figure 3.1.a). This situation brings additional restriction on the placement of fixed parts. On the other hand, Type II and Type III have same flexibility since arrangements of fixed parts relative to translational motion are not limited.

It can be claimed that Type II's easiness of operation is the worst one when the second criterion is accounted. During the motion of this mechanism, direction of link (5)'s translational motion is provided with the help of the plastic guide surrounding the glass, i.e. there occurs a friction force between link (5) and link (1). Therefore, this friction force makes the movement of the mechanism harder. In contrast, Type I and Type III have same easiness of operation since their only difference is that Type III has angulated linkages and Type I has straightly shaped linkages.

When two criteria are examined together, Type III is come out to be the most suitable mechanism type for this design. Because for both comparisons, only Type III appears to be superior to others.

3.3 Kinematic Synthesis of the Mechanism

Kinematic study of the design starts with the kinematic synthesis of the mechanism. Kinematic synthesis is done to find the specific mechanism that generates the defined function, path or in this case provides the pre-described body guidance. In the kinematic synthesis stage, all constant link parameters are derived parametrically.

First of all, the desired motion is described and geometric constraints of the design are defined. Afterwards, according to these geometric constraints, synthesis of the mechanism is completed. Whilst synthesizing the mechanism, first the required motion for the window is acquired, and then the link lengths for this motion are found.

3.3.1 Geometric Constraints

In previous chapters, it is stated that the aim is to design a window regulator mechanism that provides a translational motion for the window of the car door. Generally car door and window geometry is designed before the window regulator mechanism. Therefore, the desired motion for the window is prescribed. Thus, before performing synthesis of the mechanism, aforesaid translational motion should be defined clearly.

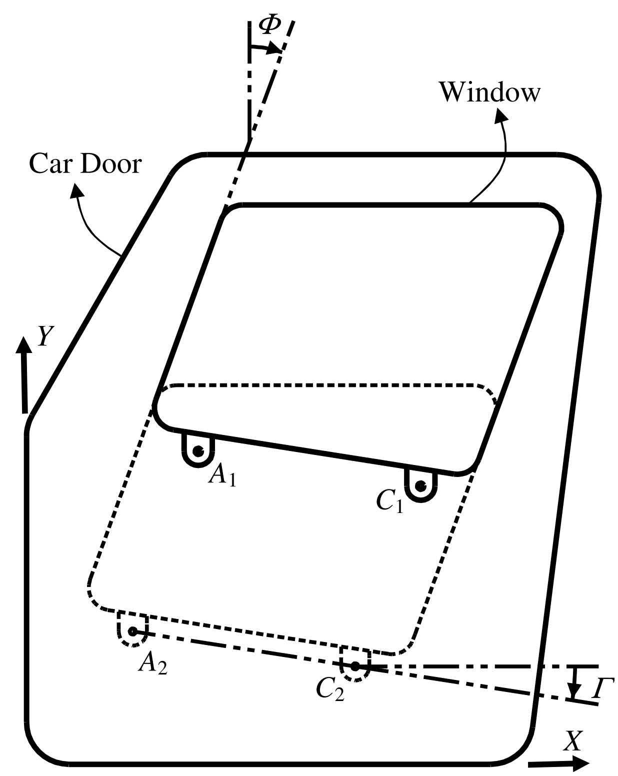


Figure 3.2 Schematic representation of the car door with the window

In Figure 3.2, a schematic representation of car door is shown. The highest and the lowest positions of the window are also illustrated in the figure. In the figure, global axis frame, *X*-*Y* is shown. *Y*-axis is defined opposite to gravitational axis. The output link of the window regulator mechanism is rigidly attached to the window at the plastic holders (Points *A* and *C* in Figure 3.2).

To fully define the translational motion of the window, parametric constraints are defined. The clockwise angle formed from *Y* axis to the axis of translational motion is defined as Φ. Also, the clockwise angle formed from *X* axis to axis of the holder (formed from point *A* to *C* in Figure 3.2) is labeled as Γ.

To obtain the required displacement of the window, side view of the car door is examined (Figure 3.3).

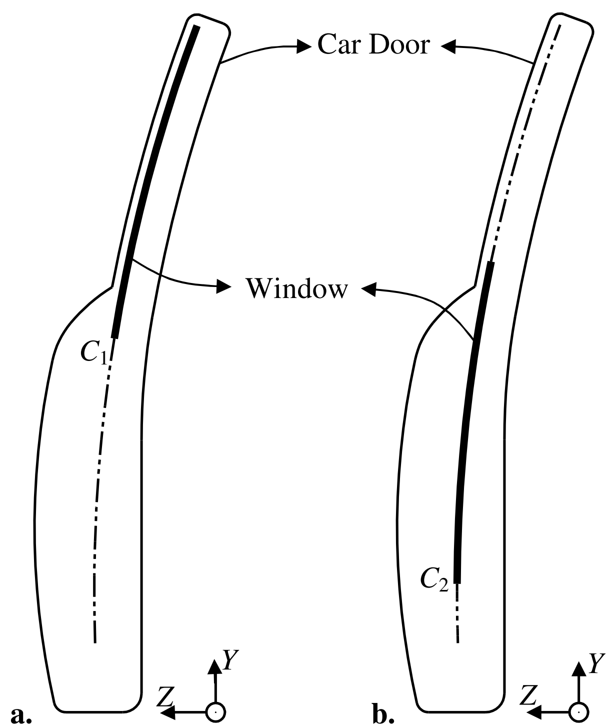


Figure 3.3 Side view of the car door

Nowadays, car doors are designed in curved shape. Because of this, the window can not make rectilinear translation during the motion, thus the trajectory of the window can not be a straight line in the side view. In fact, the window is forced to make a rotational motion and the trajectory of the window comes out to be an arc which is a part of a big circle (Figure 3.3).

In Figure 3.4, an exaggerated view of the trajectory of the window is shown.

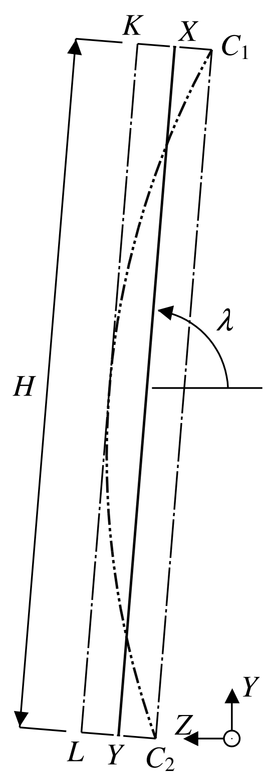


Figure 3.4 Exaggerated view of the trajectory of the window

C_1 is the top position and C_2 is the lowest position of the window holder. Line KL is drawn tangent to the window trajectory and also parallel to line C_1C_2. Line XY is equally distant to lines KL and C_1C_2. The angle formed form the Z-axis to line XY is λ. The distance from the line XY to lines KL and C_1C_2 is defined as δ.

Operating plane of the mechanism is placed coincident with line *XY*. The output link of the mechanism is rigidly attached to the window holder. Therefore, the required displacement for the output link is *H*.

Also, operating plane of the mechanism should bend maximum δ amount at the top and lowest position. It is clear that δ will be small enough since the trajectory of the window is part of a big circle. Therefore, bending of the mechanism parts will not cause much problem.

3.3.2 Synthesis of the Required Motion

In the previous part, the translational movement of the window is defined. The window is rigidly attached to the output link of the mechanism. Therefore, output link of the mechanism should also make the same translational movement.

The synthesis of the mechanism can be handled treating the slider-crank mechanism as a part of the whole window regulator mechanism. In other words, window regulator mechanism can be obtained using a slider-crank mechanism. In Figure 3.5, this approach for the synthesis of the mechanism is shown.

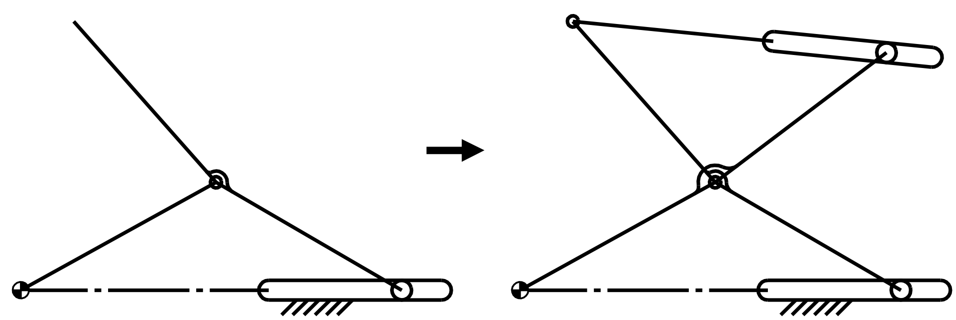


Figure 3.5 Approach for the synthesis of the mechanism

Therefore, this movement is synthesized in two steps. First step is obtaining the direction of the movement using the slider-crank portion of the mechanism. Then, translational movement of the output link is obtained using the whole mechanism.

3.3.2.1 Obtaining the Direction of the Movement

The direction of the movement is defined with the geometric constraints in Chapter 3.3.1. Here, the aim is to obtain this direction of the movement using the slider-crank portion of the mechanism according to these constraints (Figure 3.6).

