



# **Design and Simulation of Tail Lifts for Truck**

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Bachelor's Thesis

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<p>Abstract</p> <p>Tail lifts are widely used in the EU and USA. They are the most common equipment fixed on trucks which are used for loading and unloading cargo because they make transportation to be more economical, more efficient and safer. With the rapid development of China economy and logistics industry, tail lifts will certainly have a significant development in Chinese market.</p> <p>The mechanical system is certainly the most important part of tail lift because it is carrying the load. So the safety of the system should be guaranteed. The goal of the thesis is to design a reasonable and safe mechanical system of tail lift for a certain type of truck.</p> <p>Studying the principle mechanism was the first step. When this process was finished, creating the models for all components in Inventor system began. After completing the models, the dynamic simulation and stress analysis were conducted. Finally, according to the results of stress analysis, the revised model was created.</p> <p>There are two sets of mechanical systems in this thesis. The first set has some disadvantages which result in large stress in the system. The modified set improved the strength and made the maximum stress and deformation in limit. The whole mechanical system of tail lift has been accomplished.</p>			
<p>Keywords</p> <p>tail lift, four-bar linkage, Inventor, stress analysis, simulation</p>			
Confidentiality			

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## DEFINITION OF ITEMS

Mechanism – A mechanical device which has the purpose of transferring motion and/or force from the source to the output.

Planar four-bar linkage – The simplest type of closed loop linkage which consists of three mobile links and a fixed link.

Bars/Links – Individual parts/components of a mechanism which connected to one another through pin joints ensure a relative motion of rotation.

Parallelogram mechanism – One kind of planar four-bar linkage in which four links make a parallelogram and can realize translational motion.

Kinematics analysis – A method which create the desired motions of the subject mechanical parts and then mathematically compute the positions, velocities, and accelerations that those motions will create on the parts.

Vector loop equation for a four-bar linkage – An alternate approach to linkage position analysis creates a vector loop around the linkage as first proposed by Raven. The links are represented as position vectors.

Derivation – A derived part is a new part that references an existing part to copy bodies and other information such as sketches, work features, and parameters associatively.

Finite Element Method (FEM) – A numerical technique for finding approximate solutions to boundary value problems.

Factor of Safety (FOS) – A term describing the structural capacity of a system beyond the expected loads or actual loads.

Stress analysis – An engineering discipline covering methods to determine the stresses and strains in materials and structures subjected to forces or loads.

Von Mises Stress – An item determining whether an isotropic and ductile metal will yield when subjected to a complex loading condition.

## 1 INTRODUCTION

### 1.1 Background

Transportation is seen as one of the basic human needs and has a significant impact on a country's economy; productivity usually correlates well with the amount of transportation of goods and people. Transportation takes place on the ground, sea, and in the air and can be subdivided into the areas automotive, railway, naval, and aerospace.

The development of logistics and transportation is expanding with an incredible speed in the recent decennium, especially from 2003 to 2011, the highways total was lengthen form 1,809,900 km to 4,106,400 km, in addition, there were 6,406,600 transport vehicles until the end of 2006. Nevertheless, China's logistic systems have generally inferior technology and substandard equipment, especially in the aspect of handling appliances; there are manual handling cars, hand push forklifts and simply normal lifting appliances still accounted for more than 70%.

With expansion of total transportation, enormous potential of logistic market, and ascendant of third-party logistics, numerous and high technologic logistic equipment and systems were definitely needed to promotes the higher work efficiency, more company benefit, and less labor.

With the development of China's economy, modern logistics and transportation sector has also been a rapid development, as one of the essential equipment of modern cargo transport vehicles, vehicle tail lift, has also got the attention of a lot of research departments. The tail lift is the lift which is equipped on the stern of truck, it is able to automatically lift as well as able to load and unload. It is also a highly regulated piece of ancillary equipment for fitment to a vehicle. One kind of tail lift is shown in Figure 1.1. Tail lift designers have a legal duty to ensure that tail lifts are constructed and installed to appropriate standards and that they are supplied with information to allow them to be used safely. Cargo tail lift, can not only greatly improve the handling efficiency, saving human resources, but also can improve the handling personnel security as well as reduce cargo damage rate.

Locomotion of a mechanical system consisting of two rigid bodies, a main body and a tail, connected by a cylindrical joint, is considered. The system moves in a resistive

fluid and is controlled by periodic angular oscillations of the tail relative to the main body.



*Figure 1.1 Tail lift*

## 1.2 Development of tail lift

The first generation was produced in the end of 1930s, which had single cylinder for lift and manual operation for overturn. The maximum load is approximate 500 kg, and the dip angle between breast board and ground is  $9^{\circ}\sim 10^{\circ}$ . Until now, this type production is on the active service in the region of Southeast Asia and Japan. [1, P.35]

The second generation was produced in European market since the beginning of 1950s. This production is the improvement of the first generation, depending on that; people installed the hydro-cylinder for flipping closed. Therefrom, lift and turn were realized through two single cylinders separately. The operator turned over the board by experience. The usual types were one which had four hydro-cylinders, and the other had two hydro-cylinders. The loading limitation was over 500 kg, and dig angle was approximate  $10^{\circ}$ . This type of production was mainly applied in United States and Southeast Asia. [1, P.35]

After twenty years of development, people added the fifth cylinder on the second generation production. This fifth cylinder took effect of “memory”, controlling the turning motion of tail lift automatically. In this way, the “memory” leads the lift process to be more safe and smooth. The dig angle usually was  $8^{\circ}\sim 10^{\circ}$ , in addition, if it doubles as the door, the dig angle could also be less than  $8^{\circ}$ . In present, this kind of production is commonly applied in Europe and America. [1, P.35]

The fourth generation appeared in the early nineties and had the same principle of cylinder system with previous. The only difference is the increment of memorial range of motion. On the other hand, the biggest improvement of new design was a special structure on the carrier platform, changed from the two-part activities connection to one-part. The platform can not only automatically flip, but also touch down like a sinking action in order to decrease the dig angle to  $6^{\circ}$ , or even less than  $6^{\circ}$ . [5]

Overall, these four generations of products in the worldwide market had been produced in China, but its development is still in the infancy segment. The expansion of the domestic market still needs the time and opportunities.

Considering about Chinese market, despite the development of logistics equipment in China had a late start, but made rapid process along with the rapid development of logistics industry currently. Vehicle tail lift as a light logistics equipment has huge market space in China. Comparing with European countries to China, the utilization rate is 60% versus 0.1%. The contrast of sales status between Europe and America and China during 2008~2010 is presented in Table 1.1. [2, P.114]

*Table 1.1 The contrast of sales status between Europe & America and China during 2008~2010 [2, P.114]*

<b>Year</b>	<b>2008</b>	<b>2009</b>	<b>2010</b>
<b>Europe &amp; America</b>	250,000	200,000	230,000
<b>China</b>	4,000	6,000	9,000

The sustained rapid development, especially the rapid development in the logistics and transport industry provide the extremely colossal space for vehicle tail lift gate. From Table 1.1 it was probably predicted that the demand of tail lift will extend at an annual rate of more than fifty percent. Four common categories of tail lift prevailing in European countries are shown in Figure 1.2. [2, P.114]





Figure 1.2 Four common categories of tail lift

### 1.3 Objectives of the thesis

The whole tail lift system is shown in Figure 1.3, it consists of the following components as **load platform** which is for supporting loads, **electronic control system** which consists of electric cabinet and sub-controller and controls all motions of tail lift, **oil sources** which consist of electrical motor, oil pump, hydraulic control valve, and fuel tank, which provides enough power and force to do all the work, **pipng system** and **transmission mechanism** which consists of hydraulic cylinders, rocker and square beam.

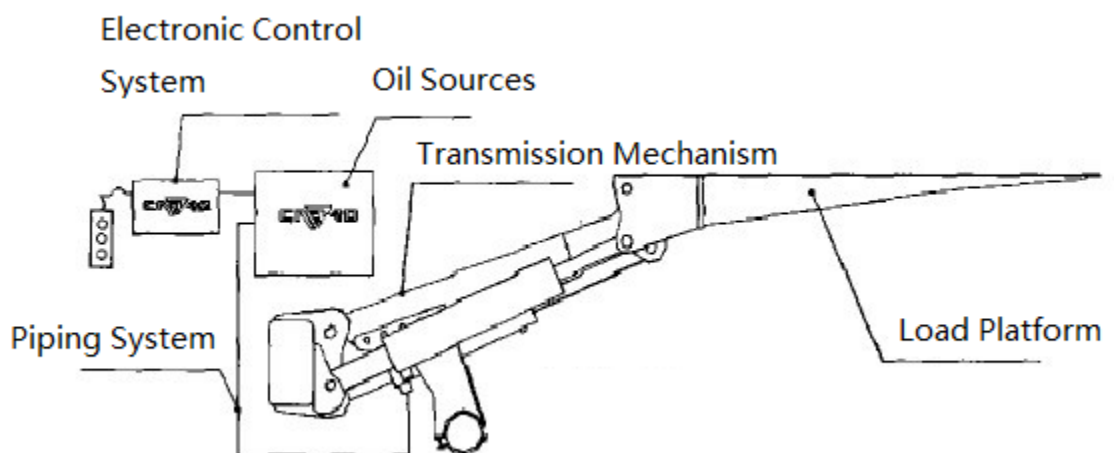


Figure 1.3 Tail lift system

The original objectives of this thesis about tail lifts are:

1. To be more economical: Installing a tail lift can greatly reduce the cost of employment.
2. To be more efficient: By using tail lift, loading and unloading can be completed easily, resources can be saved, work efficiency can be improved, and the economic performance of the vehicle can be developed.
3. To be safer: Using the tail lift makes the process of loading and unloading be safer since it can be done well without human collision damage. So it can avoid personal injury. Especially for the vehicle equipped with gas and dangerous liquid.

In this thesis, the following work should be done:

1. Analyzing the mechanical principle of tail lift.
2. Calculating the basic dimensions of tail lift for a special type of truck.
3. Creating the three-dimensional model with Inventor 2013.
4. Assembling the tail lift and checking its feasibility.

## 2 ANALYSIS OF MECHANICAL PRINCIPLE OF TAIL LIFT

### 2.1 Basic background knowledge used in tail lift

#### 2.1.1 Planar four-bar linkage

Various linkage mechanisms are widely applied in different areas, for example, deployable mechanism of artificial satellite solar panels, drive mechanism of the manipulator, human prosthesis and so on. It is well-known that planar four-bar linkage mechanical system, whose links perform specific oscillations relative to each other, can realize some kind of motion. It is the simplest closed-loop linkage, which has three movable links and four pin joints.

As shown in Figure 2.1, there is a revolute four-bar mechanism, which has base AD is fixed as **frame**. AB called **driver** or **input link** is acting on the middleware component BC to move CD (**follower** or **output Link**). Therefore people designated the link opposite the frame BC as **coupler link**, and the links which are hinged to the frame AB and CD are entitled as **side links**. [3, P.134]

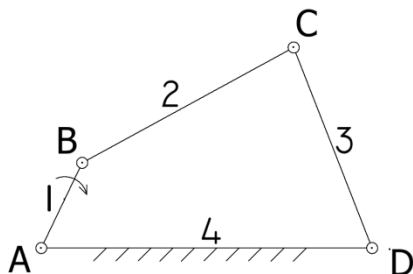


Figure 2.1 Closed-loop four-bar linkage mechanism

A Link which is free to rotate through 360 degree with respect to a second link will be said **revolve** relative to the second link (not necessarily a frame). Side links are divided into two groups, which one can revolved circularly to the frame named **crank**, the other cannot revolved named **rocker**.

There are four types of planar linkage mechanism. All of the present link mechanism evolved from three fundamental types, which respectively are:

If the shorter side link revolves and the other one rocks, it is called **crank-rocker mechanism** as shown in Figure 2.2; if both of the side links revolve, it is called **dou-**

**ble-crank mechanism** as shown in Figure 2.3, and if both of the side links rock, it is called **double-rocker mechanism** as shown in Figure 2.4.

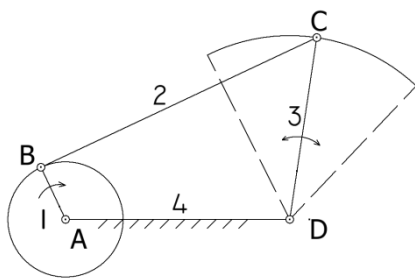


Figure 2.2 Crank-rocker mechanism

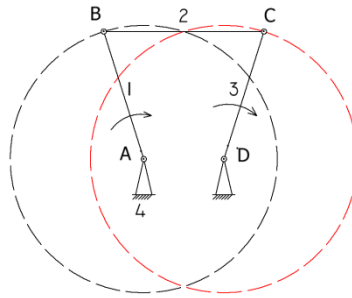


Figure 2.3 Double-crank mechanism

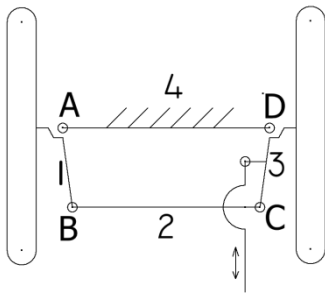


Figure 2.4 Double-rocker mechanism

### 2.1.2 Classification and Grashof's theorem

Assuming the frame is horizontal there are four possibilities for the input and output links:

A crank: can rotate a full 360 degrees;

A rocker: can rotate through a limited range of angles which does not include  $0^0$  or  $180^0$ ;

A 0-rocker: can rotate through a limited range of angles which includes  $0^0$  but not  $180^0$ ;

A  $\pi$ -rocker: can rotate through a limited range of angles which includes  $180^0$  but not  $0^0$ . [2]

The **Grashof condition** for a four-bar linkage states: if the sum of the shortest and longest link of a planar quadrilateral linkage is less than or equal to the sum of the remaining two links, then the shortest link can rotate fully with respect to a neighboring link. In other words, the condition is satisfied if [4]

$$s + l \leq p + q$$

s = length of shortest bar;

$l$  = length of longest bar;  
 $p, q$  = lengths of intermediate bar.

Table 2.1. Classification of Four-bar Mechanisms [4]

Case	$s+l$ v.s. $p+q$	Shortest Bar	Type
1	<	Frame	Double-crank
2	<	Side link	Crank-rocker
3	<	Coupler	Double-Rocker
4	=	Any	Change Point
5	>	Any	Double-rocker

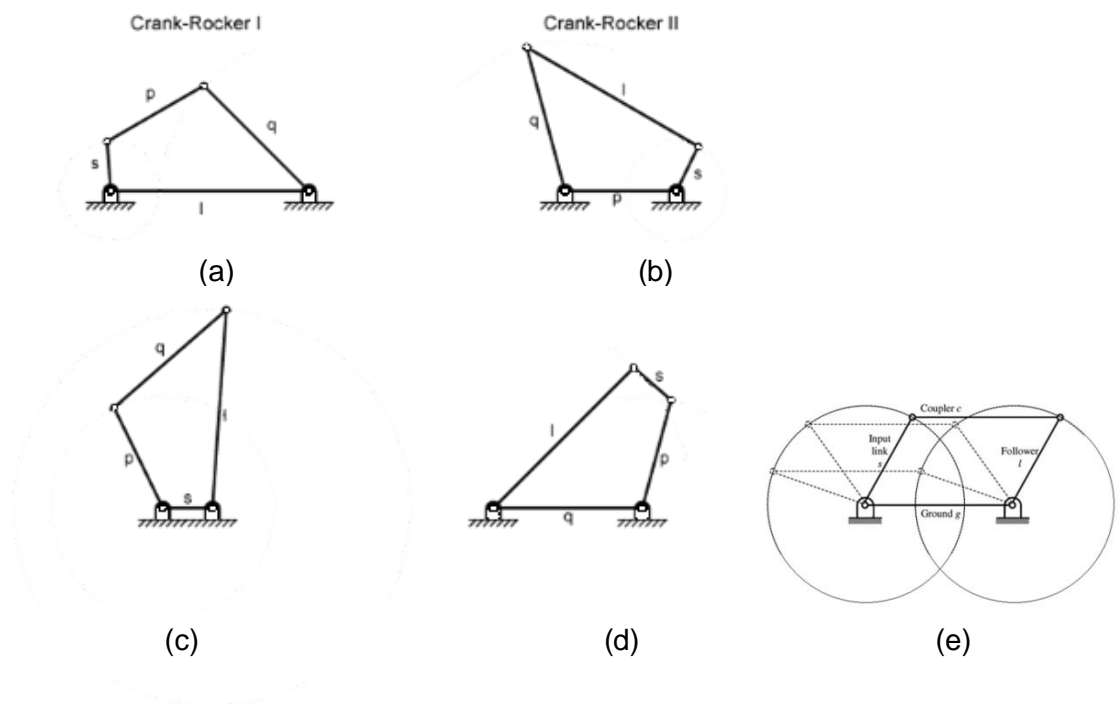


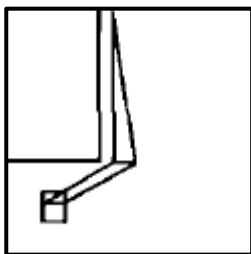
Figure 2.5 Types of four bar Linkage [5]

Parallellogram mechanism, in which two equal length links are not adjacent. All four inversions of this mechanism yield double crank mechanisms (Figure. 2.5(e)). In parallellogram mechanism, the opposite linkages are synchronous.

