International Journal of Enhanced Research in Science Technology & Engineering, ISSN: 2319-7463
Vol. 4 Issue 2, Feb.-2015, pp: (121-134), Impact Factor: 1.252, Available online at: www.erpublications.com

Design of rear lifts for vehicles

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Abstract: Rear lifts are widely used in the world. They are the most common equipment's fixed on vehicles, which are used for loading and unloading cargoes because they make transportation to be more economical, more efficient and safer. The mechanical system is certainly the most important part of rear lift because it carries the load. So the safety of the system should be guaranteed. The goal of the research is to design a reasonable and safe mechanical system of rear lift for a certain type of vehicles.
Studying the principle mechanism was the first step. When this process was finished, creating the models for all components in solid works software was the second step. 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. There are two sets of mechanical systems in this research. 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 limited. The whole mechanical system of rear lift has been accomplished.

Keywords: Rear lift, four-bar linkage, solid works, stress analysis, modification vehicle.

1. INTRODUCTION

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.
Locomotion of a mechanical system consisting of two rigid bodies, a main body and a rear, connected by a cylindrical joint, is considered. The system moves in a resistive fluid and is controlled by periodic angular oscillations of the rear relative to the main body.

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 9^0~10^0. Until now, this type production is on the active service in the region of Southeast Asia and Japan. [1]. 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 10^0.

This type of production was mainly applied in United States and Southeast Asia. [1].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 rear lift automatically. In this way, the "memory" leads the lift process to be more safe and smooth. The dig angle usually was 8^0~10^0, in addition, if it doubles as the door, the dig angle could also be less than 8^0. In present, this kind of production is commonly applied in Europe and America. [1]. 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 6^0, or even less than 6^0. [5].The sustained rapid development, especially the rapid development in the logistics and transport industry provide the extremely colossal space for vehicle rear lift gate. Four common categories of rear lift prevailing in European countries are shown in Figure 1.1. [2]

Fig. 1.1: Four common categories of rear lift

The aim of this study

The whole rear lift system is shown in Figure 1.2, 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 rear 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, piping system and transmission mechanism which consists of hydraulic cylinders, rocker and square beam.

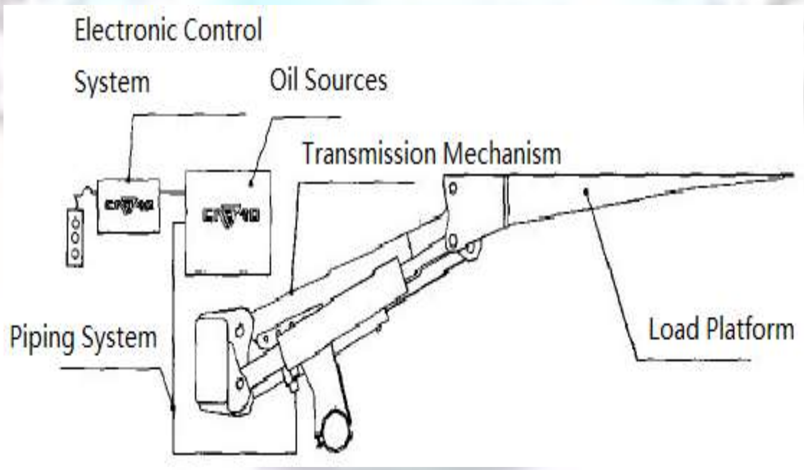


Fig. 1.2: Rear lift system

The original objectives of this research about rear lifts are:

1. To be more economical: Installing a rear lift can greatly reduce the cost of employment.
2. To be more efficient: By using rear 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 rear 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 research, the following work should be done:

1. Analyzing the mechanical principle of rear lift.
2. Calculating the basic dimensions of rear lift for a special type of vehicle.
3. Creating the three-dimensional model with solid work 2011.
4. Assembling the rear lift and checking its feasibility.

2. ANALYSIS OF MECHANICAL PRINCIPLE OF REAR LIFT

2.1 Basic background knowledge used in rear lift

2.1.1 Planar four-bar linkage

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].

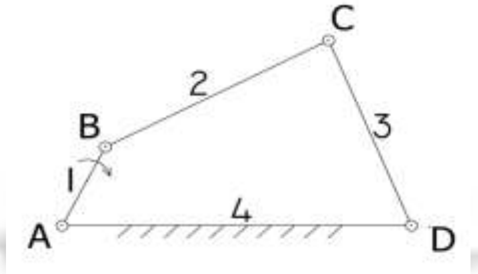


Fig. 2.1 Closed-loop four-bar linkage mechanisms

A Link which is free to rotate t. 2.1: 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 double-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.