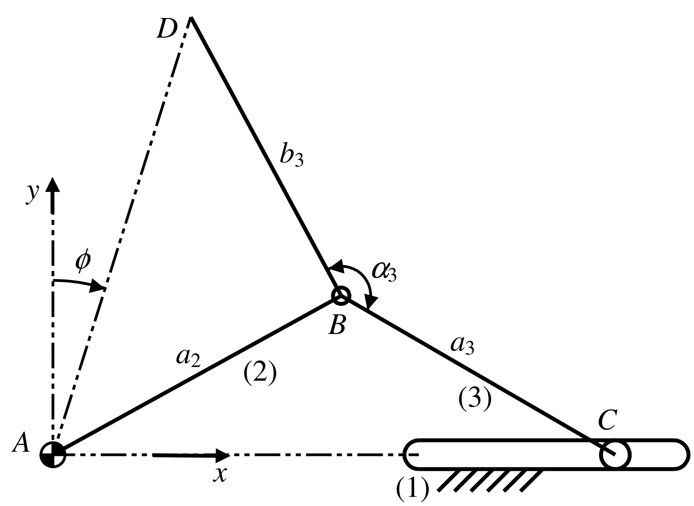


Figure 3.6 Slider-crank portion of the mechanism

The desired direction of movement is achieved using the Cardanic motion. In this case, while the mechanism functions, it is desired to make path of point *D* to describe a straight line. Thus, according to the Cardanic motion (please refer to Section 1.3), the condition for point *D* to move along the line *AD* is,

$$|AB| = |BC| = |BD| \tag{3.1}$$

Also, the constant link angle of linkage (3), α_3 is found using Equation (1.1) as,

$$\alpha_3 = \pi - 2 \cdot \phi \tag{3.2}$$

In Figure 3.7, general position of the mechanism in the car door is considered to obtain the relationships with the geometric constraints. *X-Y* axis frame is the global axis frame as also defined in Figure 3.2. On the other hand, in the *x-y* axis frame, *x*-axis is directed towards the fixed slide (towards point *A* to point *C* in Figure 3.7). The angle α_0 can be defined as the rotation angle from *X-Y* axis frame to *x-y* axis frame. Note that α_0 is not a geometric constraint; so it can be taken as a design parameter.

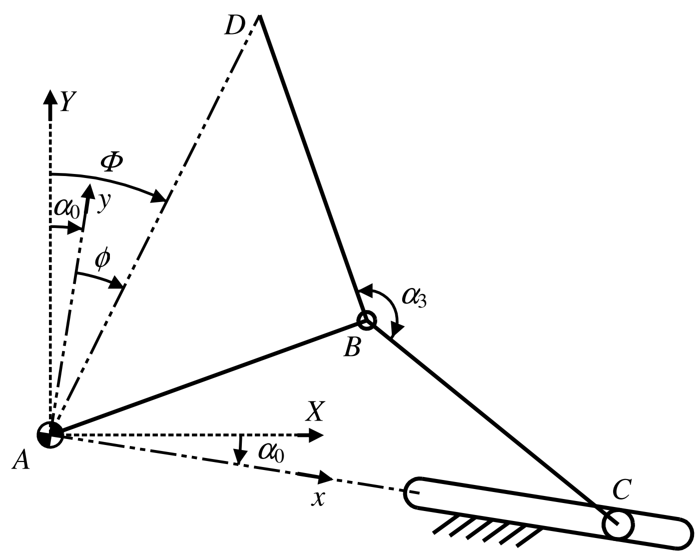


Figure 3.7 General position of the slider-crank mechanism in the car door

Therefore,

$$\Phi = \phi + \alpha_0 \tag{3.3}$$

Using Equations (3.2) and (3.3), α_3 can be derived as,

$$\alpha_3 = \pi - 2 \cdot (\Phi - \alpha_0) \tag{3.4}$$

Then, using Equations (3.1) and (3.3) the required direction of movement can be obtained. This movement of the slider-crank mechanism provides basis for the window regulator mechanism.

3.3.2.2 Obtaining the Translational Movement

When obtaining the translational movement of the output link, whole mechanism is considered (Figure 3.8). So other portion of the mechanism is attached to the previously synthesized slider-crank mechanism.

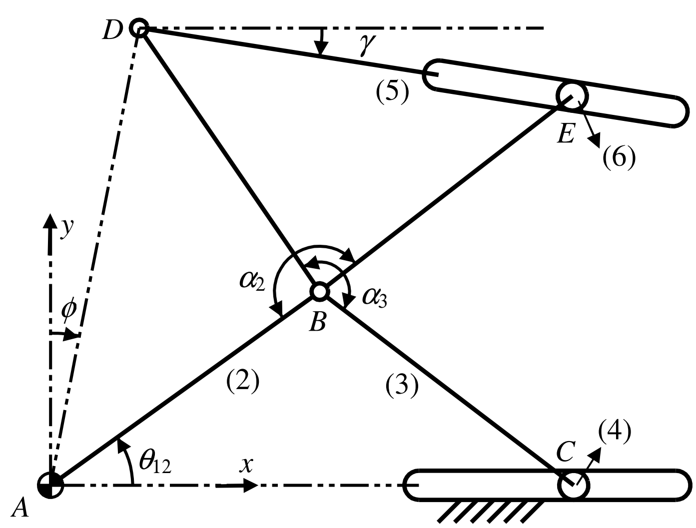


Figure 3.8 Window regulator mechanism

The angle, γ is defined as the clockwise angle from x-axis to link (5). Crank angle, θ_{12} is defined as the counter-clockwise angle from the x-axis to link (2). To obtain the translational motion for link (5), geometry is used (Figure 3.9).

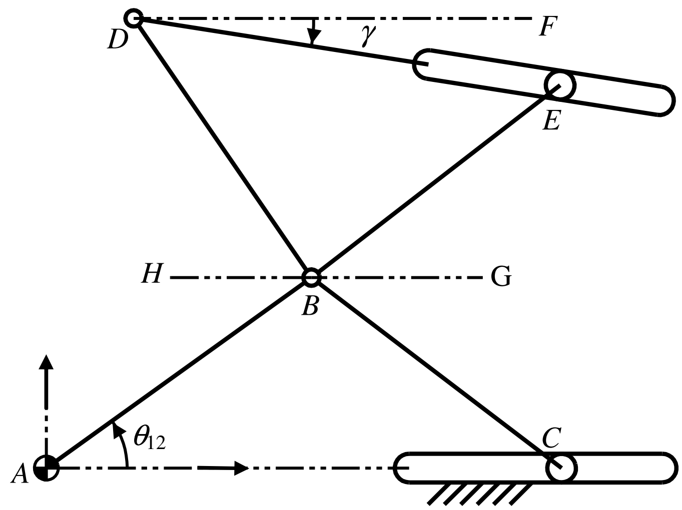


Figure 3.9 Obtaining the Translational Movement

It is previously found that $|AB| = |BC|$, so;

$$\angle CAB = \angle ACB = \theta_{12} \tag{3.5}$$

On the other hand, *HG // AC*, thus;

$$\angle ABH = \angle CAB = \theta_{12} \tag{3.6}$$

$$\angle CBG = \angle ACB = \theta_{12} \tag{3.7}$$

Also, using Equations (3.6) and (3.7);

$$\angle GBD = \angle CBD - \angle CBG = \alpha_3 - \theta_{12} \tag{3.8}$$

$$\angle HBE = \angle ABE - \angle ABH = \alpha_2 - \theta_{12} \tag{3.9}$$

Afterwards, with Equations (3.8) and (3.9);

$$\angle HBD = \pi - \angle GBD = \pi - \alpha_3 + \theta_{12} \tag{3.10}$$

$$\angle GBE = \pi - \angle HBE = \pi - \alpha_2 + \theta_{12} \tag{3.11}$$

Then, angle $\angle DBE$ is found using (3.10) and (3.11) as;

$$\angle DBE = \pi - \angle HBD - \angle GBE$$
$$\angle DBE = \alpha_3 + \alpha_2 - 2\theta_{12} - \pi \tag{3.12}$$

After that, if $|BD| = |BE|$, the triangle *BDE* becomes an isosceles triangle. Therefore, angle $\angle BDE$ is found using (3.12) as;

$$\angle BDE = \frac{\pi - \angle DBE}{2}$$
$$\angle BDE = \pi + \theta_{12} - \frac{\alpha_3 + \alpha_2}{2} \tag{3.13}$$

It is known that *HG // DF*, so;

$$\angle BDF = \angle HBD = \pi - \alpha_3 + \theta_{12} \tag{3.14}$$

Lastly, angle γ is found using Equations (3.13) and (3.14) as;

$$\gamma = \angle BDF - \angle BDE$$
$$\gamma = \frac{\alpha_2 - \alpha_3}{2} \tag{3.15}$$

From the Equation (3.15), it is found out that angle γ is independent from the input crank angle; θ_{12} and depends only on the constant link angles. Therefore, angle γ is constant in all input crank angles. Thus, link (5) makes a translational motion as crank rotates.

While performing the derivation, the condition $|BD| = |BE|$ is used. Also, using Equation (3.1), the general condition for the output link to make the required translational motion is derived;

$$|AB| = |BC| = |BD| = |BE| \tag{3.16}$$

In Figure 3.10, again general position of the mechanism in the car door is considered to obtain the relationships with the geometric constraints defined in chapter 3.3.1.

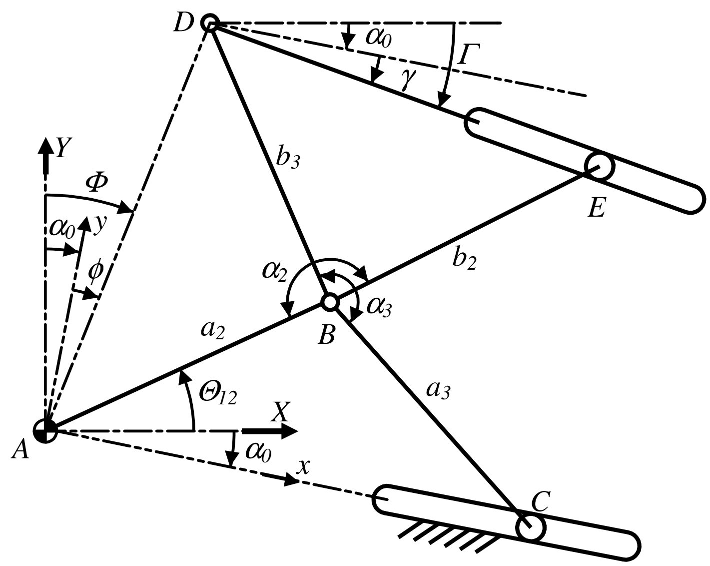


Figure 3.10 General position of the window regulator mechanism in the car door

From the Figure 3.10;

$$\Gamma = \gamma + \alpha_0 \tag{3.17}$$

Using Equations (3.15) and (3.17) with (3.4), angle α_2 is found depending on the constant parameters as;

$$\alpha_2 = \pi + 2 \cdot (\Gamma - \Phi) \tag{3.18}$$

Note that, α_2 depends only on geometric constraints so that α_2 can be obtained when Γ and Φ are specified.