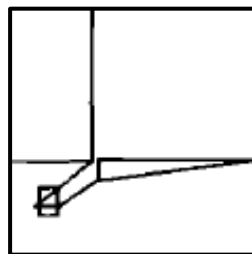
## 2.2 Decomposition of movement of tail lift

The tailboard lift's movement can be decomposed to three actions,

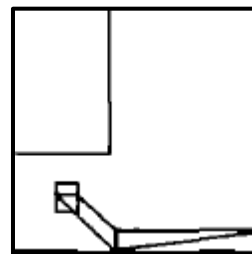
- i. Rollover motion: opening and closing the tailboard are realized by way of roll-over motion. In this process the tailboard rotate by the lower edge of the tailboard. The tailboard are opened while it moves from position as shown in Figure 2.6 (a) to (b), and it is closed while it moves from position as shown in Figure 2.6 (f) to (g);
- ii. Lifting motion: lifting movements of tailboard involve downloading and uploading, which are presented in Figure 2.6 from position (b) to (c) and from position (e) to (f), respectively.
- iii. Rollover motion: this rollover motion makes the tailboard touch ground and take off from the ground. Touching ground motion is shown in Figure 2.6, from position (c) to (d), and taking off motion is from (d) to (e).



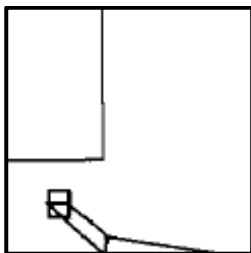
(a) Closed



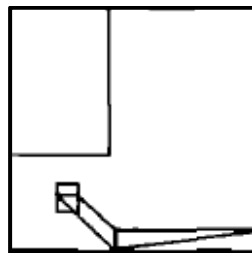
(b) Open horizontal



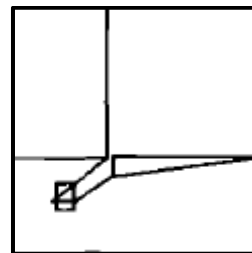
(c) Lift down



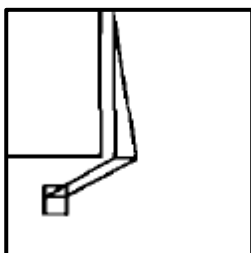
(d) Touch ground



(e) Take off



(f) Lift up



(g) Close back

*Figure 2.6 Movement proves*

### 2.3 Realization of rollover motion and lifting motion

#### Realization of rollover motion

Rollover motion can be realized by the mechanism shown in Figure 2.7 which can vividly present the process of turning movement of tailboard.

In Figure 2.7, triangle DCE represents the tailboard, link OA represents machine frame, line OD is rocker, AP means hydraulic cylinder and PC is piston.

At the start, the tailboard DCE is closed vertically as shown in Figure 2.7(a). The first step target movement is letting the tailboard rotate 90 degree until the board position change from vertical to horizontal which is shown in Figure 2.7(b). It will turn down by rotating with the axis D while piston PC moves from up to down. Analogous principle but reversed process when the tailboard turns back from horizontal position, i.e. it will turn up while piston PC moves from down to up. So the tailboard can turn by way of reciprocating of piston PC. In order to realize this movement, hydraulic cylinder can be selected.

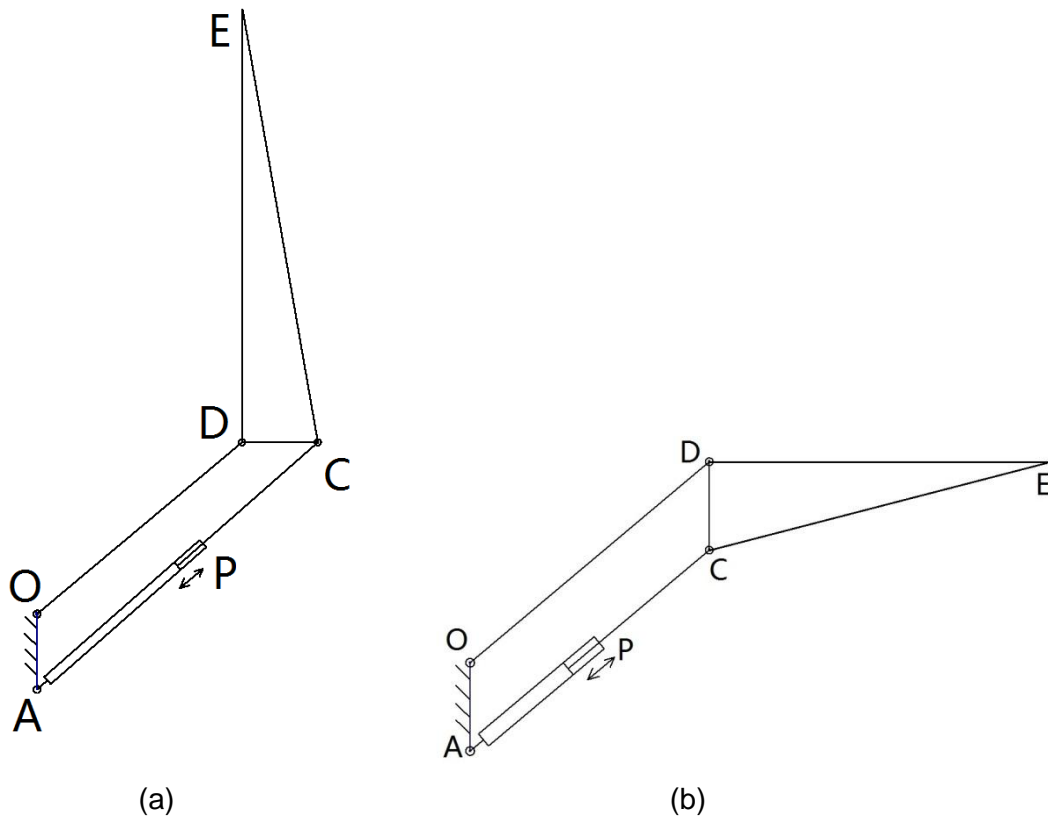


Figure 2.7 Schematic diagram of turning process

## Realization of lifting motion

Even if there are many different types of mechanisms can realize the translation motion. It is parallelogram mechanism that is the best and simplest one.

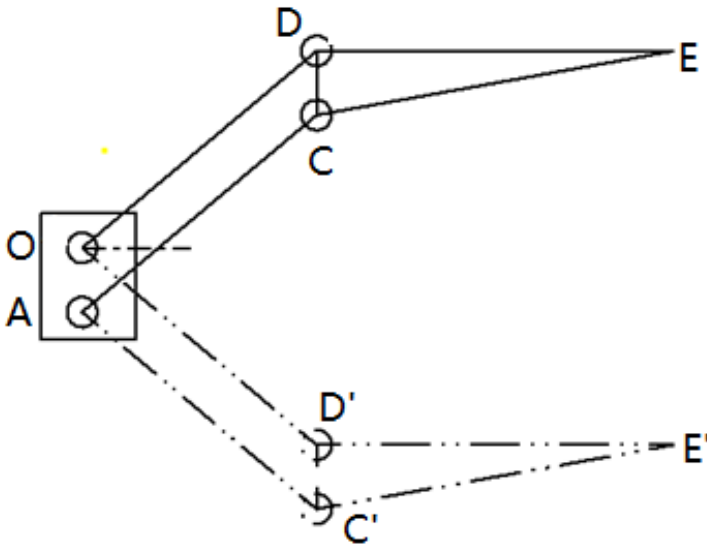


Figure 2.8 Schematic diagram of lifting process

As shown in Figure 2.8, there are four bars named OA, AC, CD and DO. In which,  $OA = CD$ ,  $AC = OD$  and  $OA < AC$ . Linkage OA is machine frame, impose a rotation movement on linkage OD, then making it revolve by O point as the center. As a result, AC should be always parallel with OD and DC can keep the vertical situation, i.e. DC can move by the way of translation motion. Expanding CD became DCE. Therefore, linkage DE can keep horizontal for lifting. [1, P.35]

Lift uploading and downloading is the primal movement and purpose of tail lift system; the basic principle of lifting process is a parallelogram mechanism. OD is the upper rod and AC is the lifting cylinder. O, A are hinge joints which are fixed on the frame. C, D are connected on the tailboard. Closed loop OACD is a parallelogram mechanism which is moving down from position E to E' during lifting process.

The dynamic force of the movement comes from the outer couple cylinders which control parallelogram mechanism OACD.

In practical application, there are two sets of four-bar parallelogram mechanism are placed on both sides of vehicle's stringer, and have synchronized action. DCE is the loading platform motioned above. [1, P.35]

During design process, there following three problems should be addressed:



- a) The driving force of rotating cylinder
- b) The point of action on rotating cylinder and function form
- c) When point C contacts to the earth, there should be a revolving action around point D, in order to make E also contact to earth for loading. [1, P.35]

#### Realization of subsidence motion

As shown in Figure 2.9, when Point C' touches the ground, it will trigger the subsidence process which has the same principle of the turning process. The rotating cylinders couple's elongate makes the tailboard sinking down and applying the board bottom to the ground.

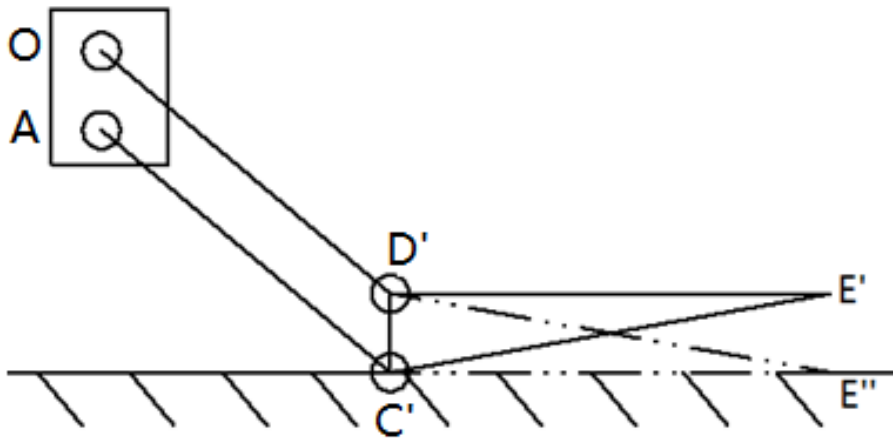


Figure 2.9 Principle of subsidence process

### 3 DESIGN PROCESS

#### 3.1 Design requirements

##### 3.1.1 Function requirements

The tail lift gate can realize the following actions, turning over to open and closed, lifting up and down for loading and unloading, and tilting when the board touches ground.

##### 3.1.2 Safety requirements

Ensure that all equipment are:

- a) Sufficiently strong, stable and suitable for the proposed use. Similarly, the load and anything attached (e.g. timber pallets, lifting points) must be suitable;
- b) Positioned or installed to prevent the risk of injury, e.g. from the equipment or the load falling or striking people;
- c) Visibly marked with any appropriate information to be taken into account for its safe use, e.g. safe working loads. Accessories, e.g. slings, clamps etc., should be similarly marked. [6]

In addition, the labors must ensure that:

- a) Lifting operations are planned, supervised and carried out in a safe manner by people who are competent;
- b) Where equipment is used for lifting people it is marked accordingly, and it should be safe for such a purpose, e.g. all necessary precautions have been taken to eliminate or reduce any risk;
- c) Where appropriate, before lifting equipment (including accessories) is used for the first time, it is thoroughly examined. Lifting equipment may need to be thoroughly examined in use at periods specified in the Regulations (i.e. at least six-monthly for accessories and equipment used for lifting people and, at a minimum, annually for all other equipment) or at intervals laid down in an examination scheme drawn up by a competent person. All examination work should be performed by a competent person;

- d) Following a thorough examination or inspection of any lifting equipment, a report is submitted by the competent person to the employer to take the appropriate action. [6]

### 3.2 Determination of technical parameters

The primary technical parameters concerned are: rated lifting mass, lifting route, lifting speed, measurements of rods, size of platform, diameters and working routes of hydro-cylinders, etc.

Generally, at the beginning of design process, there are several parameters which are carriage width, distance from carriage bottom to ground, distance between auto girders, height from girders to ground, measurements of rear overhang, etc. The following design has to be done basically according to those known parameters.



*Figure 3.1 Front and rear overhangs [7]*

Overhangs are the lengths of a car, at the front and rear, which extend beyond the wheelbase. They are normally described as front overhang and rear overhang. In Figure 3.1 above, distance A is the length of front overhang and distance B is the length of rear overhang. Practicality, style, and performance are affected by the size and weight of overhangs. [7]

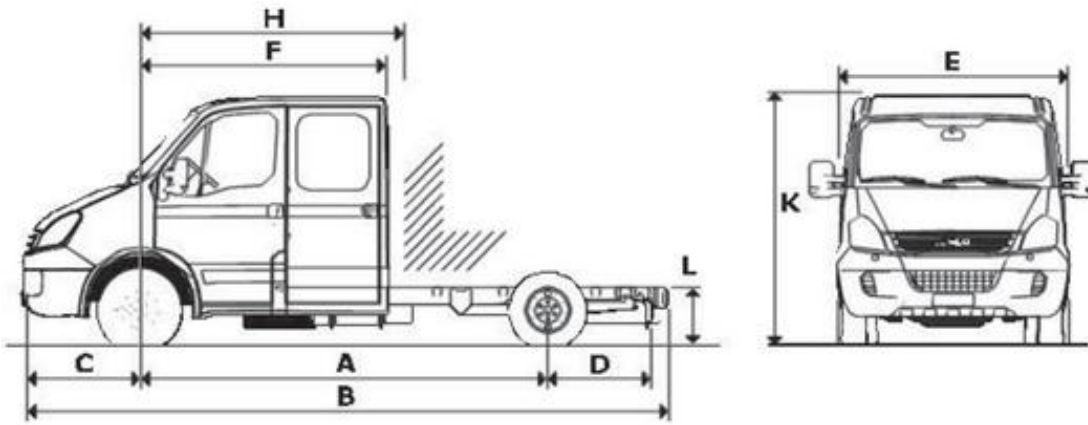


Figure 3.2 Instruction of Truck measurements [8]

Figure 3.2 above shows the partial definitions of a common truck which are (A) Wheelbase, (B) Overall length, (C) Front overhang, (D) Rear overhang, (E) Maximum cab width, (F) Front axle to back of cab, (H) Front axle to body, (K) Overall height (unloaded), and (L) Frame height at rear. [8]

### 3.2.1 Parameters of Design sample

In connection with this issue and around condition, truck NXG1160D3ZAL2X manufactured by XCMD (XCMS Construction Machinery Co., Ltd.) was used in this thesis. The basic parameters of Truck NXG1160D3ZAL2X are showed in Table 3.1.

Table 3.1 Parameters of Truck NXG1160D3ZAL2X

TYPE	CARGO SIZE		REAR HANGOVER HEIGHT	CARGO BOTTOM HEIGHT	GIRDER SPACE
	HEIGHT	WIDTH			
NXG1160D3ZAL2X (XCMD)	2600	2500	860	1300	870

The simple diagrammatic drawing (Figure 3.3) is showing the parameters of Table 3.1, the height of truck cargo is 2600 mm, and the cargo width is 2500 mm. And the height form bottom of cargo to ground is 1300mm, the height of rear overhang is 860 mm and the girder space is 870 mm which are show in below.