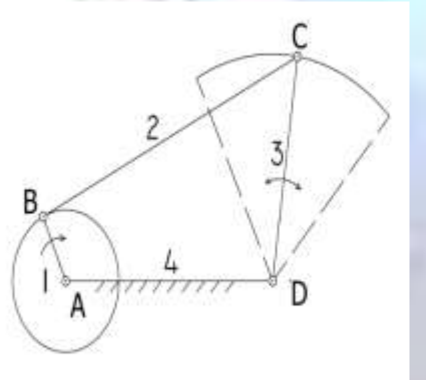


Fig. 2.2: Crank-rocker mechanism

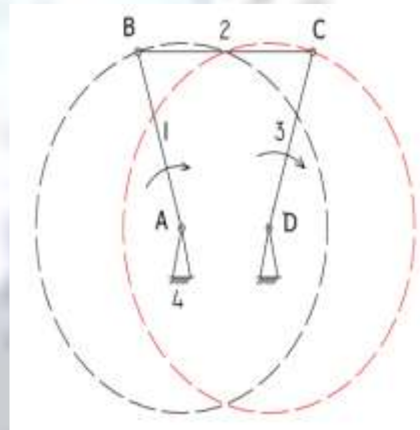


Fig. 2.3: Double-crank mechanism

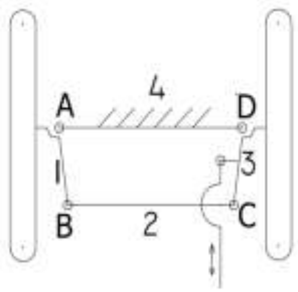


Fig. 2.4: Double-rocker mechanism

2.1.2 Classification and Grashof's theorem

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

1. A crank: can rotate a full 360 degrees;
2. A rocker: can rotate though a limited range of angles which does not include 0^0 or 180^0;

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

$$s + 1 \leq p + q$$

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+lv.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

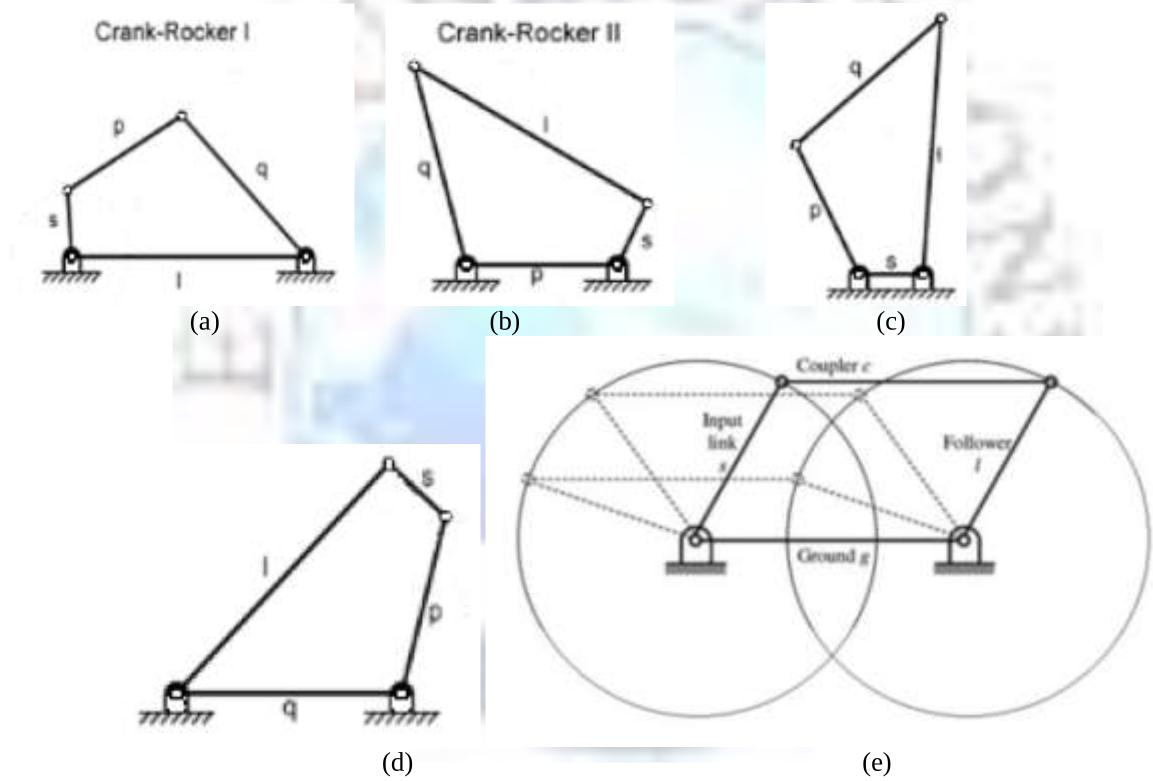


Fig. 2.5: Types of four bar Linkage [5]