3.3.3 Deriving the Link Lengths

The last step for the synthesis of the mechanism is deriving the link lengths of the mechanism. To perform this, open and closed positions of the window must be known (Figure 3.2). These are the top and the lowest positions of the output link of the mechanism.

Afterwards, a suitable position of the mechanism in the car door should be decided. This is done by specifying the position of the fixed pivot (point *A* in Figure 3.10) and the direction of the fixed slide (angle α_0). Then, using the top and the lowest positions of the output link (Figure 3.11), lengths of the linkages and total crank angle, $\theta_{12,total}$ is found.

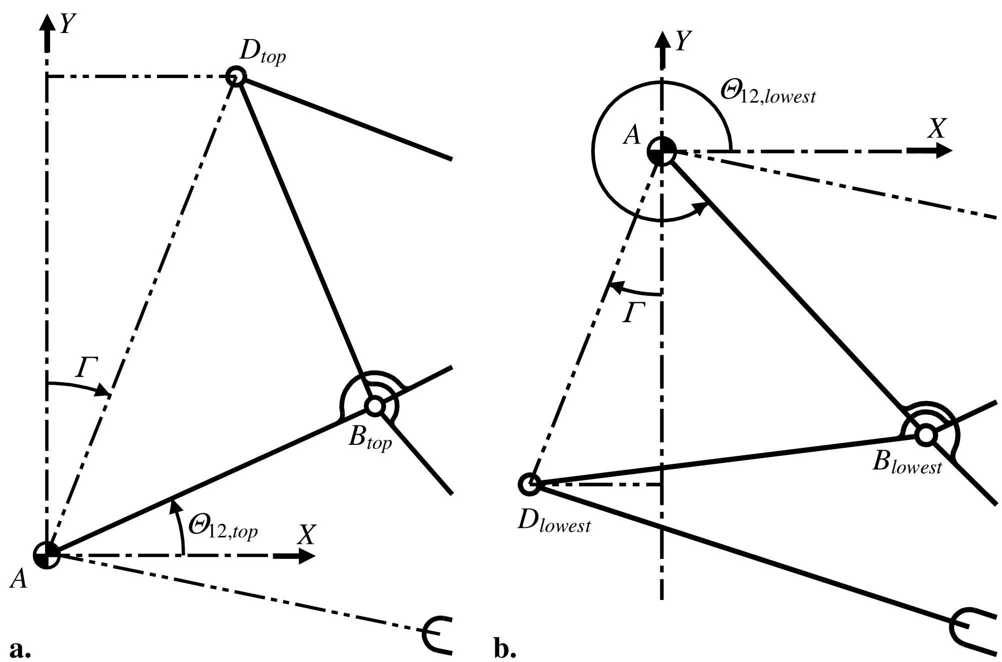


Figure 3.11 Some portions of the window regulator mechanism **a.** at the top position **b.** at the lowest position

Previously it is found that all link lengths of the scissor-mechanism are equal (Equation 3.16). Length of these elements is defined as r. Also, Y-axis components of the point D are defined as Y_{top} at the top position and Y_{lowest} at the lowest position. In addition, the crank angle is defined as $\Theta_{12,top}$ at the top position and $\Theta_{12,lowest}$ at the lowest position. Then, from the trigonometry two equations are written as;

$$Y_{top} = 2 \cdot r \cdot \cos\left(\pi/2 - \Theta_{12,top} - \Gamma\right) \cdot \cos\left(\Gamma\right) \tag{3.19}$$

$$Y_{lowest} = 2 \cdot r \cdot \cos\left(\Theta_{12,lowest} + \Gamma - 3 \cdot \pi/2\right) \cdot \cos\left(\Gamma\right) \tag{3.20}$$

There are three unknowns in these two equations; r, $\Theta_{12,top}$ and $\Theta_{12,lowest}$. One of these can be selected as a design parameter and other two can be found in terms of the design parameter. Therefore, if $\Theta_{12,top}$ is selected as a design parameter; r and $\Theta_{12,lowest}$ can be found using Equations (3.19) and (3.20) as;

$$r = \frac{Y_{top}}{2 \cdot \cos\left(\pi/2 - \Theta_{12,top} - \Gamma\right) \cdot \cos\left(\Gamma\right)} \tag{3.21}$$

$$\Theta_{12,lowest} = 3 \cdot \pi/2 - \Gamma + \cos^{-1}\left(\frac{Y_{lowest}}{2 \cdot r \cdot \cos\left(\Gamma\right)}\right) \tag{3.22}$$

Also, total crank rotation can be found as;

$$\Theta_{12,total} = \Theta_{12,top} + \Theta_{12,lowest} \tag{3.23}$$

Note that, while finding Y_{top} and Y_{lowest}, the kinematic constraint defined in chapter 3.3.1 must be used, i.e. following equation must be satisfied;

$$H = Y_{top} + Y_{lowest} \tag{3.24}$$

Note that, position the fixed pivot (point A in Figure 3.11) and the direction of the fixed slide (angle α_0) are also free design parameters. In summary, all free design parameters are A_x, A_y, α_0 and $\Theta_{12,top}$. But, restrictions due to obstacles in the car door should be taken into account while changing these design parameters.

3.4 Kinematic Analysis of the Mechanism

After synthesizing the mechanism, the motion of the mechanism should be derived. Thus, kinematic analysis is performed to obtain the position of the mechanism for every specified input joint variable.

Firstly, joint variables and symbolic representation of link lengths are assigned (Figure 3.12).

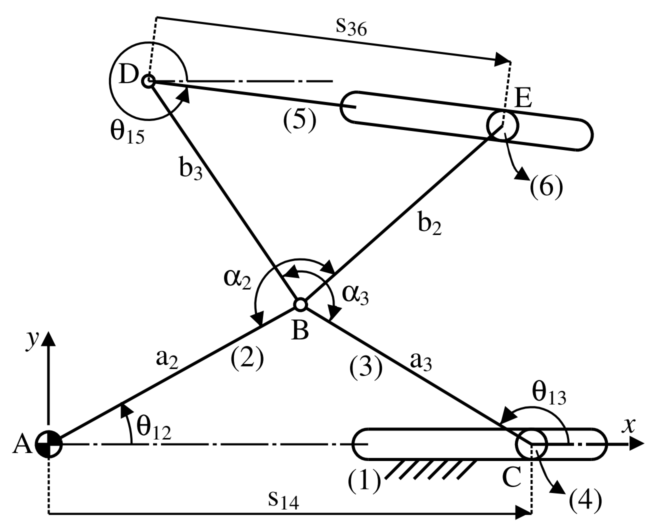


Figure 3.12 Schematic representation of the window regulator mechanism with joint variables

The window regulator mechanism is a single degree of freedom mechanism. Therefore, input joint variable is selected as, θ_{12} and the other dependent joint variables are found depending on this input joint variable.

There are two independent loops in the window regulator mechanism. So, two loop closure equations should be solved in order. First loop closure equation can be written using centric slider crank portion of the mechanism.

$$\overrightarrow{AB} = \overrightarrow{AC} + \overrightarrow{CB} \tag{3.25}$$

$$a_2 \cdot e^{i\theta_{12}} = s_{14} + a_3 \cdot e^{i\theta_{13}} \tag{3.26}$$

Equation (3.26) is a vector equation and can be solved as;

$$s_{14} = a_2 \cdot \cos\theta_{12} + \sqrt{a_2^2 \cdot \cos^2\theta_{12} + a_3^2 - a_2^2} \tag{3.27}$$

$$\theta_{13} = \operatorname{atan}_2(a_2 \cdot \cos\theta_{12} - s_{14}\ ,\ \ a_2 \cdot \sin\theta_{12}) \tag{3.28}$$

When $a_2 = a_3 = r$, Equations (3.27) and (3.28) become;

$$s_{14} = 2 \cdot r \cdot \cos\theta_{12} \tag{3.29}$$

$$\theta_{13} = \pi - \theta_{12} \tag{3.30}$$

Second loop can be written from the upper portion of the mechanism (Figure 3.13).

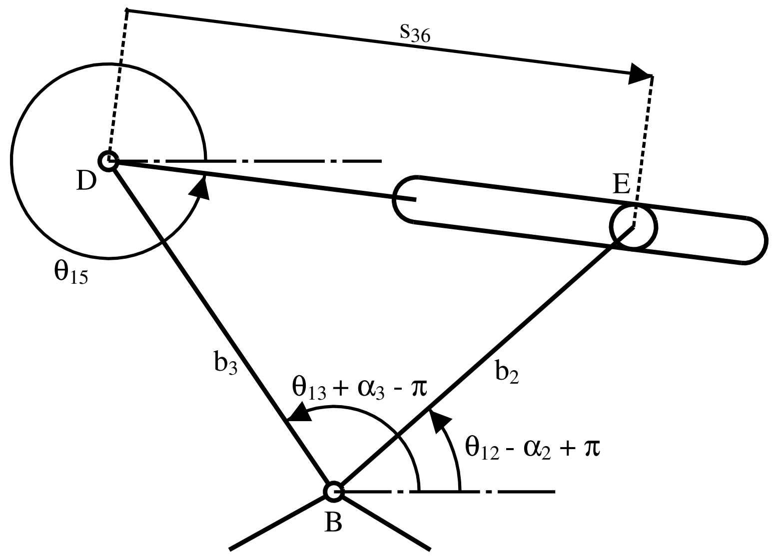


Figure 3.13 Schematic representation of the upper portion of the window regulator mechanism

$$\overrightarrow{BD} + \overrightarrow{DE} = \overrightarrow{BE} \tag{3.31}$$

$$b_3 \cdot e^{i(\theta_{13} + \alpha_3 - \pi)} + s_{36} \cdot e^{i\theta_{15}} = b_2 \cdot e^{i(\theta_{12} - \alpha_2 + \pi)} \tag{3.32}$$

Equation (3.32) can be solved for unknowns as;

$$s_{36} = \sqrt{b_2^2 + b_3^2 - 2 \cdot b_2 \cdot b_3 \cdot \cos(\theta_{13} + \alpha_3 - \theta_{12} + \alpha_2)} \tag{3.33}$$

$$\theta_{15} = \operatorname{atan}_2 \begin{bmatrix} b_2 \cdot \cos(\theta_{12} - \alpha_2 + \pi) - b_3 \cdot \cos(\theta_{13} + \alpha_3 - \pi), \\ b_2 \cdot \sin(\theta_{12} - \alpha_2 + \pi) - b_3 \cdot \sin(\theta_{13} + \alpha_3 - \pi) \end{bmatrix} \tag{3.34}$$

When $a_2 = a_3 = r$, Equations (3.33) and (3.34) become;

$$s_{36} = \sqrt{2 \cdot r^2 \cdot \left[1 - \cos(\theta_{13} + \alpha_3 - \theta_{12} + \alpha_2)\right]} \tag{3.35}$$

$$\theta_{15} = \text{atan}_2 \begin{bmatrix} r \cdot [\cos(\theta_{12} - \alpha_2 + \pi) - \cos(\theta_{13} + \alpha_3 - \pi)], \\ r \cdot [\sin(\theta_{12} - \alpha_2 + \pi) - \sin(\theta_{13} + \alpha_3 - \pi)] \end{bmatrix} \tag{3.36}$$

These equations can be used to find the joint variables for every crank angle.