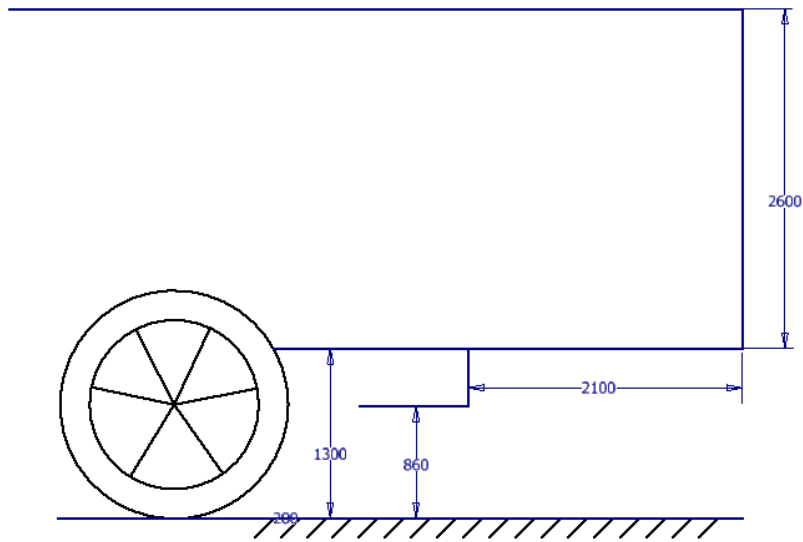


Figure 3.3 Diagrammatic drawing of truck from profile projection

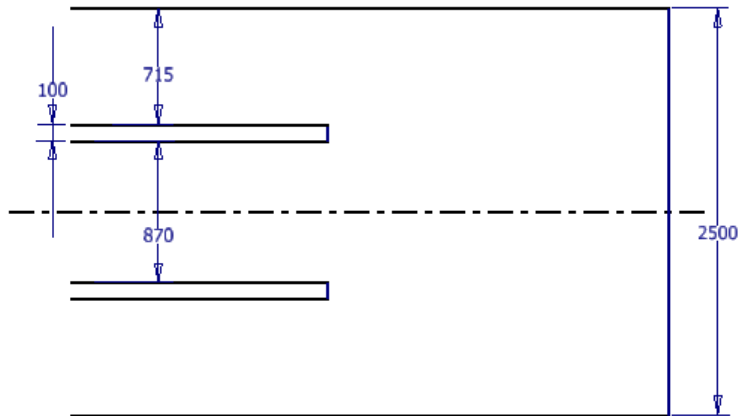


Figure 3.4 Diagrammatic drawing of truck from bottom projection

### 3.2.2 Determination of the dimensions

The theoretical lengths of the upper and lower rods are simplified two rockers of Parallelogram mechanism. The dimension is determined according to the arrangement of the selected chassis of the body and the modified compartment. As the initial parameters, the lengths should not be too large, because of the excessive length, the load force on board lead much more torque on the frame, and the needed thrust of cylinders is greaten as well. Also it should not be too small, otherwise, the angle of rotation is increased and the travel of cylinders is lengthened as well.

Generally, the extreme position of upper rod and lower cylinder determine the theoretical lengths of them. In Figure 3.5, upper and lower rods are abstracted as a paral-

lelogram mechanism and the both extreme positions are shown on it and  $H_0$  is the height of the tail board when it is opened horizontally.  $L_0$  is the theoretical length of the upper and lower rods. In here, at the up extreme position, the inclination angle is  $\beta$  which is normally approximately  $40^\circ$  to  $50^\circ$ .

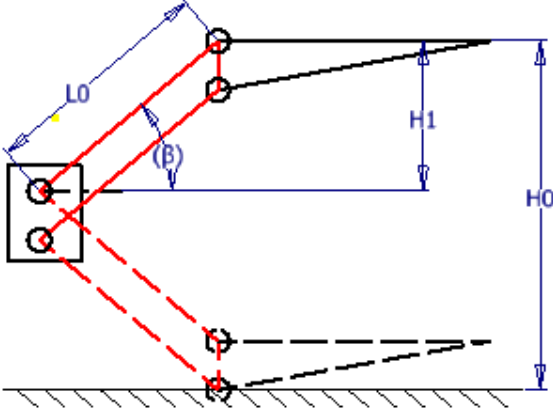


Figure 3.5 Determination of theoretical lengths

When the truck is selected, the height is determined which is  $H_0 = 1300$  mm, from upon Figure 3.3, theoretical length  $L_0$  is,

$$L_0 = \frac{H_1}{\sin \beta} \quad (1)$$

$H_1$  is the vertical distance from cabin floor plant to the bracket hinge point of the upper rod;  $\beta$  is the angle between upper extreme position and horizontal level.

In preliminary analysis, the both angles between two extreme positions and horizontal level are equally  $\beta$ . Then

$$H_1 = \frac{1}{2}(H_0 - C) \quad (2)$$

$C$  is the length of coupler link of the parallelogram mechanism, and normally  $C$  is approximately  $0.14L_0$ , merging formula (1) and (2), then

$$L_0 = \frac{1}{2}(H_0 - C) \frac{1}{\sin \beta} \quad (3)$$

Then

$$L_0 = \frac{0.5H_0}{\sin \beta + 0.075} \quad (4)$$

When  $\beta$  is  $40^\circ \sim 50^\circ$ ,  $\sin \beta = 0.64 \sim 0.77$ ,  $H_0 = 1300$  mm, then

$$L_0 = (0.70 \sim 0.59) H_0 = 910 \sim 767 [\text{mm}]$$

In order to facilitate the calculation, the result was made as

$$L_0 = 850 [\text{mm}], \text{ then } C = 0.14L_0 = 120 [\text{mm}]$$

### 3.2.3 Kinematics analysis of tail board

#### i. Vector method

Here a link is represented by a vector, using complex notation, and a displacement equation is obtained by considering the closed loop formed by the vectors representing the links. The number of such loops varies from one mechanism to another. A convention was adopted that the angle of orientation of any vector will be measured positively with respect to positive x-axis in counter clockwise direction. [9, P.127,128]

Using the well-known Euler's equation from trigonometry,

$$e^{\pm j\theta} = \cos \theta \pm j \sin \theta \quad (5)$$

Vector  $R$  can be written in complex polar form as

$$\mathbf{R} = r e^{j\theta} \quad (6)$$

In order to get familiarized with useful manipulation techniques, let us consider vector equation representing vector triangle of FIGURE 2.9.

$$\mathbf{C} = \mathbf{A} + \mathbf{B} \quad (7)$$

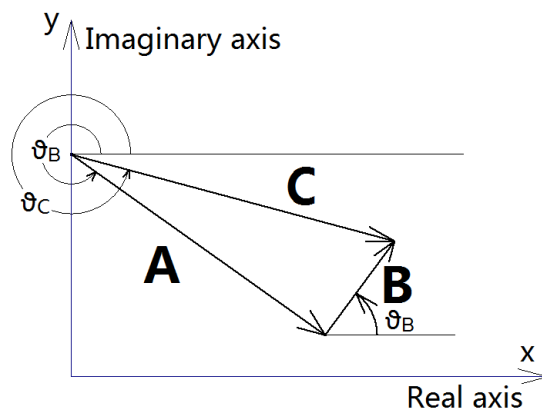


Figure 3.6 Convention of representation of vector equation

The equation can be rewritten in complex polar form as,

$$C e^{j\theta_C} = A e^{j\theta_A} + B e^{j\theta_B} \quad (8)$$

To evaluate the two unknown  $C$  and  $\theta_C$ , real and imaginary parts were separated by using Euler's equation.

Thus,

$$C(\cos \theta_C + j \sin \theta_C) = A(\cos \theta_A + j \sin \theta_A) + B(\cos \theta_B + j \sin \theta_B) \quad (9)$$

Equating real and imaginary parts separately,

$$C \cos \theta_C = A \cos \theta_A + B \cos \theta_B \quad (10)$$

and

$$C \sin \theta_C = A \sin \theta_A + B \sin \theta_B \quad (11)$$

squaring and adding on the two sides of equations (10) and (11),  $\theta_C$  is eliminated and a solution for  $C$  is obtained as,

$$C = \sqrt{(A \cos \theta_A + B \cos \theta_B)^2 + (A \sin \theta_A + B \sin \theta_B)^2} \quad (12)$$

Therefore

$$C = \sqrt{A^2 + B^2 + 2AB \cos(\theta_B - \theta_A)} \quad (13)$$

The choice for positive value of square root is arbitrary; a negative value of square root would yield a negative value for  $C$  which has values of  $\theta_C$  differing by  $180^\circ$  from the previous value. [9, P.127,128]

The angle  $\theta_C$  is obtained by dividing equation (11) by (10) on respective sides. Thus,

$$\tan \theta_C = \frac{A \sin \theta_A + B \sin \theta_B}{A \cos \theta_A + B \cos \theta_B} \quad (14)$$

or

$$\theta_C = \tan^{-1} \left( \frac{A \sin \theta_A + B \sin \theta_B}{A \cos \theta_A + B \cos \theta_B} \right) \quad (15)$$

In equation (13) signs of numerator and denominator must be considered separately in ascertaining proper quadrant of  $\theta_C$ . [9, P.127,128]

## ii. Case of four-bar mechanism

In this case, the writer applied the vector method to calculate the length of the oscillating hydraulic cylinder  $AC$ . [10]

### a) Kinematics analysis of turning process

During the turning process, points  $O$  and  $A$  are fixed on the frame, but the constraint is released on the joint  $D$ . The whole mechanism can be treated as a planar four-bar mechanism  $OACD$  with fixed linkage  $OA$  and  $OD$  to realize the movement. [10]

Point  $O$  is the hinge point which is between the upper rod  $OD$  and rear frame. Point  $A$  is the hinge point which is between oscillating hydraulic cylinder  $AC$  and rear frame.



Point C is the hinge point between the tailboard and oscillating hydraulic cylinder AC. [10]

Firstly set  $\beta$ ,  $\omega_1$ , and  $\varepsilon_1$  respectively are angular displacement, angular velocity, angular acceleration of the upper rod OD. At the beginning,  $\beta = \beta_0$ . The same way,  $\alpha$ ,  $\omega_2$ , and  $\varepsilon_2$  respectively are angular displacement, angular velocity, angular acceleration of the tail board and here  $\alpha = 0^\circ \sim 90^\circ$ . In addition,  $v_3$  is the relative velocity of cylinder rod to the oscillating hydraulic cylinder AC. The length of four bars are  $l_1 = l_{OD}$ ,  $l_2 = l_{DC}$ ,  $l_3 = l_{AC}$ ,  $l_4 = l_{OA}$ . [10]

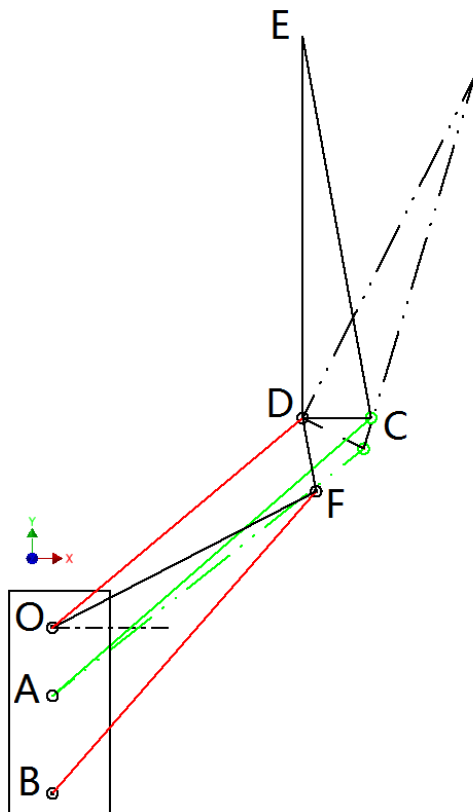


Figure 3.7 Schematic diagram of vector equation for turning process

The loop-closure equation was written in complex polar form for the four-bar mechanism.

$$\mathbf{OD} + \mathbf{DC} = \mathbf{OA} + \mathbf{AC} \quad (16)$$

Then

$$l_1 \cos \beta_0 + l_2 \cos \alpha = l_3 \cos \gamma \quad (17)$$

and

$$l_1 \sin \beta_0 - l_2 \sin \alpha = l_3 \sin \gamma + l_4 \quad (18)$$

Solving both equations (17) and (18) and then simplifying them into:

$$l_3 = \sqrt{(l_1 \cos \beta_0 + l_2 \cos \alpha)^2 + (l_1 \sin \beta_0 - l_2 \sin \alpha + l_4)^2} \quad (19)$$

$$\gamma = \tan^{-1} \frac{l_1 \sin \beta_0 - l_2 \sin \alpha + l_4}{l_1 \cos \beta_0 + l_2 \cos \alpha} \quad (20)$$

Then, coordinated equations of joint C are,

$$x_C = l_1 \cos \beta_0 + l_2 \cos \alpha \quad (21)$$

$$y_C = l_1 \sin \beta_0 - l_2 \sin \alpha \quad (22)$$

Equation (19) to time was differentiated ,

$$v_3 = -\frac{l_2 \omega_2}{l_3} [l_1 \sin(\alpha + \beta_0) + l_4 \cos \alpha] \quad (23)$$

Therefore,

$$\omega_2 = \frac{-l_3 v_3}{l_2 [l_1 \sin(\alpha + \beta_0) + l_4 \cos \alpha]} \quad (24)$$

In equation (24),  $v_3 = \frac{2Q}{\pi D_3^2}$

Q [mm<sup>3</sup>/s] is oil flow rate of hydraulic system

$D_3$  [mm] is cylinder diameter of the oscillating hydraulic cylinder

Last equation (24) is differentiated to time is:

$$\varepsilon_2 = \frac{-v_3^2}{l_2 [l_1 \sin(\alpha + \beta_0) + l_4 \cos \alpha]} + \frac{\omega_2 l_3 v_3 [l_1 \cos(\alpha + \beta_0) - l_4 \sin \alpha]}{l_2 [l_1 \sin(\alpha + \beta_0) + l_4 \cos \alpha]^2} \quad (25)$$

[10]

b) Kinematics analysis of lifting process

During the lifting process, points O and A are fixed on the frame, but the constraint is released on the joint D. The whole mechanism can be treated as a parallelogram mechanism OACD with fixed linkage OA to realize the lifting movement. [10]

As following, mechanism ODF is the U-shape frame. Point F is the hinge joint with lifting hydraulic cylinder BF, and point B is the hinge joint between frame and lifting hydraulic cylinder. [10]

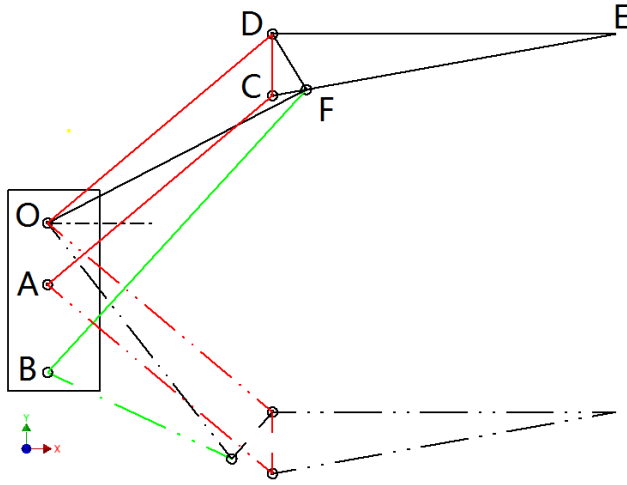


Figure 3.8 Schematic diagram of vector equation for lifting process

Let us set angle  $DPF = \theta$ ,  $l_5 = l_{OF}$ ,  $l_6 = l_{BF}$ ,  $l_7 = l_{OB}$ , and  $l_8$  is the project of  $OB$  on the x-axis,  $l_9$  is the project of  $OB$  on the y-axis.  $v_6$  is the relative velocity of cylinder rod to the lifting hydraulic cylinder  $BF$ .  $\beta$ ,  $\omega_1$  and  $\varepsilon_1$  respectively are angular displacement, angular velocity, angular acceleration of the U shaped frame. [10]

The loop-closure equation could be written in complex polar form for the triangle  $OFB$ ,

$$\mathbf{OF} + \mathbf{BO} = \mathbf{BF} \quad (26)$$

Respectively project equation (26) to x and y axis can be obtained,

$$l_5 \cos(\beta - \theta) - l_8 = l_6 \cos \delta \quad (27)$$

$$l_5 \sin(\beta - \theta) - l_9 = l_6 \sin \delta \quad (28)$$

Simultaneous equations (27) and (28) can be done,

$$l_6 = \sqrt{[l_5 \cos(\beta - \theta) - l_8]^2 + [l_5 \sin(\beta - \theta) + l_9]^2} \quad (29)$$

$$\delta = \tan^{-1} \frac{l_5 \sin(\beta - \theta) + l_9}{l_5 \cos(\beta - \theta) - l_8} \quad (30)$$