3.5 Basic Restrictions on the Mechanism Design

Aside from the geometric restrictions discussed in section 3.3.1, there are also basic restrictions on the window regulator mechanism design. These are; limits on the placement of the fixed slide to the door chassis and restrictions on the velocity characteristic of the window.

3.5.1 Placement of the Fixed Slide to the Door Chassis

The mechanism is connected to the door chassis from fixed pivot and the fixed slide (Figure 3.14). Due to the car door geometry and obstacles in the car door, there may be some limitations on the placement of the fixed slide to the door chassis. Therefore, obtaining the distance of the fixed slide to the fixed pivot is useful while designing the mechanism.

The distance of the fixed slide to the fixed pivot can be found from the maximum and minimum of joint variable s_{14} ($|AC|$ in Figure 3.12). Maximum s_{14}, $s_{14,\max}$ occurs in the extended position of the centric slider-crank portion of the mechanism. Therefore,

$$s_{14,\max} = |AB| + |BC|$$

$$s_{14,\max} = a_2 + a_3 \tag{3.37}$$

Minimum s_{14}, $s_{14,\min}$ occurs at the top or lowest position of the mechanism depending of the designed mechanism (Figure 3.14). In the figure, $s_{14,\min}$ occurs at the lowest position of the mechanism.

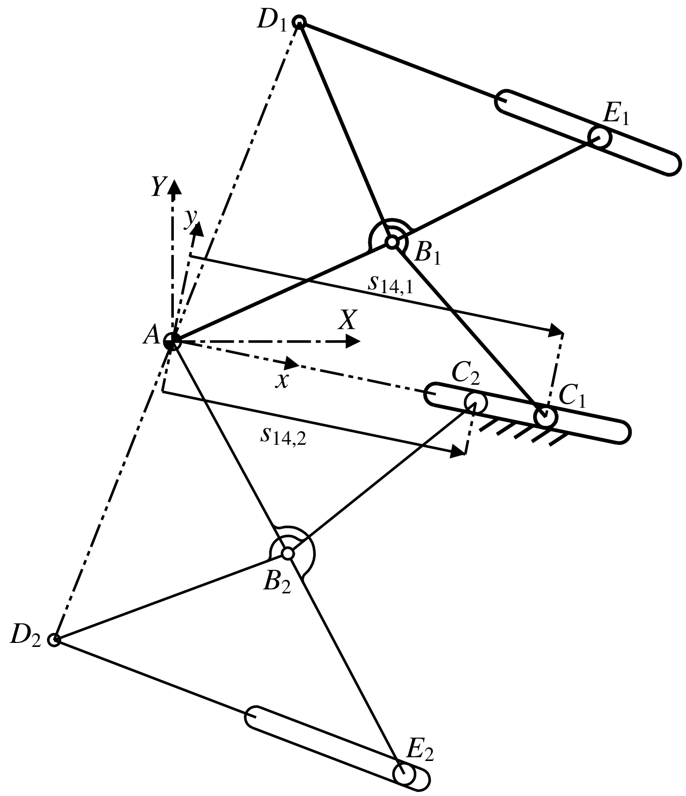


Figure 3.14 Schematic representation of the window regulator mechanism at the top and lowest position

$s_{14,\min}$ can be found using the equation of s_{14} derived in chapter 3.4 (Equation 3.29).

$$s_{14,\min} = \min\left(2 \cdot r \cdot \cos\theta_{12}\right) \tag{3.38}$$

Note that, both $s_{14,\max}$ and $s_{14,\min}$ are dependent on the designed mechanism parameters. Consequently, changing the design parameters, $s_{14,\max}$ and $s_{14,\min}$ can be arranged to find in a suitable position for the fixed slide.

3.5.2 Velocity Characteristic of the Window

Velocity characteristic of the window is important for the costumer. Generally, the velocity of the window should not deviate too much during operation. Therefore, this restriction should be considered during mechanism design.

It is known that window is rigidly attached to link (5) of the mechanism. Therefore, the velocity of link (5) should be found. Also, it is known that link (5) performs translational motion, so velocity of link (5) can be found using any point on it. In addition, the relevant velocity is the global vertical velocity of the window (along *Y*-axis in Figure 3.2). Thus, *Y*-axis component of the position of point *D*, Y_D can be written as (Figure 3.10),

$$Y_D = 2 \cdot r \cdot \cos(\pi/2 - \Phi - \Theta_{12}) \cdot \cos(\Phi) \tag{3.39}$$

Note that, lengths of scissor elements are equal to each other according to equation (3.16).

Vertical velocity of the window is found by taking the derivative of Equation (3.39) with respect to time as,

$$\dot{Y}_D = 2 \cdot r \cdot \sin(\pi/2 - \Phi - \Theta_{12}) \cdot \cos(\Phi) \cdot \dot{\Theta}_{12} \tag{3.40}$$

In Equation (3.40), $\dot{\Theta}_{12}$ is the angular velocity of the link (2). Assuming constant velocity for the input crank angle, $\dot{\Theta}_{12}$ can be obtained using the total opening or closing time of the window, t_{total} as,

$$\dot{\Theta}_{12} = \Theta_{12,total} \big/ t_{total} \tag{3.41}$$

From Equation (3.40), it can be observed that $\dot{Y}_D$ is a part of a sinusoidal function since the mechanism does not operate for full crank rotation. Also, this sinusoidal function is ranging from maximum and minimum of the input crank angle Θ_{12}.

Note that; from Equation (3.40), vertical velocity of the window is dependent on the designed mechanism parameters. As a result, changing the design parameters, vertical velocity of the window can be arranged in a desired way.

3.6 Force Analysis of the Mechanism

Force analysis of the window regulator mechanism is performed to obtain the driving torque of the mechanism which is applied to the input link. To obtain the driving torque, virtual work method is used. Because, virtual work method eliminates the necessity for solving multiple equations occurring in Newton-Euler approach and gives directly the external forces and torques acting on the system.

In addition, the virtual work method is applied neglecting the dynamic forces acting on the links. Because, accelerations of joint variables are relatively small therefore dynamic forces appears to be small when compared with others forces acting on the mechanism. Also, mass of sliders are neglected since their weights are very small when compared to other weights.

General equation for virtual work can be written as;

$$\delta U_{total} = \sum_j \vec{F}_j \bullet \delta\vec{r}_j + \sum_j \vec{T}_j \bullet \delta\vec{\theta}_j = 0 \tag{3.42}$$

In equation (3.38); $\vec{F}_j$ and $\vec{T}_j$ are external forces and torques on the mechanism, $\delta\vec{r}_j$ and $\delta\vec{\theta}_j$ are virtual displacements.

In Figure 3.15, a schematic representation of the mechanism with external forces and torques is shown.

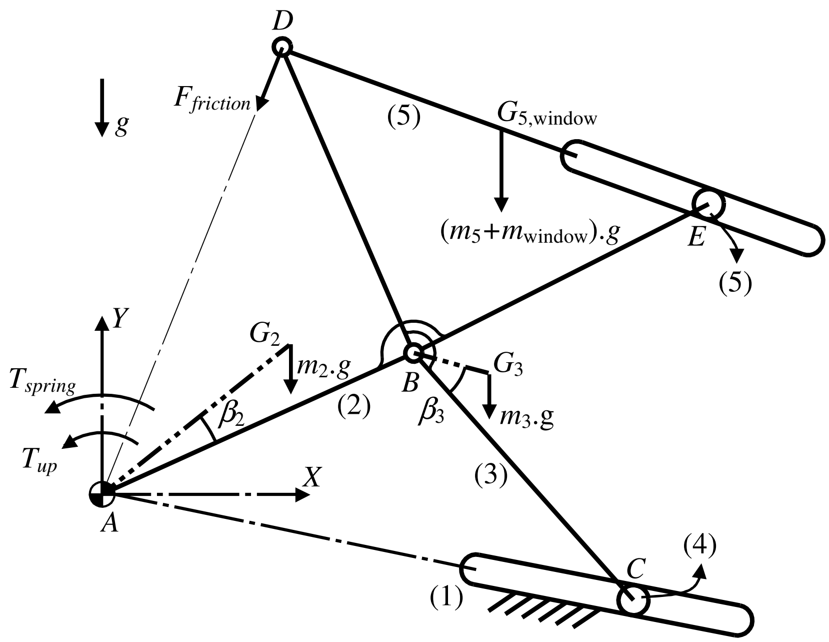


Figure 3.15 External forces and torques acting on window regulator mechanism while the mechanism is going upwards

G_2 and G_3 are center of gravities of links (2) and (3) and m_2 and m_3 are weights of links (2) and (3). Since link (5) and window are rigidly attached to each other, they are treated as a single part. Thus, $G_{5,window}$ is the center of gravity of link (5) and window, also m_5 and m_{window} are weights of link (5) and window.

$F_{friction}$ is the total friction force applied to the window by the plastic guides. It is taken as a single friction force at point *D*. In addition, joint frictions are neglected assuming these frictions are relatively small when compared with gravitational forces.

T_{up} is the torque applied to link (2) by the motor while the mechanism going upwards. T_{down} is the torque applied to link (2) by the motor while the mechanism going downwards and is opposite to T_{up}.

To balance the driving torque, a spiral spring can be applied to link (2) at the fixed point. So, T_{spring} is the torque applied to link (2) by spiral spring. The application of this spiral spring can be found in following chapters.

In Figure 3.16, virtual displacements of points of application of external forces and torques are shown.

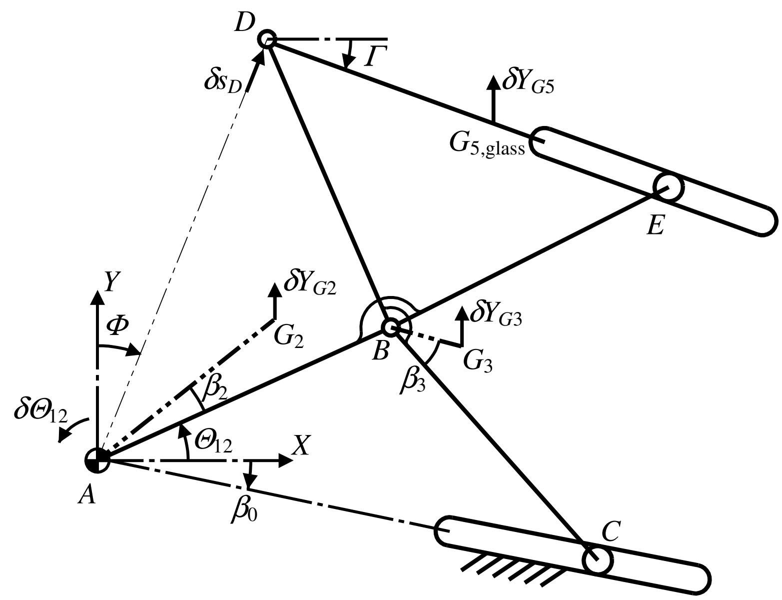


Figure 3.16 Virtual displacements of points of application of external forces and torques

Using equation (3.38), while the mechanism is going upwards virtual work expression can be re-written as;

$$
\begin{aligned}
0 = T_{up}.\delta\ \Theta_{12} + T_{spring}.\delta\ \Theta_{12} - m_2.g.\delta Y_{G2} - m_3.g.\delta Y_{G3} \\
-(m_5 + m_{glass}).g.\delta Y_{G5} - F_S.\delta s_D
\end{aligned}
\tag{3.43}
$$

Using Equation (3.43), T_{up} can be found as;

$$T_{up} = -T_{spring} + m_2.g.\frac{\delta Y_{G2}}{\delta\ \Theta_{12}} + m_3.g.\frac{\delta Y_{G3}}{\delta\ \Theta_{12}} + (m_5 + m_{glass}).g.\frac{\delta Y_{G5}}{\delta\ \Theta_{12}} \\ +F_S \cdot \frac{\delta s_D}{\delta\ \Theta_{12}} \tag{3.44}$$

Similarly, while the mechanism is going downwards;

$$0 = -T_{down}.\delta\ \Theta_{12} + T_{spring}.\delta\ \Theta_{12} - m_2.g.\delta Y_{G2} - m_3.g.\delta Y_{G3} \\ -(m_5 + m_{glass}).g.\delta Y_{G5} + F_S.\delta s_D \tag{3.45}$$

Using Equation (3.45), T_{down} can be found as;

$$T_{down} = T_{spring} - m_2.g.\frac{\delta Y_{G2}}{\delta\ \Theta_{12}} - m_3.g.\frac{\delta Y_{G3}}{\delta\ \Theta_{12}} - (m_5 + m_{glass}).g.\frac{\delta Y_{G5}}{\delta\ \Theta_{12}} \\ +F_S \cdot \frac{\delta s_D}{\delta\ \Theta_{12}} \tag{3.46}$$

To obtain the virtual displacements, first displacements of relevant points are written,

$$Y_{G2} = |AG_2| \cdot \sin(\Theta_{12} + \beta_2) \tag{3.47}$$

$$Y_{G3} = |AB| \cdot \sin\Theta_{12} - |BG_3| \cdot \sin(\Theta_{12} + 2 \cdot a_0 - \beta_3) \tag{3.48}$$

$$Y_{G5} = |AB| \cdot \sin\Theta_{12} + |BD| \cdot \sin(a_3 - \Theta_{12} - 2 \cdot a_0) - |DG_5| \cdot \sin\Gamma \tag{3.49}$$

$$s_D = 2 \cdot |AB| \cdot cos(\pi/2 - \Theta_{12} - \Phi) \tag{3.50}$$

Then virtual displacements are obtained as,

$$\delta Y_{G2} = |AG_2| \cdot cos(\Theta_{12} + \beta_2) \cdot \delta\ \Theta_{12} \tag{3.51}$$

$$\delta Y_{G3} = \left(|AB| \cdot cos\Theta_{12} - |BG_3| \cdot cos(\Theta_{12} + 2 \cdot a_0 - \beta_3)\right) \cdot \delta\ \Theta_{12} \tag{3.52}$$

$$\delta Y_{G5} = \left(|AB| \cdot \cos\Theta_{12} - |BD| \cdot \cos(a_3 - \Theta_{12} - 2 \cdot a_0)\right) \cdot \delta\,\Theta_{12} \tag{3.53}$$

$$\delta s_D = 2 \cdot |AB| \cdot \sin(\pi/2 - \Theta_{12} - \Phi) \cdot \delta\,\Theta_{12} \tag{3.54}$$

Rearranging Equations (3.51) to (3.54),

$$\frac{\delta Y_{G2}}{\delta\,\Theta_{12}} = |AG_2| \cdot \cos(\Theta_{12} + \beta_2) \tag{3.55}$$

$$\frac{\delta Y_{G3}}{\delta\,\Theta_{12}} = |AB| \cdot \cos\Theta_{12} - |BG_3| \cdot \cos(\Theta_{12} + 2 \cdot a_0 - \beta_3) \tag{3.56}$$

$$\frac{\delta Y_{G5}}{\delta\,\Theta_{12}} = |AB| \cdot \cos\Theta_{12} - |BD| \cdot \cos(a_3 - \Theta_{12} - 2 \cdot a_0) \tag{3.57}$$

$$\frac{\delta s_D}{\delta\,\Theta_{12}} = 2 \cdot |AB| \cdot \sin(\pi/2 - \Theta_{12} - \Phi) \tag{3.58}$$

Finally, Equations (3.44) and (3.46) can be used with Equations (3.55) to (3.58) to obtain T_{up} and T_{down}.

3.7 Design of the Spiral Spring

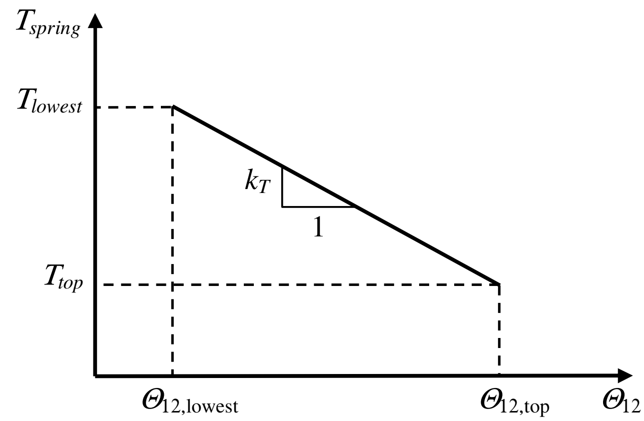
The spiral spring is applied at the fixed pivot and used to balance the driving torque. Because while opening or closing the window, two different torque characteristics occurs for the driving torque (Equations (3.44) and (3.46)).Using a spiral spring these torques can be adjusted to acceptable values.

Assuming the behavior of the spiral spring is linear, the torque applied by the spring can be written as;

$$T_{spring} = k_T \cdot \left(\Theta_{initial} + (\Theta_{12,top} - \Theta_{12})\right) \tag{3.59}$$

In Equation (3.59), k_T is the torsional stiffness of the spiral spring, $\Theta_{initial}$ is the initial compression of the spring, $\Theta_{12,top}$ is the designed maximum value of the crank angle and Θ_{12} is the crank angle for the current position of the mechanism.

The linear behavior of the spiral spring is shown in Figure (3.17).



3.17 Behavior of the spiral spring

Note that, torsional stiffness and initial compression of the spiral spring are design parameters. Therefore, these two design parameters can be changed to obtain the suitable driving torque characteristics.

CHAPTER 4

IMPLEMENTATION OF THE DESIGN

4.1 Introduction

In this chapter, implementation of the design is presented. The implementation is carried out for a RENAULT car door as a part of a study made with OYAK-RENAULT. A car door chassis with necessary parts is supplied by OYAK-RENAULT. In addition, CAD data of the car door is provided. Therefore, a prototype of aforementioned window regulator design is implemented on this car door.

Firstly, kinematic synthesis of the window regulator mechanism is done. Then, embodiment of the mechanism is performed to obtain the general dimensions of parts which will be manufactured. Afterwards, force analysis and spiral spring design is carried out. Lastly, manufacturing and assembly of the design is handled.

Note that, throughout the whole chapter, detailed dimensions are not given according to privacy policies. For that reason, only results of the study are presented.

4.2 Kinematic Synthesis

As a first step of the kinematic synthesis, geometric constraints for the mechanism are found according to CAD data. In addition, a suitable fixed pivot and fixed slide position is selected in the car door chassis considering possible obstacles. Then, according to the basic restrictions on the mechanism, free design parameter is arranged to obtain the window regulator mechanism. The schematic representation of the resultant mechanism is given in Figure 4.1.