Then, coordinated equations of joint C are,

$$x_C = l_1 \cos \beta \quad (31)$$

$$y_C = l_1 \sin \beta - l_2 \quad (32)$$

Last equation (29) to time is differentiated:

$$v_6 = -\frac{l_5 \omega_1}{l_6} [l_8 \sin(\beta - \theta) + l_9 \cos(\beta - \theta)] \quad (33)$$

Therefore,

$$\omega_1 = \frac{l_6 v_6}{l_5 [l_8 \sin(\beta - \theta) + l_9 (\beta - \theta)]} \quad (34)$$

in equation (34),  $v_6 = \frac{2Q}{\pi D_6^2}$

$Q$  [mm<sup>3</sup>/s] is oil flow rate of hydraulic system

$D_3$  [mm] is cylinder diameter of the lifting hydraulic cylinder

Equation (34) to time is differentiated,

$$\varepsilon_1 = \frac{v_6^2}{l_5 [l_8 \sin(\beta - \theta) + l_9 \cos(\beta - \theta)]} + \frac{\omega_1 l_6 v_6 [-l_8 \cos(\beta - \theta) + l_9 \sin(\beta - \theta)]}{l_5 [l_8 \sin(\beta - \theta) + l_9 \cos(\beta - \theta)]^2} \quad (35)$$

[10]

### c) Kinematics analysis of subsidence process

Subsidence process means when point  $C$  touches the ground, then tailboard rotates around point  $D$  by oscillating hydraulic cylinder  $AC$  until the bottom of tailboard  $CE$  hugs with the ground. [10]

Simplified the subsidence process described simplified below, and  $\alpha$ ,  $\omega_2$ , and  $\varepsilon_2$  were set respectively as angular displacement, angular velocity, angular acceleration of the tail board and here  $\alpha = 0^\circ \sim 6^\circ$ . In addition,  $v_3$  is the relative velocity of cylinder rod to the oscillating hydraulic cylinder  $AC$ . The length of four bars are  $l_1 = l_{OD}$ ,  $l_2 = l_{DC}$ ,  $l_3 = l_{AC}$ ,  $l_4 = l_{OA}$ . [10]

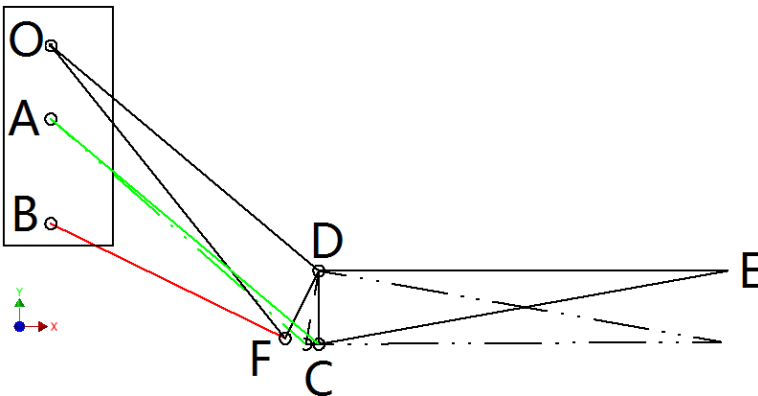


Figure 3.9 Schematic diagram of vector equation for subsiding process

The loop-closure equation could be written in complex polar form for the four-bar mechanism.

$$\mathbf{OD} + \mathbf{DC} = \mathbf{AC} + \mathbf{OA} \quad (36)$$

Respectively project equation (21) to x and y axis, leads to,

$$l_1 \cos \beta_1 - l_2 \sin \alpha = l_3 \cos \delta \quad (37)$$

$$l_1 \sin \beta_1 + l_2 \cos \alpha = l_3 \sin \delta + l_4 \quad (38)$$

simultaneous equations (37) and (38) are done,

$$l_3 = \sqrt{(l_1 \cos \beta_1 - l_2 \sin \alpha)^2 + (l_1 \sin \beta_1 + l_2 \cos \alpha - l_4)^2} \quad (39)$$

$$\gamma = \tan^{-1} \frac{l_1 \sin \beta_1 + l_2 \cos \alpha - l_4}{l_1 \cos \beta_1 - l_2 \sin \alpha} \quad (40)$$

Then, coordinated equations of joint C are,

$$x_C = l_1 \cos \beta_1 - l_2 \sin \alpha \quad (41)$$

$$y_C = l_1 \sin \beta_1 - l_2 \cos \alpha \quad (42)$$

Last equation (39) to time is differentiated,

$$v_3 = \frac{l_2 \omega_2}{l_3} [l_4 \sin \alpha - l_1 \cos(\alpha - \beta_1)] \quad (43)$$

Therefore,

$$\omega_2 = \frac{l_3 v_3}{l_2 [l_4 \sin \alpha - l_1 \cos(\alpha - \beta_1)]} \quad (44)$$

In equation (44),  $v_3 = \frac{2Q}{\pi D_3^2}$

Q [mm<sup>3</sup>/s] is oil flow rate of hydraulic system

$D_3$  [mm] is cylinder diameter of the oscillating hydraulic cylinder

Equation (44) to time is differentiated,

$$\varepsilon_2 = \frac{v_3^2}{l_2 [l_4 \sin \alpha - l_1 \cos(\beta_1 - \alpha)]} + \frac{\omega_2 l_3 v_3 [-l_4 \cos \alpha - l_1 \sin(\beta_1 - \alpha)]}{l_2 [l_4 \sin \alpha + l_1 \cos(\beta_1 - \alpha)]^2} \quad (45)$$

[10]

### 3.2.4 Calculating Results

According to the above calculating formulae and combination with the designing parameters which are shown in Table 3.2, the results can be obtained as follows:

Table 3.2 Parameters of mechanisms

[mm]							[Degree]			
$l_1$	$l_2$	$l_4$	$l_5$	$l_8$	$l_9$	$D_3$	$D_6$	$\theta_1$	$\beta_0$	$\beta_2$
850	120	120	775	10	260	63	63	14.71	22	-60

In the process of turnover from closing position to opening position, the parameters of hydraulic cylinder and the coordinates of Point C are shown in Table 3.3.

Table 3.3 Parameters of hydraulic cylinder and coordinates of point C in turnover process

Position	$l_3$	$\gamma_1$	$v_1$	$\omega_1$	$\alpha_1$	$X_{C_1}$	$Y_{C_1}$
start	1008.40	25.77	0.0012	-2.25E-05	-9.74E-10	908.11	318.42
middle	941.84	22.05	0.0012	-1.06E-05	-4.36E-11	872.96	233.56
end	850.00	22.00	0.0012	-1.06E-05	3.30E-11	788.11	198.42

In the process of lifting from the upper horizontal to the lower horizontal position, the parameters of hydraulic cylinder and the coordinates of Point C are shown in Table 3.4.

Table 3.4 Parameters of hydraulic cylinder and coordinates of point C in lifting process

Position	$l_6$	$\delta_2$	$v_2$	$\omega_2$	$\alpha_2$	$X_{C_2}$	$Y_{C_2}$
start	839.00	76.15	-0.0012	4.91E-06	3.87E-09	788.11	198.42
middle	657.06	21.12	0.0012	4.72E-06	6.47E-09	803.76	396.54
end	524.83	32.76	-0.0012	1.35E-05	-2.81E-09	425.36	198.42

In the process of landing, the parameters of hydraulic cylinder and the coordinates of Point C are shown in Table 3.5.

Table 3.5 Parameters of hydraulic cylinder and coordinates of point C in landing process

Position	$l_{32}$	$\alpha_2$	$v_{32}$	$\omega_3$	$\alpha_3$	$X_{C_3}$	$Y_{C_3}$
start	850.00	-60.00	0.0012	-1.96E-05	-2.06E-11	425.00	-856.12
middle	847.02	-60.37	0.0012	-2.18E-05	-2.11E-11	418.72	-855.96
end	844.37	-60.76	0.0012	-2.48E-05	-2.07E-11	412.46	-855.46

## 4 MODELING IN INVENTOR SOFTWARE

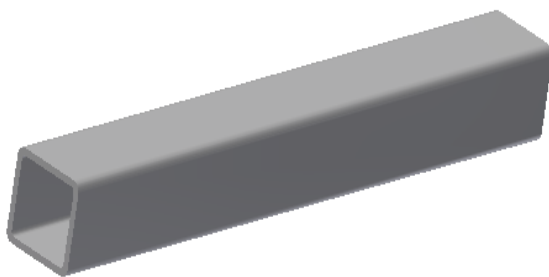
### 4.1 Creation of foundation drawing

After the parameters determination and kinematic principle analysis, the four positions of the board, and the frame of tail lift system can be fixed under by the rear frame. Then the designer should decide the size of frame, and also decide the shape of three different ear plates separately for upper rocker, lifting cylinder and rotating cylinder.

After that, the length of upper rocker and lifting cylinder and rotating cylinder were calculated by the above principles and positions. Then the foundation sketch was drawn for continuing modeling.

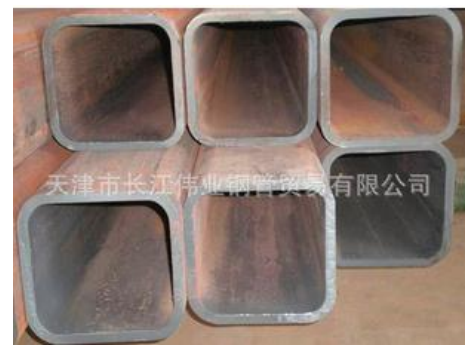
### 4.2 Modeling of frame

Frame, the base of tailboard system, is fixed on the rear overhang. Its shape was set as a hollow square column which is shown in Figure 4.1 by firstly creating double concentric squares Inner square's side length is 170 mm and the outer square's side length is 200 mm, then extruding the pattern for 1500 mm. Secondly, the eight edges are filleted with 10 mm radius. Detail drawing is shown in Appendix A-1. Steel AISI 1020 was chosen as the material of frame.



(a)

*Figure 4.1 (a) Structure of frame*



(b)

*(b) Product selling in market*

### 4.3 Modeling of ear plants

As shown in Figure 4.2,

- a) Upper ear plates is defined as connecting upper rocker through a cylindrical pins
- b) Lifting ear plates is defined as connecting lifting hydraulic cylinder through cylindrical pins
- c) Rotating ear plates is defined as connecting lifting hydraulic cylinders through cylindrical pins;

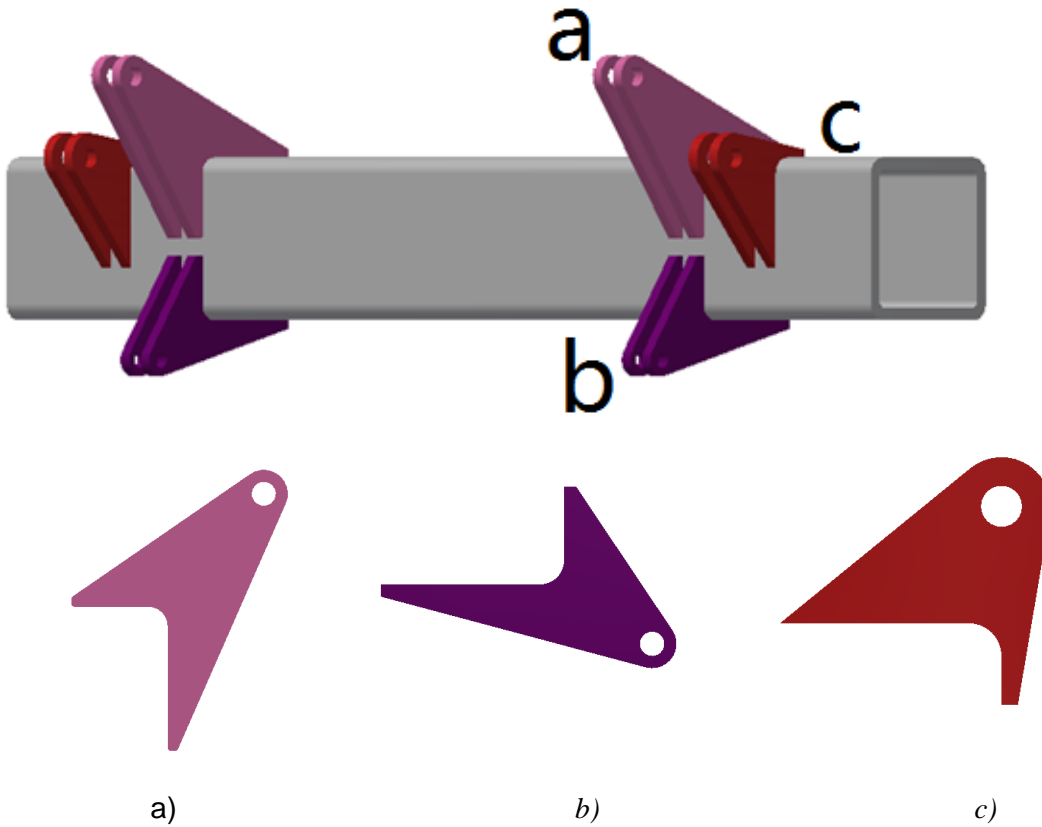


Figure 4.2 Ear Plate a) upper ear plate b) lifting ear plate c) rotating ear plate

Since the interval of rear frame in a van is 870mm, the interval of two upper ear plates is 870 mm too. According to the foundation drawing, the distances between the center of the hole in upper ear plate and the one in rotating ear plate are determined by the corresponding distance of two holes in board ear plate. They must be equivalent because they should compose a parallelogram.

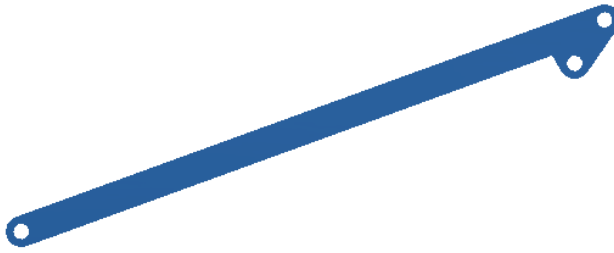
Material for each ear plate is Steel AISI 1020. More details about the ear plates are provided in detail drawing which are shown in Appendix A from A-2 to A-4 respectively.

#### 4.4 Modeling of upper rocker

Upper rocker is the stable rod connected with three endpoints; the positions of lifting ear plate and closed board ear plate were confirmed to decide the length of upper



rocker. Material for upper rocker is Steel AISI 1020. Detailed drawing is shown in Appendix A-5.



*Figure 4.3 Upper rocker*

#### 4.5 Modeling of Ear plate of board



*Figure 4.4 Ear plate of board 1*

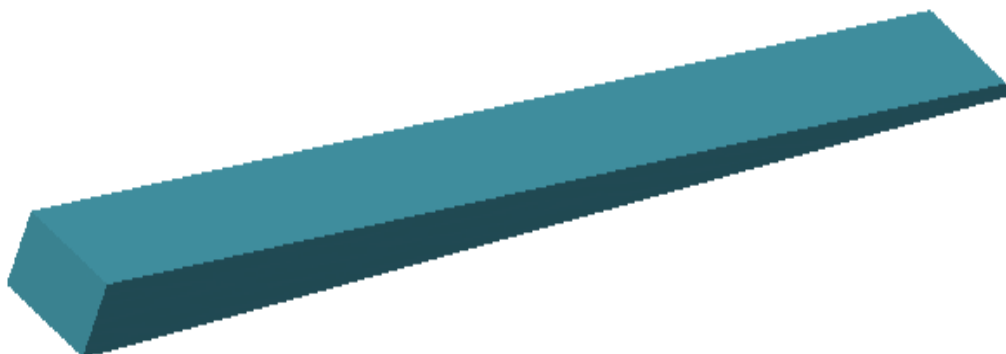


*Figure 4.5 Ear plate of board 2*

According parallelogram mechanism theory, if there are two edges confirmed, the others will be sure as well. Then the relative positions of hinges on ear plate of board are ensured. Following this, the ear plate of board could be designed. Detail drawing is shown in Appendix A from A-6 to A-7 respectively.

#### 4.6 Modeling of board

Figure 4.6 shows the model of board base, and Figure 4.7 shows the mass of board base while it is made of Aluminum. Its mass is 88.39 kg.