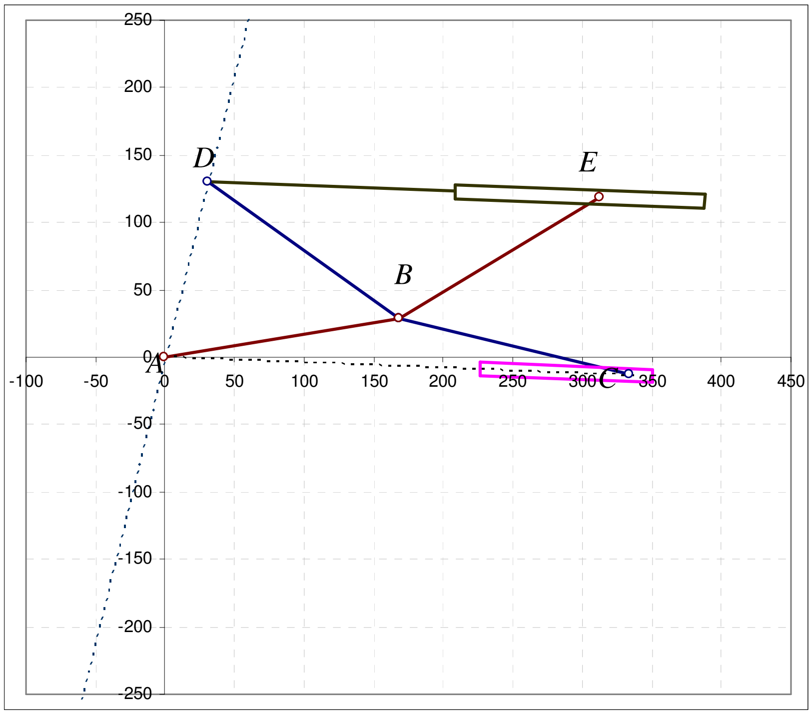


Figure 4.1 Schematic representation of the resultant mechanism

In Figure 4.2, schematic representation of the resultant mechanism at the top and lowest positions is shown.

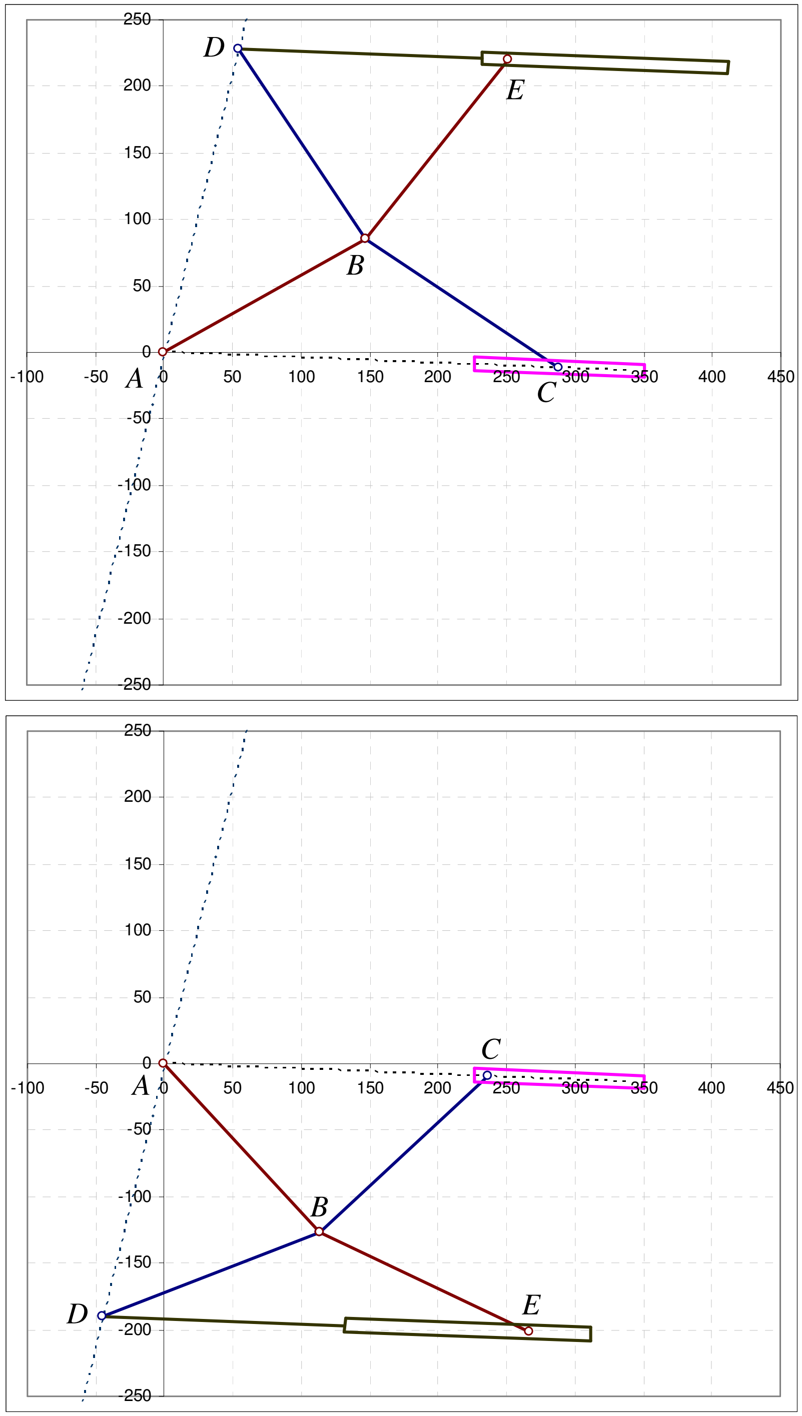


Figure 4.2 Schematic representation of the resultant mechanism at the top and lowest positions

Angular velocity of the input crank angle is found using a selected total opening time for the window. Then, velocity and acceleration of the window is obtained using this angular velocity of the input crank angle and result is shown in Figure 4.3. As observed, the acceleration of the window is relatively low.

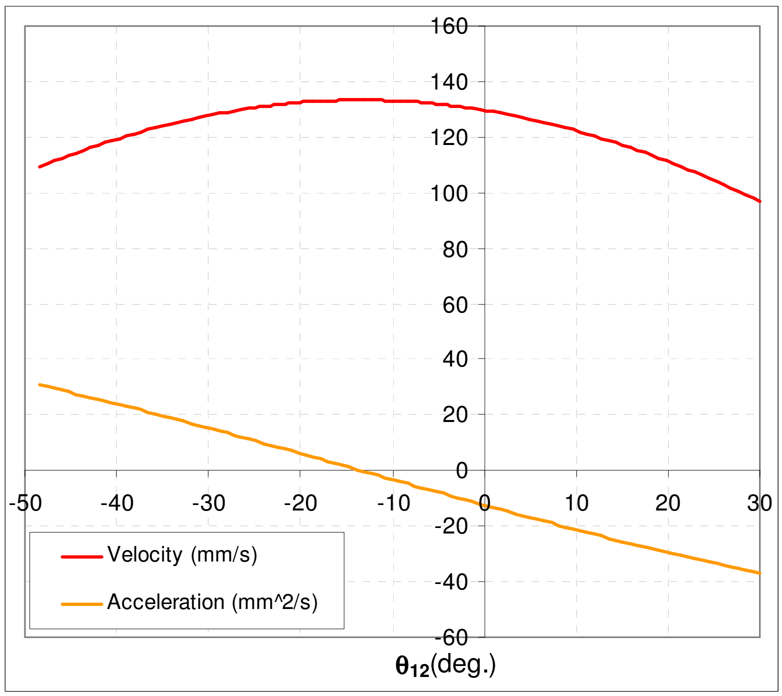


Figure 4.3 Velocity (mm/s) and acceleration (mm^2/s) of the window with respect to input crank angle (deg.)

4.3 Embodiment of the Design

After obtaining the window regulator mechanism, necessary parts are shaped in detail using a CAD program. While shaping the necessary linkages, the first concern is the strength of the parts. However, knowing that the mechanism does not work under the effect of large forces, detailed strength analysis is not performed. In addition, the physical connections between the linkages are considered in the embodiment design. Joints are arranged to reduce the friction between the linkages of the prototype. Also, manufacturing limitations play an important role while shaping the parts. Some dimensions are rearranged because of this restriction. Lastly, fixed parts are designed according to the possible connection positions to the car door chassis. In Figure 4.4, a CAD software view of the assembled window regulator is given.

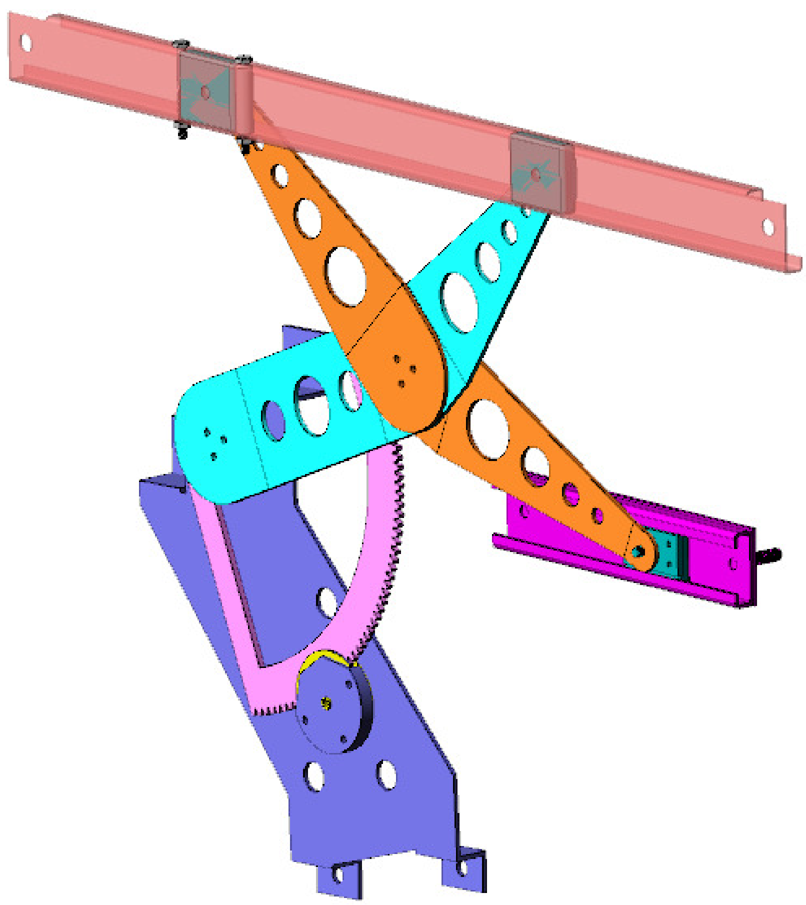


Figure 4.4 CAD software view of the assembled window regulator

Also, animation of the assembly is performed using a CAD software to validate the design. In Figure 4.5, CAD software view of the top and lowest positions of the window regulator is shown.