*Figure 4.6 Board base*

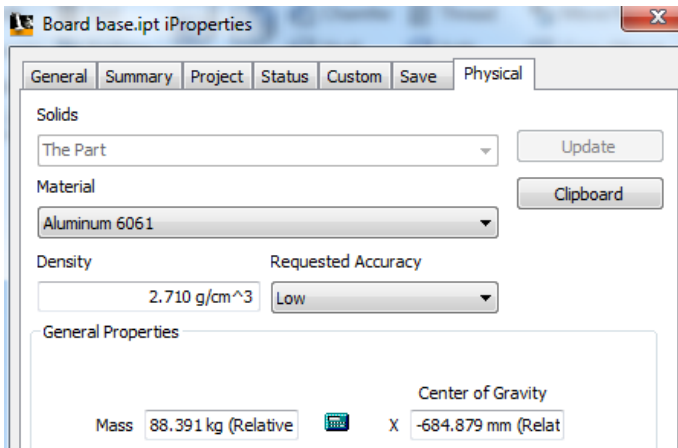


Figure 4.7 The mess of board base

Board base is used under the board and it holds up the loads with board. The shape of section was designed in the foundation drawing. The detailed drawing is shown in Appendix A from A-8.

#### 4.7 Modeling of board

The board was designed not only a work plate but also functioning as a tail gate. Therefore the parameters of board are collected from container, the size of board was set as 2500\*1900\*20[mm<sup>3</sup>]. It is shown in Figure 4.7.

Aluminum (Al) was chosen as the material of the board. Aluminum is light and it has high strength. Initially, the solid board was chosen. It has large gravity because of large dimension. As shown in Figure 4.8, its mass (257.45kg) is generated by Inventor.

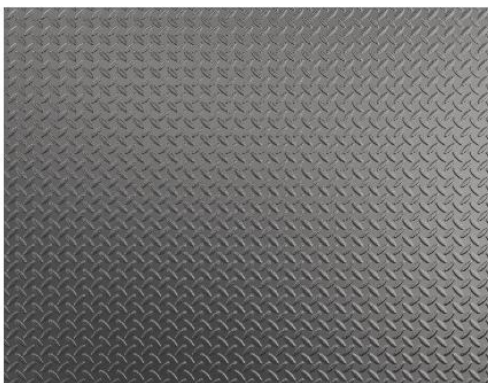


Figure 4.7 Board

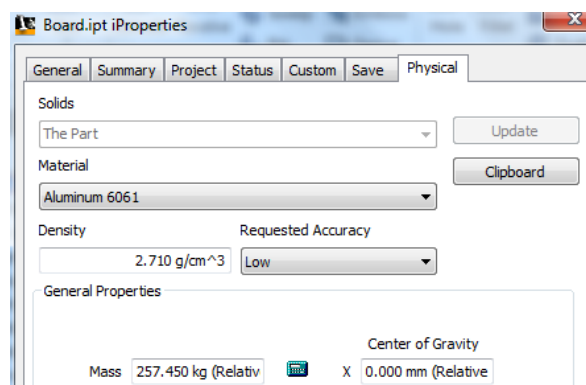


Figure 4.8 The mass of solid board

In order to further reduce its mass, the hollow structure was designed, as shown in Figure 4.9. Then, the mass was decreased to 162.5 kg as shown in Figure 4.10.

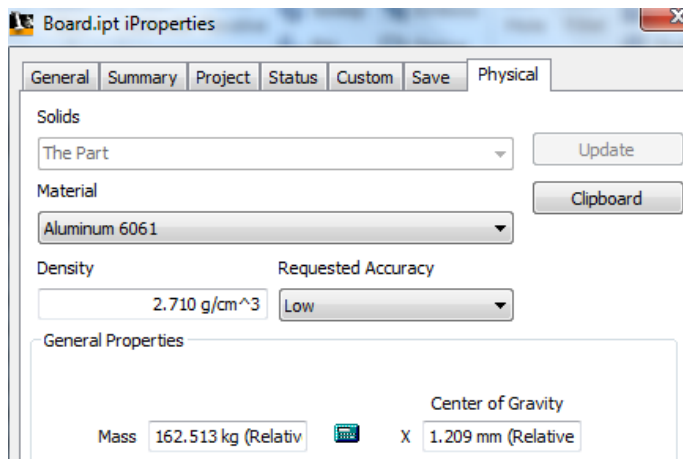


Figure 4.9 The mass of hollow Board

#### 4.8 Modeling of hydraulic cylinders

Beside these components, there still are hydraulic cylinders and pistons. Because they are standard parts, their shapes and dimensions need not to be designed. In this tail lift, Steel AISI 4130 (R683/II 2) was chosen as material of pistons and hydraulic cylinders. Because Steel AISI 4130 (R683/II 2) alloy is structural steel, it has a high static strength more than 1000 MPa and yield strength more than 850 MPa, it has impact toughness and high fatigue limit, high hardenability, high temperature with high creep strength and rupture strength, its long-term working temperature can reach 500 °C.

Table 4.1 is the summary of property of material used in this design.

Table 4.1 Property of materials in used

Name		Steel AISI 1020	Steel AISI 4130	Aluminum 6061
General	Mass Density	7.858 g/cm <sup>3</sup>	7.799 g/cm <sup>3</sup>	2.71 g/cm <sup>3</sup>
	Yield Strength	360 Mpa	276 MPa	275 MPa
Stress	Young's Modulus	205 GPa	200 GPa	68.9 GPa
	Poisson's Ratio	0.29 ul	0.3 ul	0.33 ul
	Shear Modulus	79.4574 GPa	76.9027 GPa	25.9023 GPa
Stress Thermal	Expansion Coefficient	0.000019 ul/c	0.000012 ul/c	0.0000236 ul/c
	Thermal Conductivity	49.8 W/(m K)	21.6173 W/(m K)	167 W/(m K)
	Specific Heat	470 J/(kg c)	499 J/(kg c)	1256.1 J/(kg.c)
Part Name(s)		Frame	Cylinder piston	Board
		Board ear plate	Cylinder body	Board base
		Ear plate upper	Pin ear rot	
		Ear plate rotating		
		Ear plate lifting		
		Upper rocker		
Total Mass	617.797 Kg			

## 5 STRESS ANALYSIS IN INVENTOR SOFTWARE

### 5.1 The Finite Element Method (FEM)

The finite element method (FEM) is a mathematical/ computer-based numerical technique for calculating the strength and behavior of engineering structures. Autodesk Inventor – and much other analysis software – is based on the FEM, where, simply, a component is broken down into many small elements. [13, P.235]

It's assumed a need to determine the displacement of the component. This displacement (unknown quantity) acts over each element in a predefined manner – with the number and type of elements chosen so that overall distribution through the component is sufficiently approximated. This distribution across each element is commonly presently by a polynomial – whether it is linear, quadratic, or even cubic. It is important to note that FEM is always an approximation of the actual component and by its very nature will have errors due to discretization – particularly around curved boundaries or geometrically complex components. [13, P.235]

These errors due to distribution can be reduced by either specifying more elements or using higher order polynomials to approximate the distribution of the unknown quantity over the elements – also referred to as polynomial function. Most finite element software uses the former method, specifically known as the H-refinement process, in with the software goes through an iterative process of reducing the number of elements at each iteration until the results have converged. The latter method, of using higher order polynomial at each iteration starts from 1(linear) to 2(quadratic), 3(cubic), and so on. [13, P. 235-236]

### 5.2 Static analysis

Static analysis is an engineering discipline that determines the stress in materials and structures subjected to static or dynamic forces or loads. The aim of the analysis is usually to determine whether the element or collection of elements, usually referred to as a structure or component, can safely withstand the specified forces and loads. This is achieved when the determined stress from the applied force(s) is less than the yield strength the material is known to be able to withstand. This stress relationship is

commonly referred to as factor of safety (FOS) and is used in many analysis as an indicator of success or failure in analysis. [13, P.240]

$$\text{FOS} = \frac{\text{Yield stress}}{\text{Calculated stress}}$$

Factor of safety can be based on either the yield or ultimate stress limit of the material. The FOS on yield strength aims to prevent detrimental deformations and the FOS on ultimate strength aims to prevent collapse linear analysis and hence FOS will more commonly be based on yield limit. [13, Page. 240]

### 5.3 Stress analysis workflow

The process of creating a dynamic simulation study involves four core steps [13, P.247]:

STEP 1: IDEALIZATION - Simplify Geometry, including setting up the analysis.

STEP 2: BOUNDARY CONDITIONS - Apply constraints including loads, including exporting loads from simulation.

STEP 3: RUN SIMULATION AND ANALYSIS - Analyze initial result, including convergence of results.

STEP 4: OPTIMIZATION - Modify geometry to meet design goals, including changing original material.

### 5.4 Stress analysis of board base

If the mass of goods placed on the board is 1000 kg, then the force on the board should equal about 10000 N (1000 kg\*9.8 N/kg). The load should be distributed on both board bases. Each board base carries 5000 N, and this force is divided by the upper surface of board base, then the stress on the board base surface could be calculated. The area could be calculated in Inventor as shown in Figure 5.1.

The stress on each board base is:

$$\sigma = \frac{F}{A} = \frac{5000}{532650} \text{MPa} = 0.0094 \text{MPa}$$

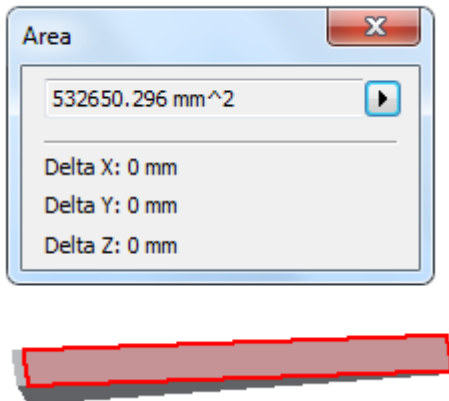


Figure 5.1 Area of board base surface

## 5.5 Stress analysis of the whole tail lift

### Von Mises Stress

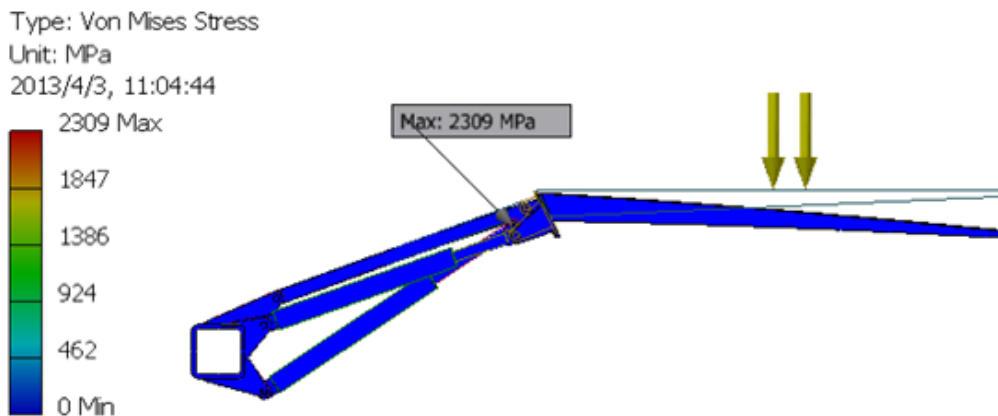


Figure 5.2 Result of Von Mises stress

By operating stress analysis in Inventor, the Von Mises Stress result shows, in Figure 5.2, which the maximum stress was at the pin joints position which is 2309 MPa, and it is incredibly high stress certainly making the pin bolts break off.

## 5.6 Conclusion of stress analysis

According to the results of FEA, the maximum stress will occur in the hinge which joins the rotating hydraulic cylinder and the board ear plate as shown in Figure 5.2.

The max stress is more than 2000 MPa; it is very huge for strength limit of steel. It means that there exists some unreasonable reasons in the tail lift structure or the materials of the components.

In writer's opinion, there are three scenarios which may solve this problem:

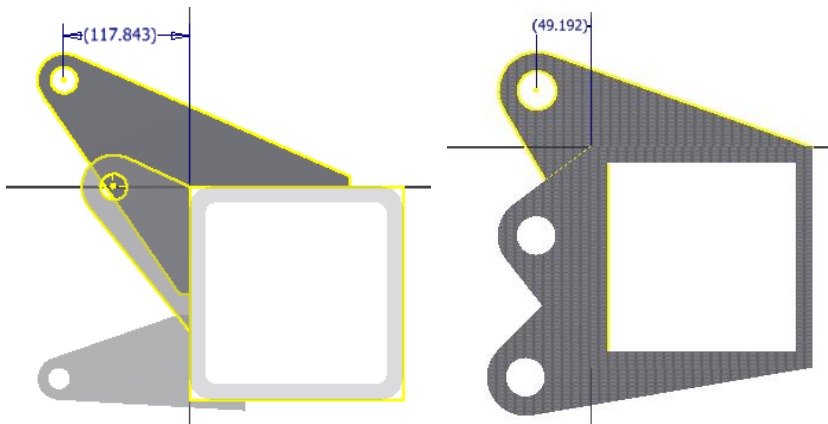
- 1) To redesign the structure of the tail lift to reduce the load in the hinge where the maximum stress will happen.
- 2) To increase the cross section of pin where maximum stress will happen or change the material of the pin, hence, the stress in the pin could be decreased.
- 3) To reduce the load limit on the board.

## 6 Modified Model of Tail Lift

### 6.1 Modification of tail lift

After considering some possible reasons which can affect the stress analysis results, all-round parameters of some components were changed in new model of tail lift. The details are illustrated as following:

- 1) The ear plates fixed on frame are shorter than before; hence the stiffness can be increased. Figure 6.1 shows one of the modified dimensions of ear plates. The others are provided in detailed drawings of components in Appendix B-1.



*Figure 6.1 The comparison of original and modified frame*

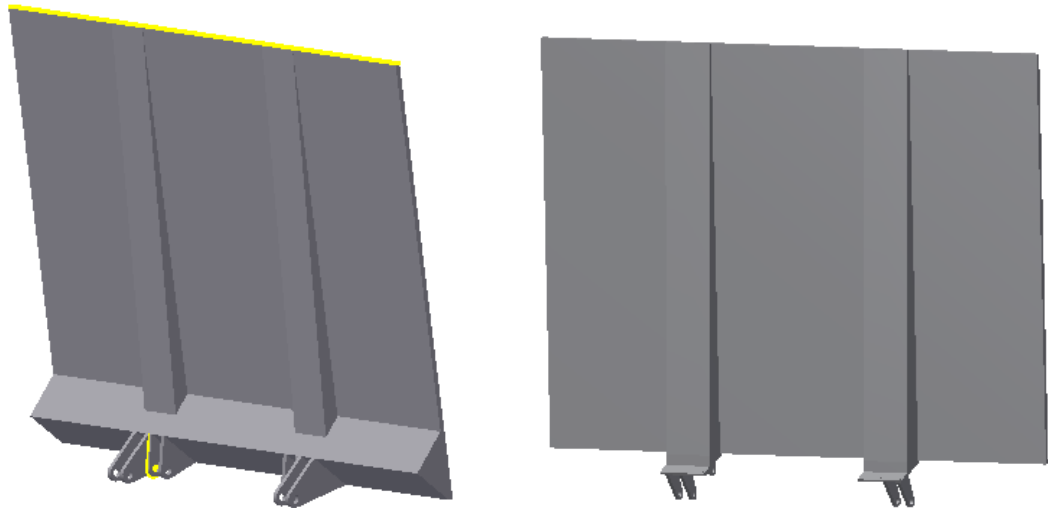
- 2) For all joint pins the diameter was changed from 25 mm to 35 mm in order to decrease the shear stress and crushing stress.
- 3) The structure of upper rocker which was modified is shown in figure 6.2. The distance between two holes which were respectively connected with board and lifting hydraulic cylinder is longer than before. Detailed drawings of modified upper rocker were shown in Appendix B-2.



*Figure 6.2 The comparison of original and modified upper rocker*



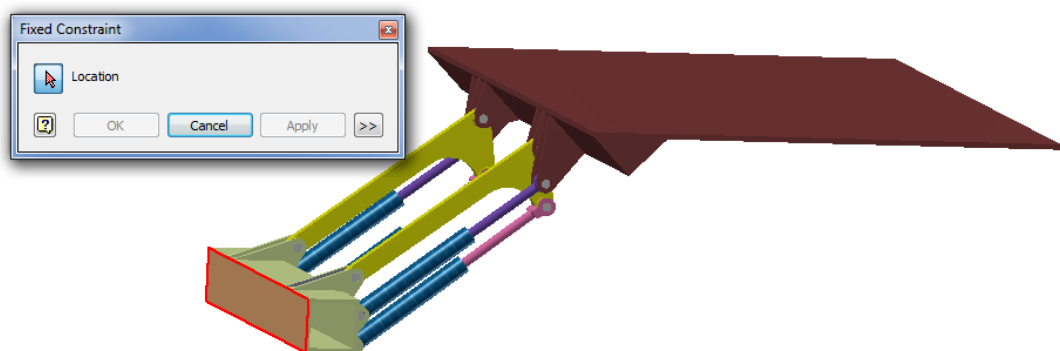
The structure of the board was modified. Figure 6.3 shows the difference between the original and modified one. In order to improve capacity of load-carrying, the section jointed with the ear plate was changed. The total weight of modified board was 421.4 kg. Detailed drawing of modified upper rocker were shown in Appendix B-3.