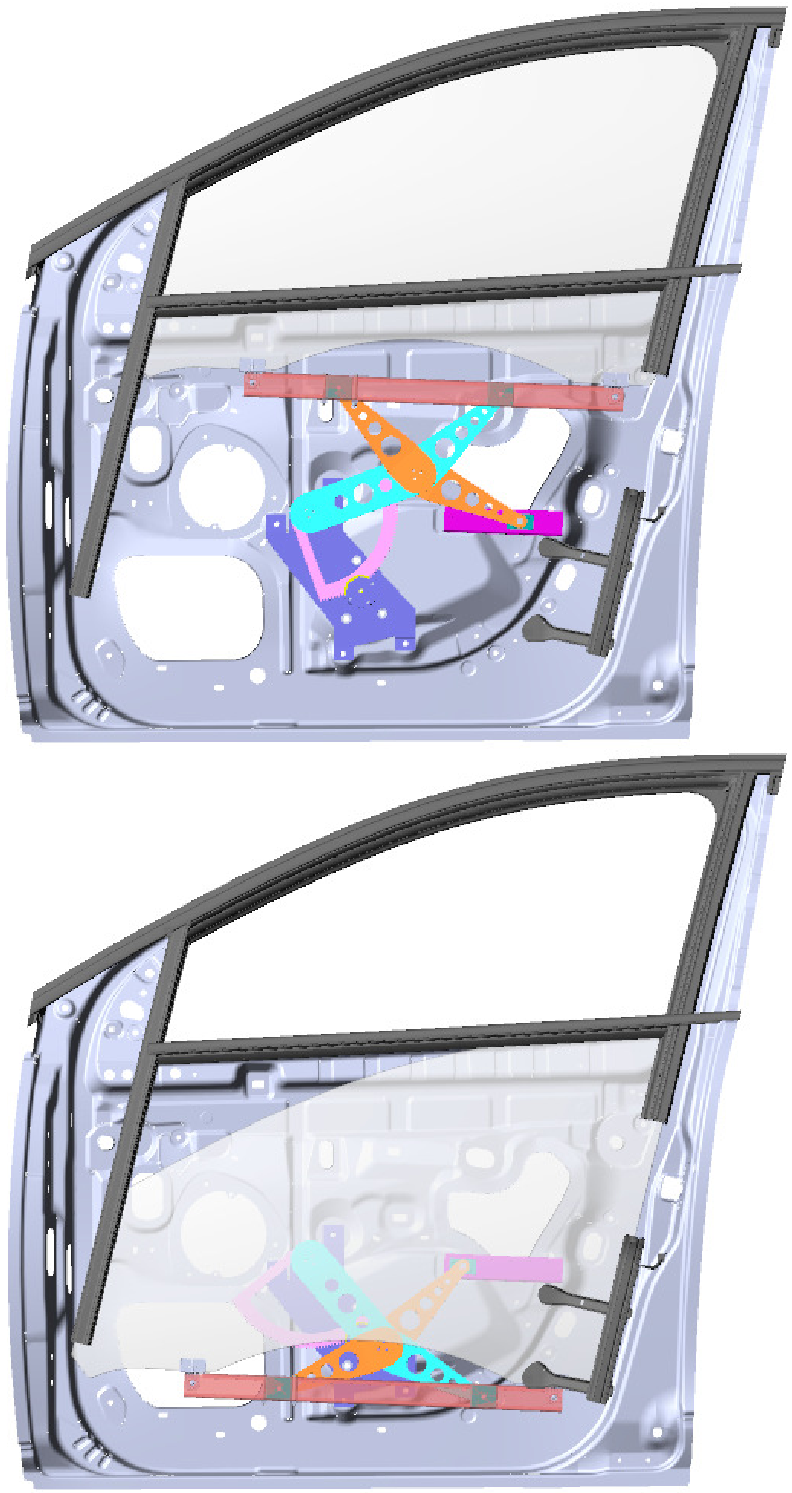


Figure 4.5 CAD software views of the top and lowest positions of the window regulator

4.4 Force Analysis and Spiral Spring Design

After the embodiment design, weights of the links are obtained. Therefore, force analysis can be done to obtain the driving torque of the mechanism. But, first the frictional force applied to the window must be known. So, frictional force is predicted by simple experiments.

In Figure 4.6, required driving torque of the mechanism with respect to the input crank angle without spiral spring is shown.

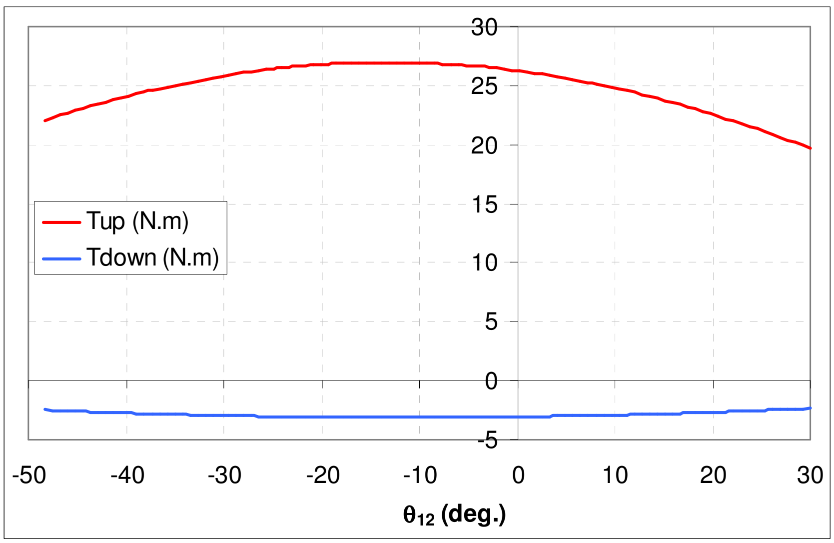


Figure 4.6 Driving torque (N.m) without spiral spring with respect to the input crank angle (deg,) while the window is going upwards or downwards

As observed from the Figure 4.6; while the window is going downwards, the required driving torque is negative. Thus; to balance the driving torque, a spiral spring is designed and the required driving torque of the mechanism with respect to the input crank angle is given in Figure 4.7.

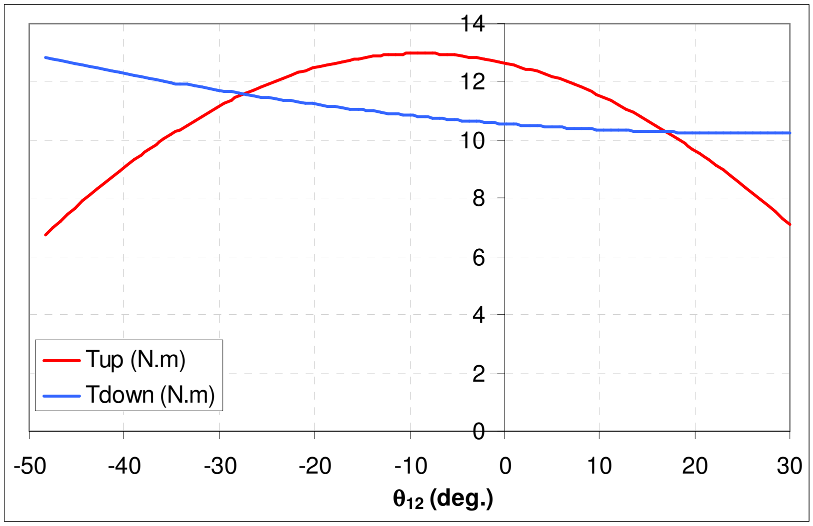


Figure 4.7 Driving torque (N.m) with spiral spring with respect to the input crank angle (deg,) while the window is going upwards or downwards

After implementing the spiral spring, the maximum required torque is dropped to nearly its half value. Therefore, motor selection is made according to these driving torque values.

4.5 Manufacturing of the Prototype

The designed window regulator mechanism is composed of more than 60 parts including the bolts and nuts (Figure 4.8). As mentioned before, most of the parts of the mechanism are custom designed and then manufactured using suitable manufacturing techniques.

To have precisely manufactured products, most of the parts are manufactured using laser cutting. Bending of the parts is made using automatic bending machines. In addition, some additional operations are made on these parts by machining like drilling of holes. Furthermore, some of the simple parts are manufactured using classic machining techniques. Also, CNC machining is used to produce some complex parts like sliders of the mechanism.

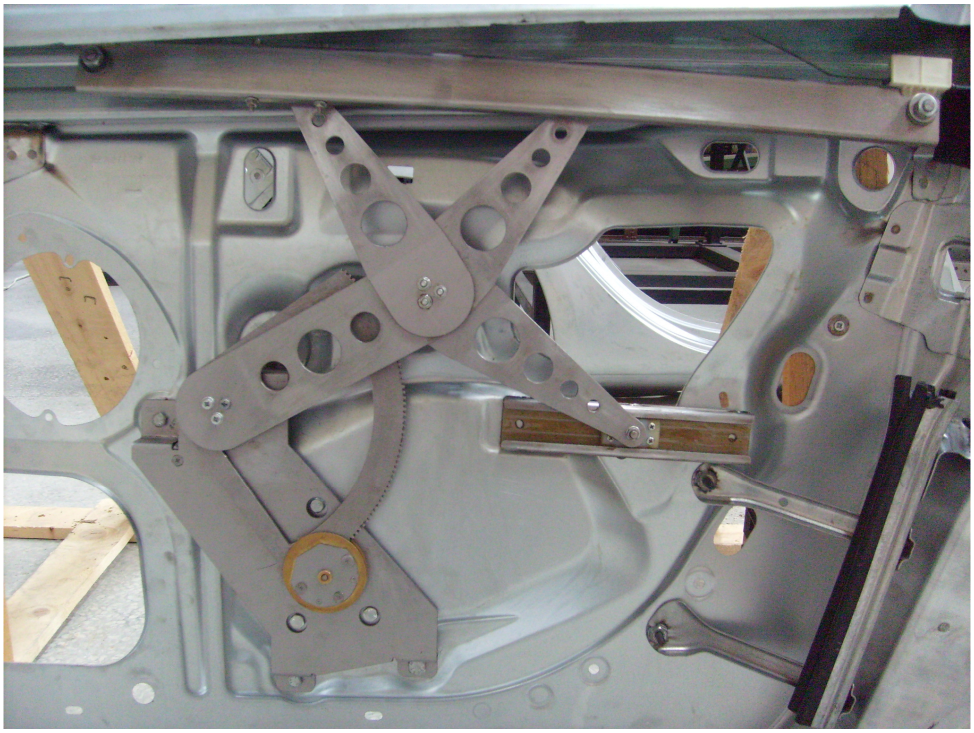


Figure 4.8 Manufactured window regulator prototype on the car door chassis

Most of the parts are made of steel however brass parts are used to reduce the friction in the joints. Also, sliders are manufactured from polyamide to decrease the friction in the sliding joints.

The manufactured parts are assembled to form the window regulator and the assembled prototype is tested on the car door chassis using a standard window regulator motor. The results are satisfactory and the window regulator is operated properly on the car door.

The assembled prototype is shown at top and lowest position in Figure 4.9. Also, in Figure 4.10 a close-up view at the top and lowest position is shown.

Figure 4.9 Assembled window regulator prototype on the car door chassis at the top and lowest positions

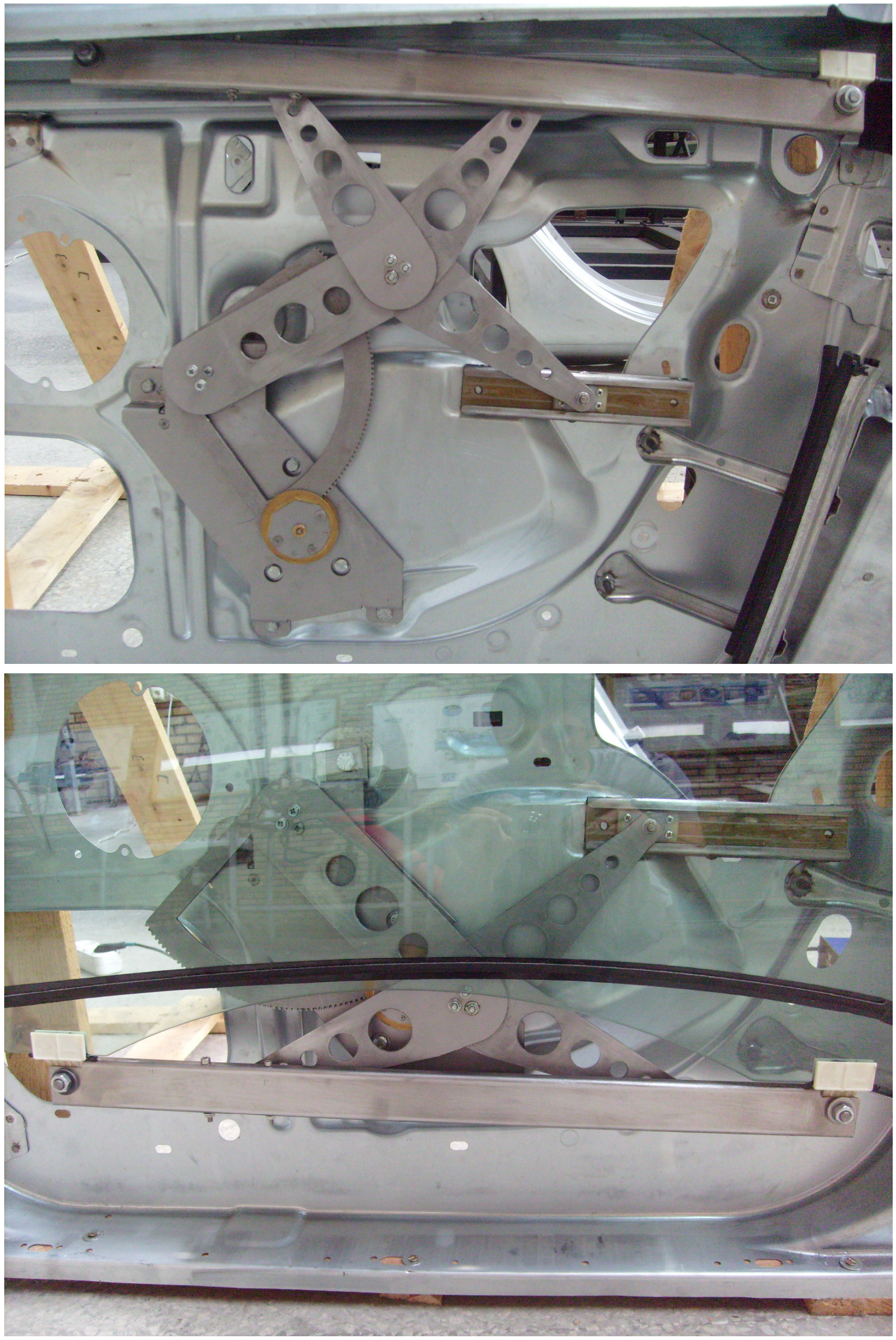


Figure 4.10 A close-up view of the assembled window regulator prototype on the car door chassis at the top and lowest positions

CHAPTER 5

DISCUSSIONS AND CONCLUSIONS

In this thesis, design of a car door window regulator is presented. Firstly, in Chapter 1, an introduction to window regulators are made and different types of window regulator mechanisms are presented. In Chapter 2, conceptual design of the window regulator is performed and a best concept is selected. In Chapter 3, detailed design of the selected window regulator concept is accomplished. In Chapter 4, the implementation of this design to a RENAULT car door is presented.

During the kinematic synthesis of the window regulator mechanism (Chapter 3.3), free design parameters are selected as position of the fixed pivot, location of the fixed slide relative to fixed pivot and input crank angle at the top position of the window. Here, an optimization study can be carried out by changing these free parameters as a future study. Then, improvements in the required driving torque or weights of parts can be achieved. However, in this case changes on the car door chassis must be made to fix the window regulator onto it.

The embodiment design of the window regulator prototype (Chapter 4.3) is affected mainly by the restrictions due to the limited manufacturing techniques. Therefore, different embodiment designs can be made for mass production of the window regulator using better manufacturing techniques. Also, the connections at the joints can be redesigned to suit the mass production process.

Detailed strength analyses are not carried out during the detailed design stage because forces acting on the links of the mechanism are relatively small. But, to improve the design, detailed strength analyses can be performed. Also an optimization study for detailed strength analyses can be made considering weights of the linkages of the window regulator mechanism. In this manner, overall weight of the mechanism can be reduced. In addition, improvements in the required driving torque can be achieved.

REFERENCES

[1] Kraemer, W. L., *Window Opener*, Patent No: US1695691A, 1928.

[2] Szkozinski, A., Direct Drive Vehicle Window Regulator, Patent No: CA2510712A1, 2006.

[3] Bell, T. H., Schoenleber, J. G., *Window Closure Device*, Patent No: US1356123A, 1920.

[4] Rietdijk, D., *Window Lifter for a Motor Vehicle*, Patent No: US2008155901A1, 2008.

[5] Eckey, F., I*mprovements in and relating to Lazy Tongs Apparatus for Opening and Closing Windows and the Like*, Patent No: GB191306793A, 1914.

[6] Walters, J. E., *Apparatus for Operating Sliding Windows*, Patent No: GB2207185A, 1989.

[7] Cousinard, A. A., *Device for Raising and Lowering Carriage Windows and the Like*, Patent No: GB157110A, 1921.

[8] Kuki, T. N., Isomura, C. Y., Suzumura, T. H., Sakakibara, A. Y., Ishihara, K. H., *Window Regulator Apparatus*, Patent No: US5009035A, 1991.

[9] Medebach, T., *Cable Operated Window Lifter*, Patent No: EP0706603B1, 1994.

[10] Seegers & Sohn, F., *Schiebefenster, insbesondere für Wagen, mit Zahnradantrieb und einem mit diesem verbundenen Stellhebel und einer Vorrichtung zum Übersetzen des Fensters über die Wetterleiste*, Patent No: DE350138C, 1922.

[11] Paul, S., *Window Regulator Means*, Patent No: US1840465A, 1932.

[12] Ternstedt Manufacturing Company, *Fensteranordnung für Kraftfahrzeuge mit zwei in einer Fensteröffnung angeordneten Scheiben*, Patent no: DE640437C, 1937.

[13] Dupuy, R. E., *Vehicle Door Glass Regulator*, Patent no: US5255470, 1993.

[14] Kriese, O., *Die folgenden Angaben sind den vom Anmelder eingereichten Unterlagen etnommen*, Patent no: DE10344286A1, 2005.

[15] Bottema, 0., Roth, B., *Theoretical Kinematic*s, North-Holland Publ. Co., New York, 1979, pp. 346-348.

[16] Söylemez, E., *Centrodes of Plane Motion Mechanisms* Section 1.3 of ME519Lecture Notes, Middle East Technical University, Ankara, 2008.

[17] Pahl G., Beitz W., *Engineering Design: A Systematic Approach*, The Design Council, London, 1984, pp. 57-151