*Figure 6.3 The comparison of original and modified board*

## 6.2 Stress Analysis of modified tail lift

The mechanisms in the two positions are analyzed according to the FEA theory and the steps. From Figure 6.4 to Figure 6.7, they show the constraint, applying loads and the results of stress and displacement respectively at upper horizontal position.



*Figure 6.4 Constraint face at upper loading position*

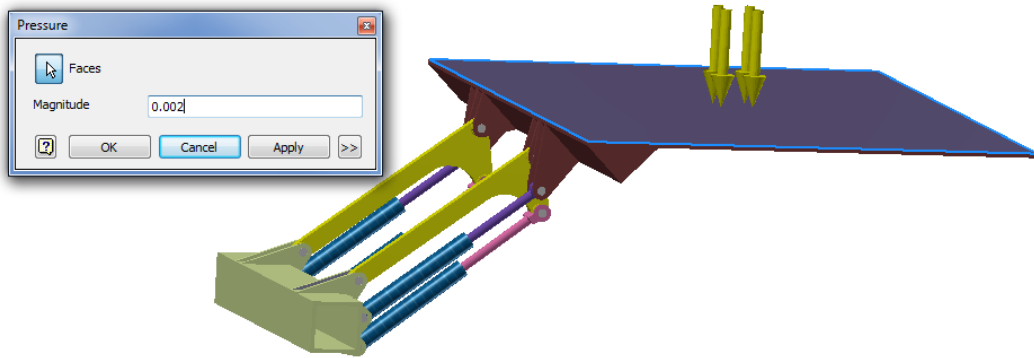


Figure 6.5 Applying loads

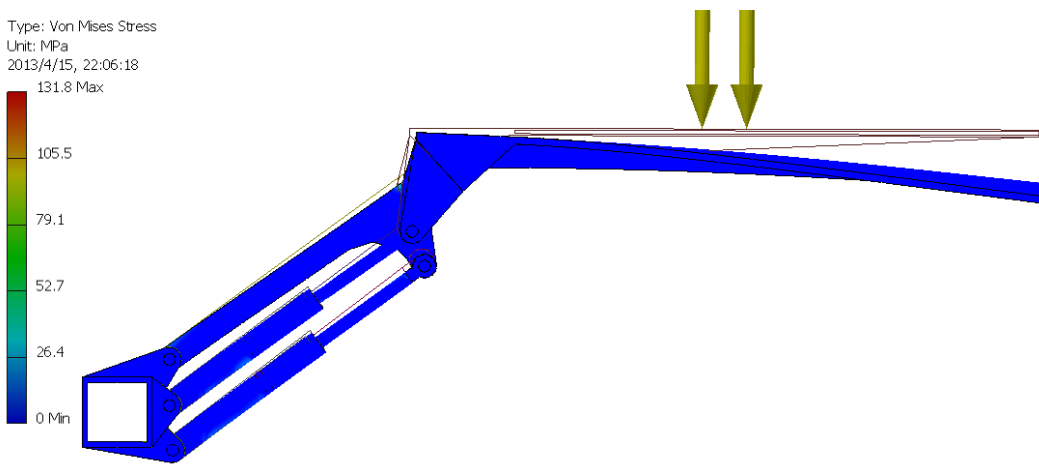


Figure 6.6 Result of Von Mises stress

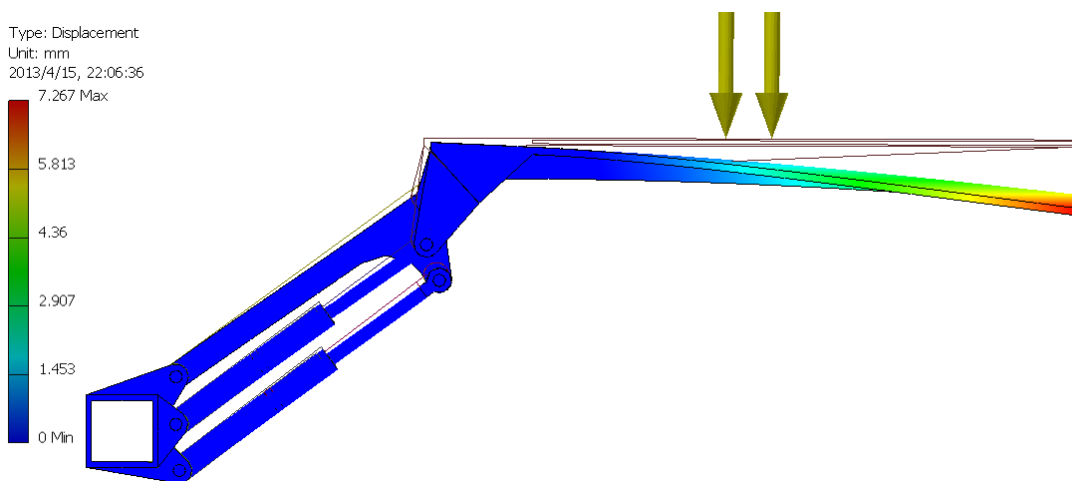


Figure 6.7 Result of displacement at upper loading position

From Figure 6.8 to Figure 6.12, they show the meshed model and mesh settings, the constraint, applying loads and the results of stress and displacement respectively at lower horizontal position.

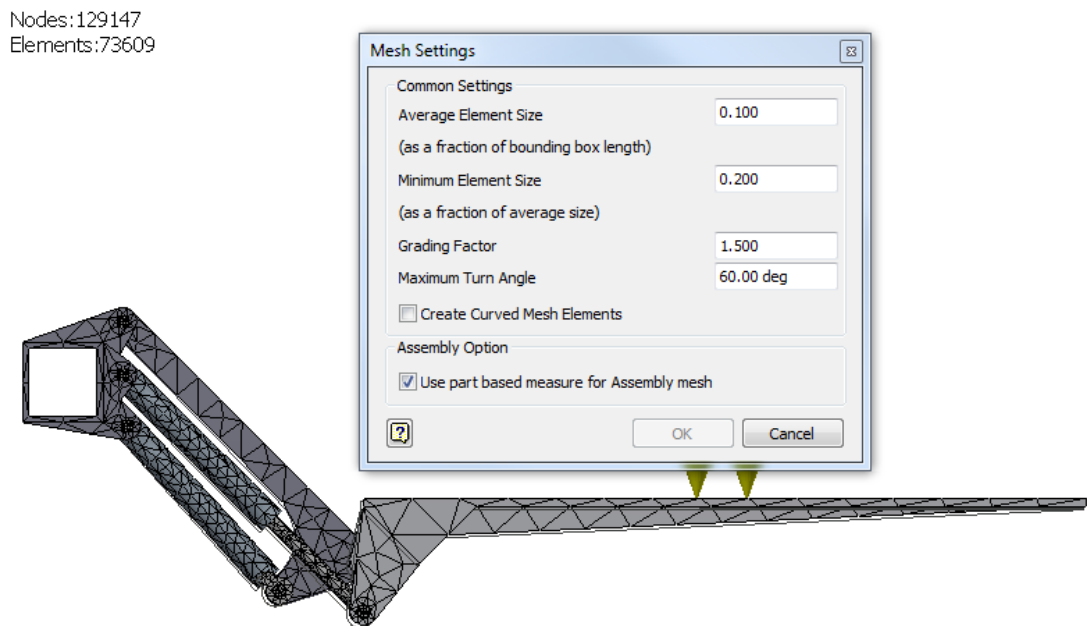


Figure 6.8 the meshed model and mesh settings

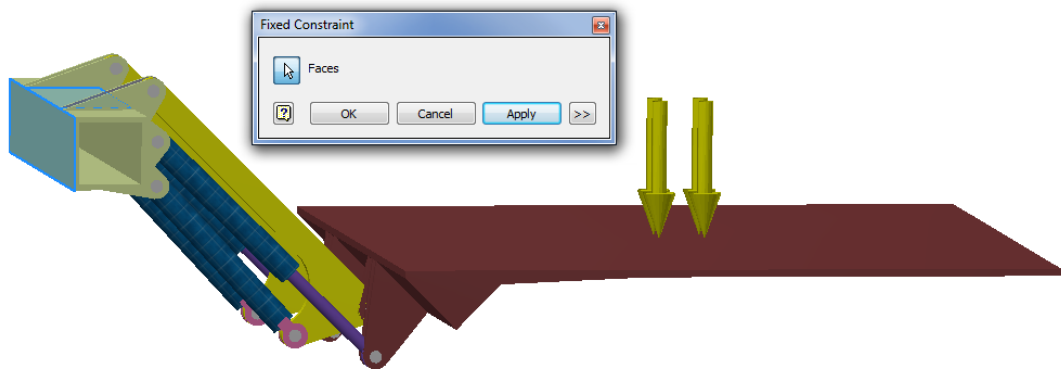


Figure 6.9 Constraint face at lower loading position

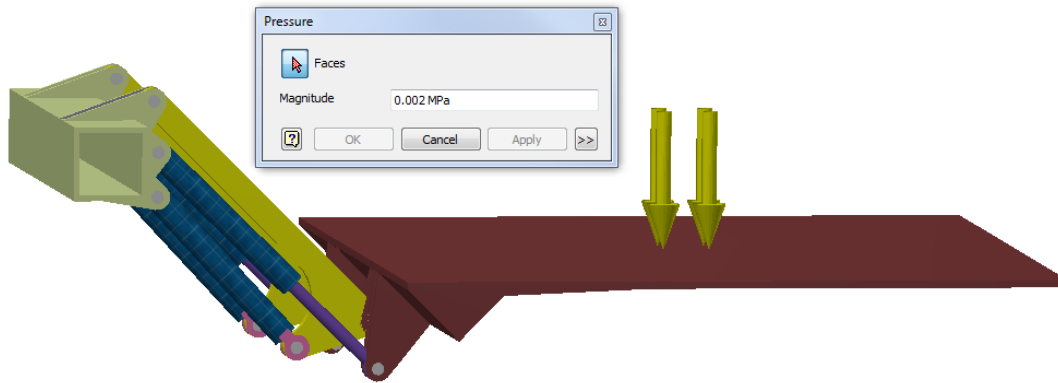


Figure 6.10 Applying loads

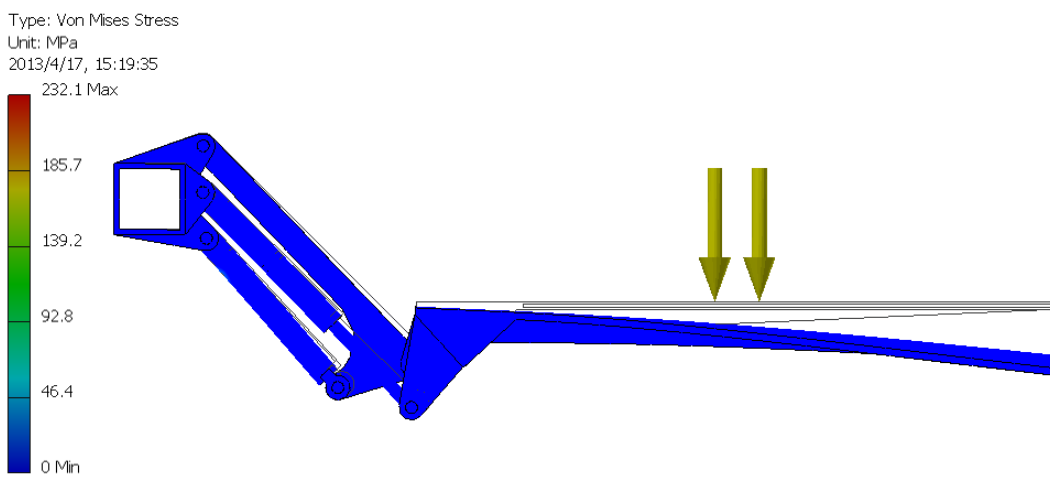


Figure 6.11 Result of Von Mises stress

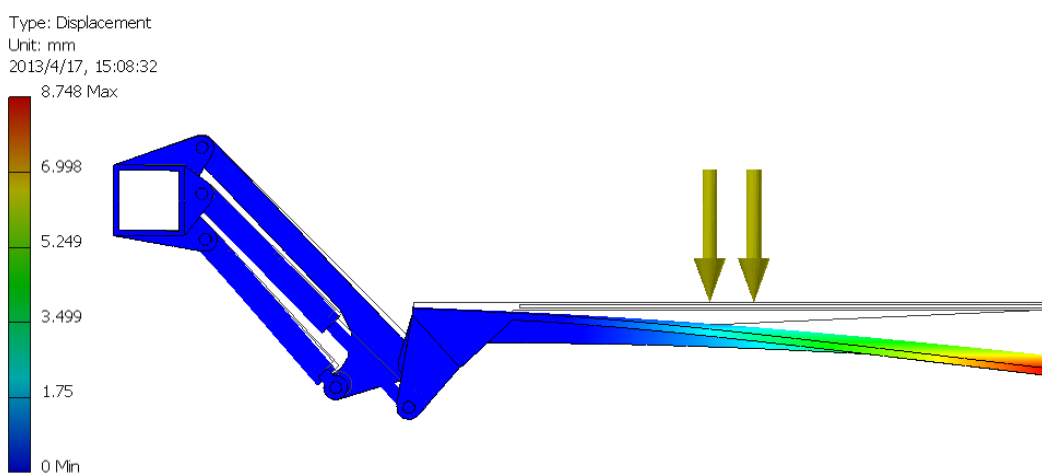


Figure 6.12 Result of displacement at lower loading position

### 6.3 Conclusion of stress analysis of modified tail lift

In comparison with the first set of tail lift, the modified tail lift has less stress in its structure. The maximum stress in mechanical system is 232.1 MPa. So the modified tail lift has greatly been improved in the strength. The maximum stress and deformation both meet the requirement of design. Until now, the whole mechanical system of tail lift has been accomplished well.

## 7 Conclusions

A mechanical system which includes the board, rocker, pin, hydraulic cylinder and the frame has been created according the dimensions of a specific type truck.

The foundation drawing was built and the models of parts can be derived from it. So it is easy to modify the dimensions of design parameters.

On the basis of the results of Inventor dynamic simulation and FEA the following conclusions can be drawn:

1. The whole mechanical system can realize the required motions.
2. The materials of all components can meet the needs of strength.

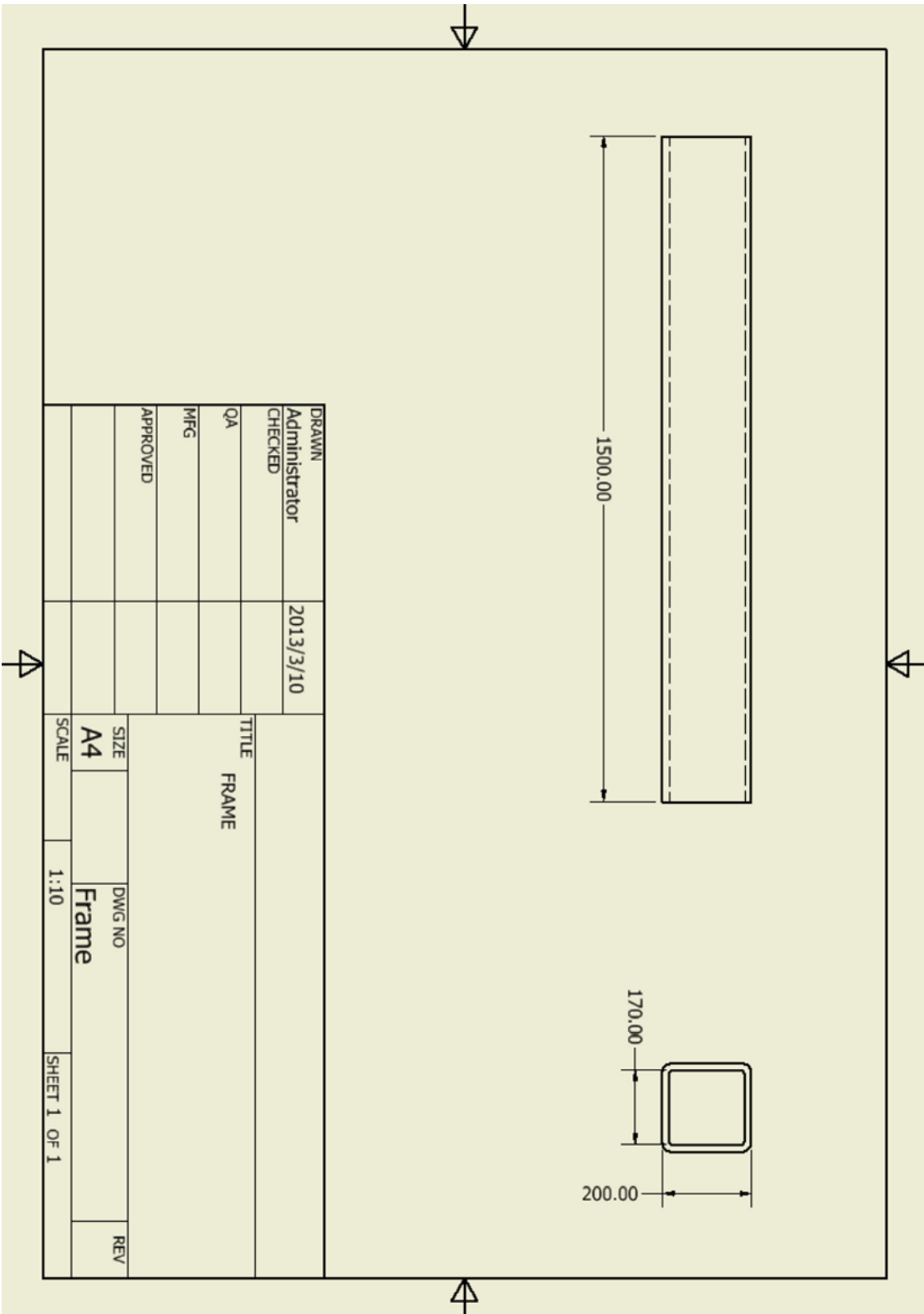
This design process can provide a systematic method for creating the mechanical system of tail lift. It is relatively reliable owing to its stress analysis. Especially by using foundation drawing, it is easy for designers to revise the existing design according to the parameters which will be changed.

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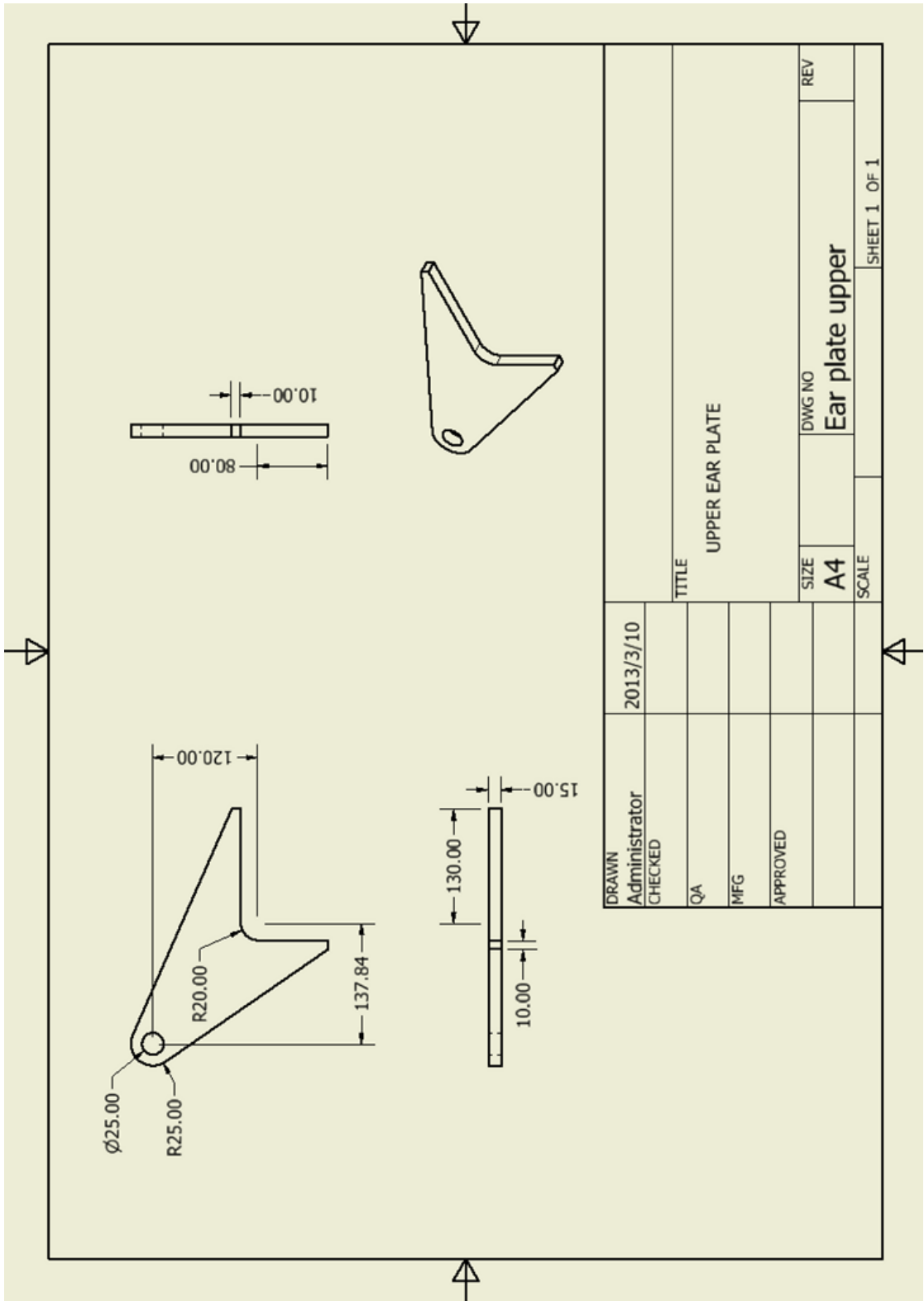
Appendix A Detail drawings of the main components of tail lift

A-1 Detail drawing of frame



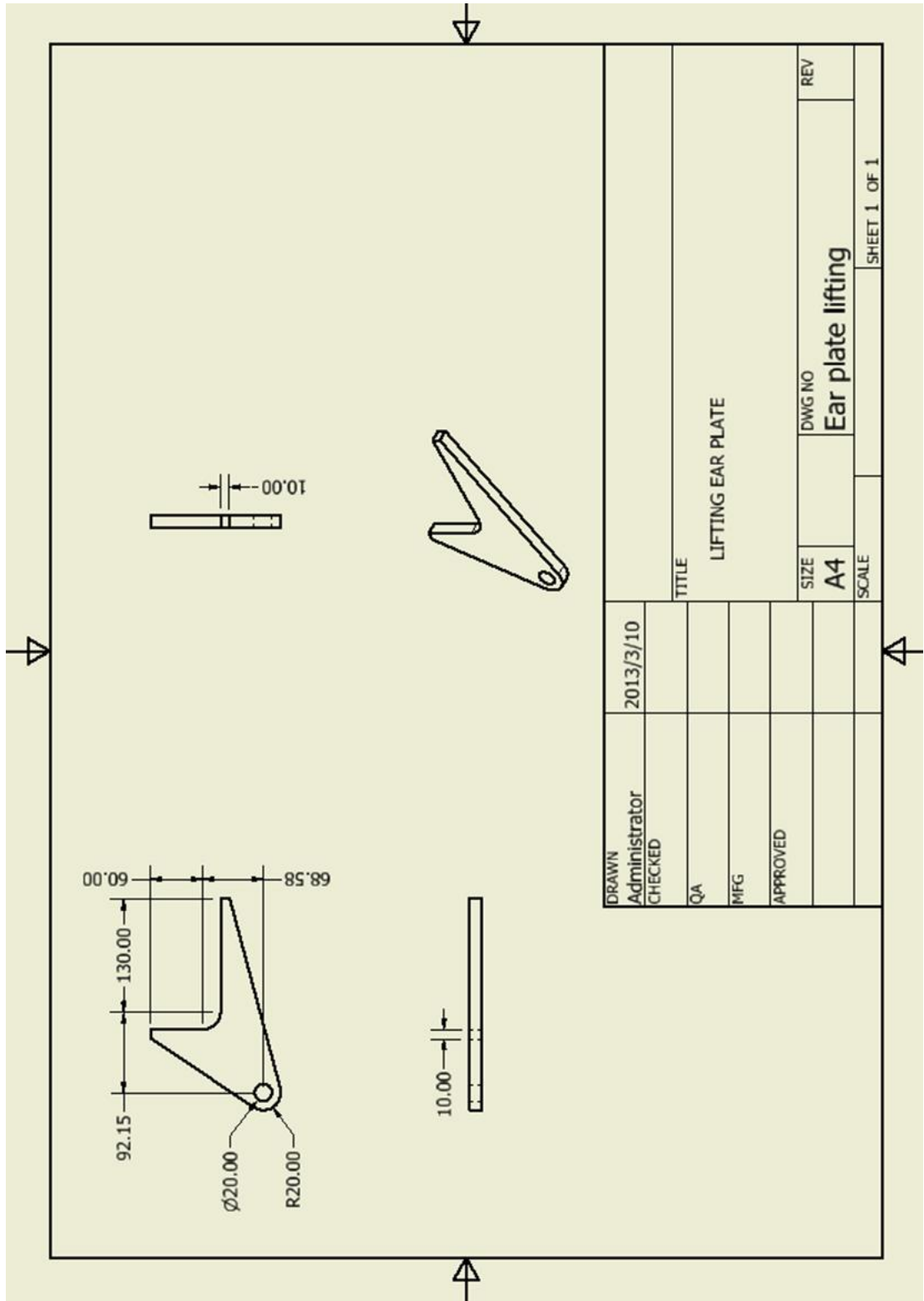


A-2 Detail drawing of ear plate upper

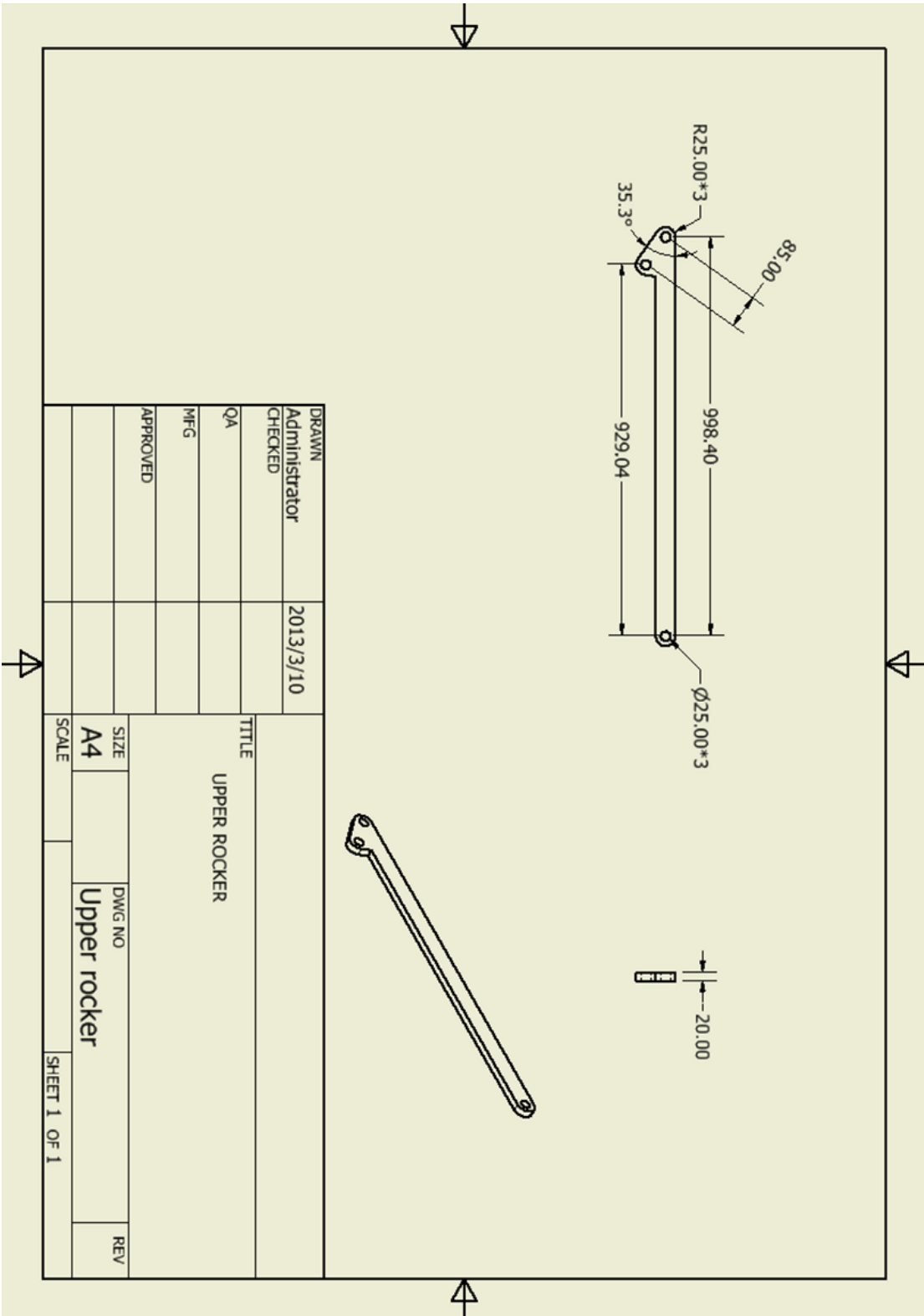




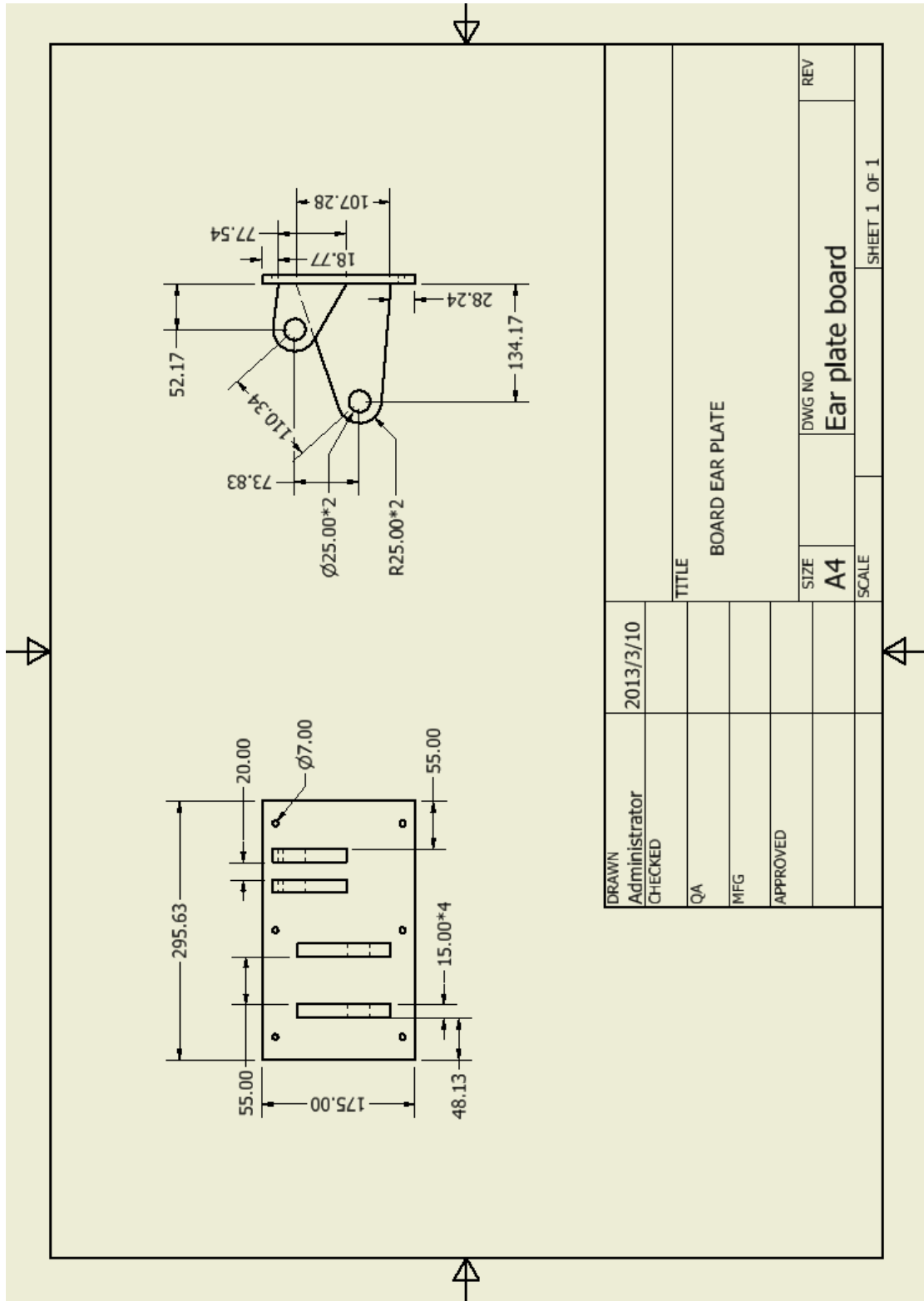
A-4 Detail drawing of ear plate lifting



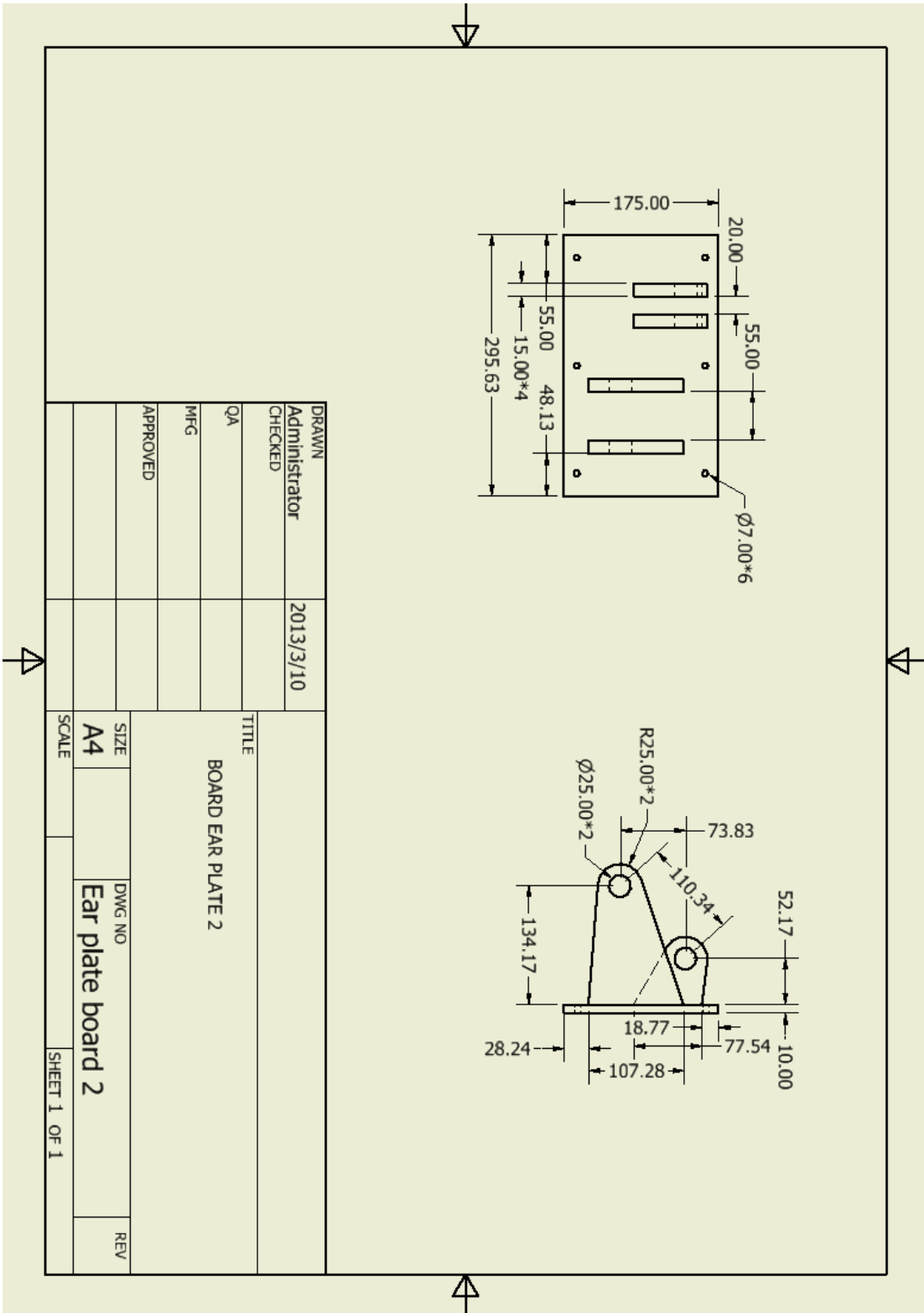
A-5 Detail drawing of upper rocker



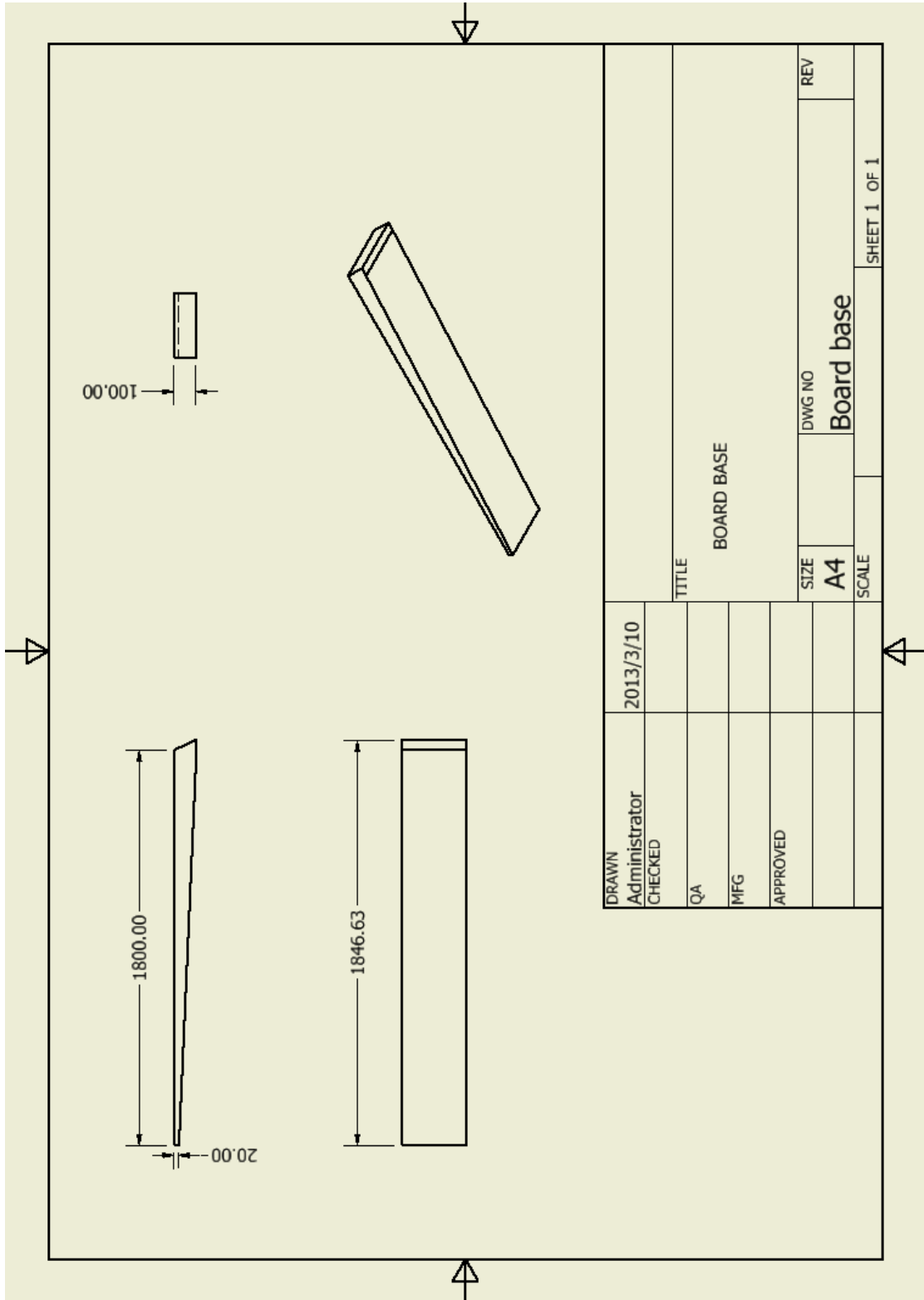
A-6 Detail drawing of ear plate board



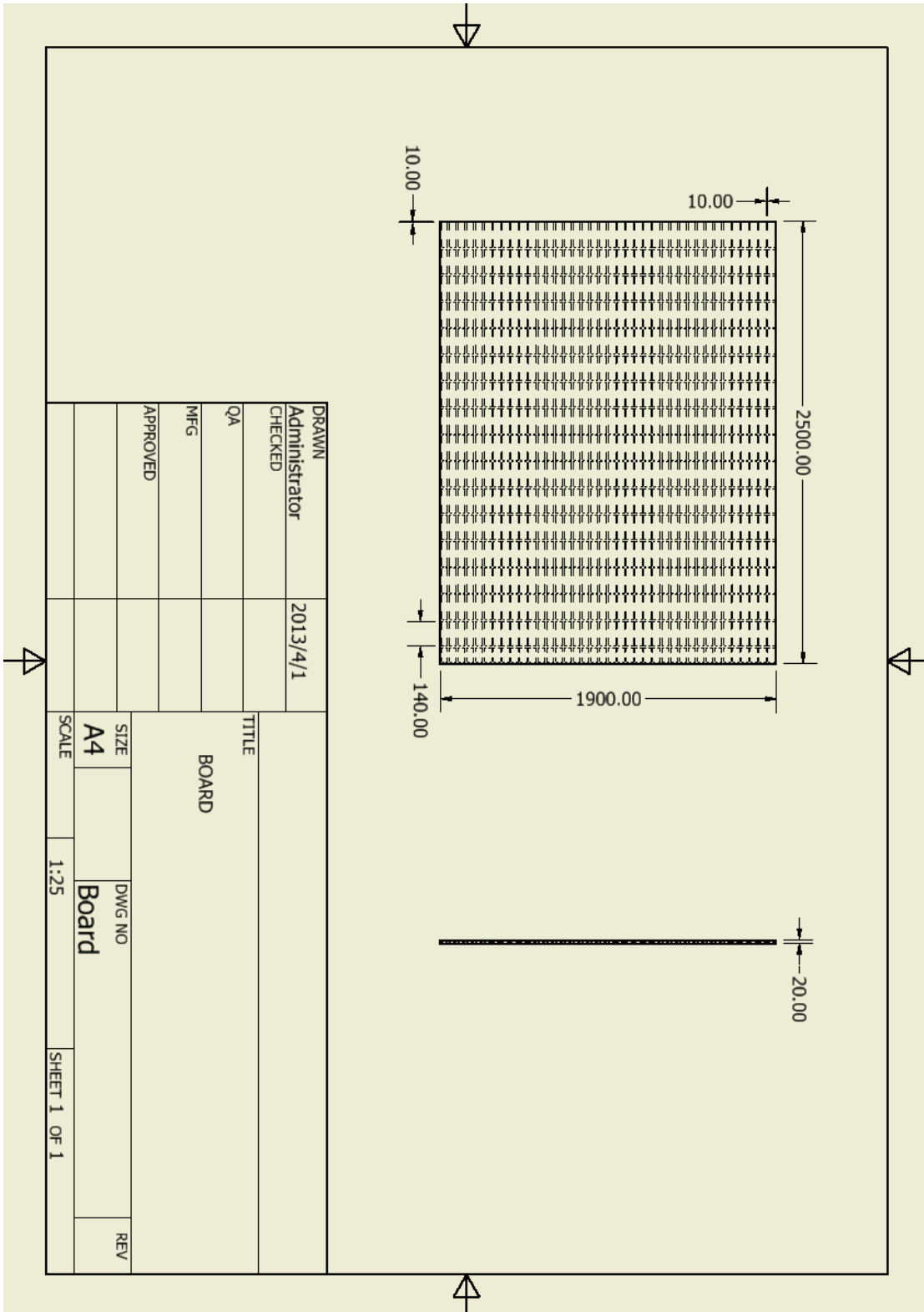
A-7 Detail drawing of ear plate board2



A-8 Detail drawing of board base



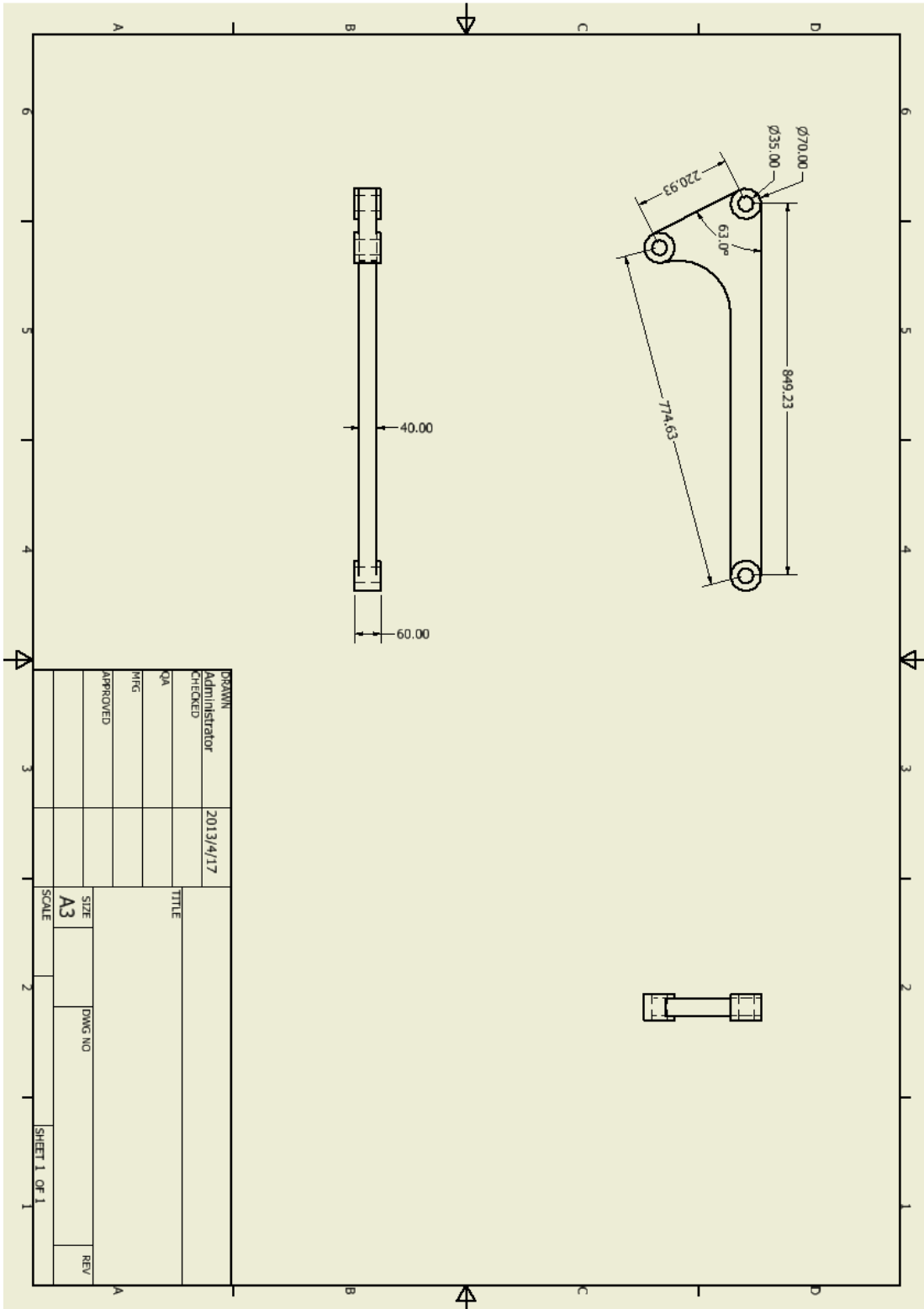
A-9 Detail drawing of board







B-2 Detail drawing of upper rocker



B-3 Detail drawing of board