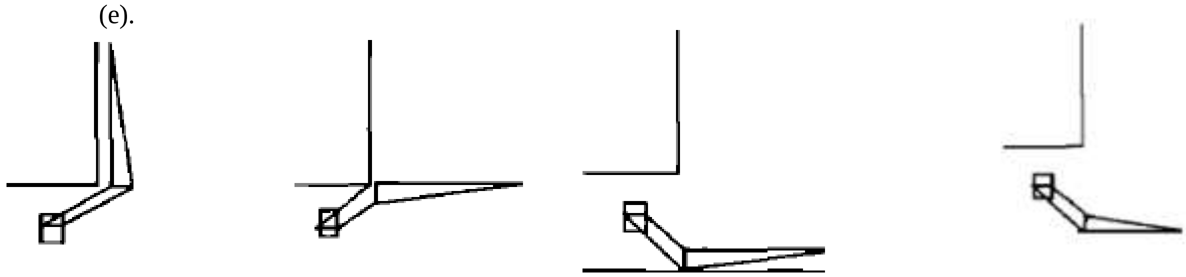
Parallelogram mechanism, in which two equal length links are not adjacent. All four inversions of this mechanism yield double crank mechanisms (Figure. 2.5(e)). In parallelogram mechanism, the opposite linkages are synchronous.

2.2 Decomposition of movement of rear lift

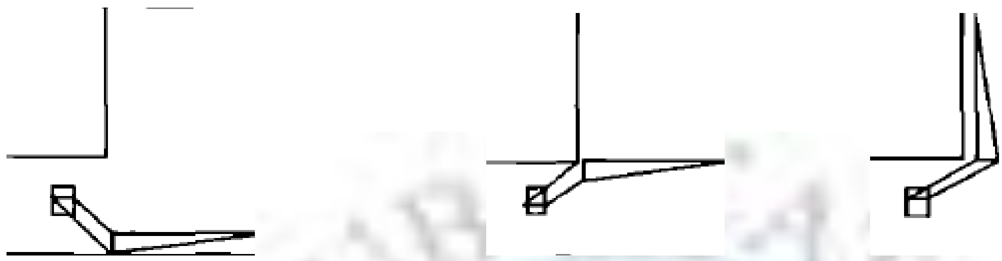
The rear board lift's movement can be decomposed to three actions,

i. Rollover motion: opening and closing the rearboard are realized by way of rollover motion. In this process the rear board rotates by the lower edge of the rear board. The rear board 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 rear board involve downloading and uploading, which are presented in Figure 2.6 from position (b) to (c) and from position (e) to (f) ,respectively.

iii. Subsidence motion: this rollover motion makes the rear board 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 horzontal (c) Lift down (d) Touch ground



(e) Take off (f) Lift up (g) Close back

Fig. 2.6: Movement proves

3. DESIGN PROCESS

3.1 Design requirements

3.1.1 Function requirements

The rear 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 below shows the partial definitions of a common vehicle 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]

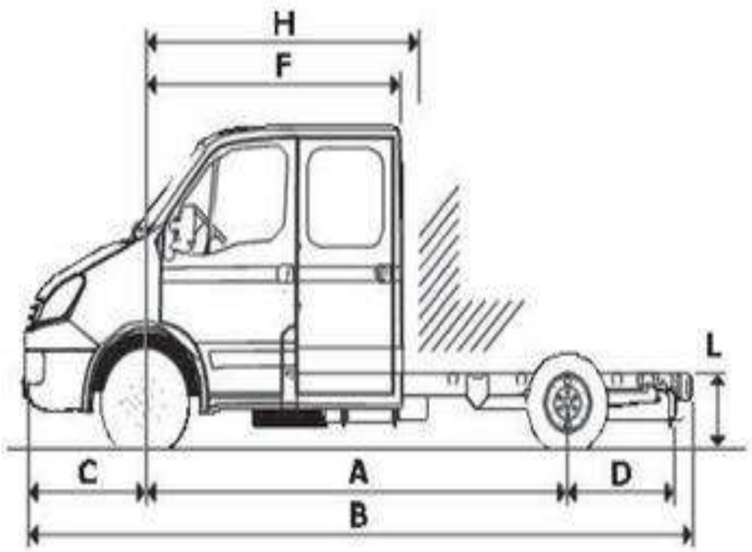


Fig. 3.1: Instruction of Vehicle measurements

3.2.1 Parameters of Design
3.2.2 sample

The basic parameters of vehicleare showed in Table 3.1.

Table 3.1 Parameters of vehicle

TYPE	CARGO SIZE		REAR HANGOVER HEIGHT	CARGO BOTTOM HEIGHT	GIRDER SPACE
Vehicle	HEIGHT	WIDTH	860	1300	870
	2600	2500			

The simple diagrammatic drawing (Figure 3.2) is showing the parameters of Table 3.1, the height of vehicle cargo is 2600 mm, and the cargo width is 2500 mm. And the height form bottom of cargo to ground is 1300 mm, the height of rear overhang is 860 mm and the girder space is 870 mm which are show in below.

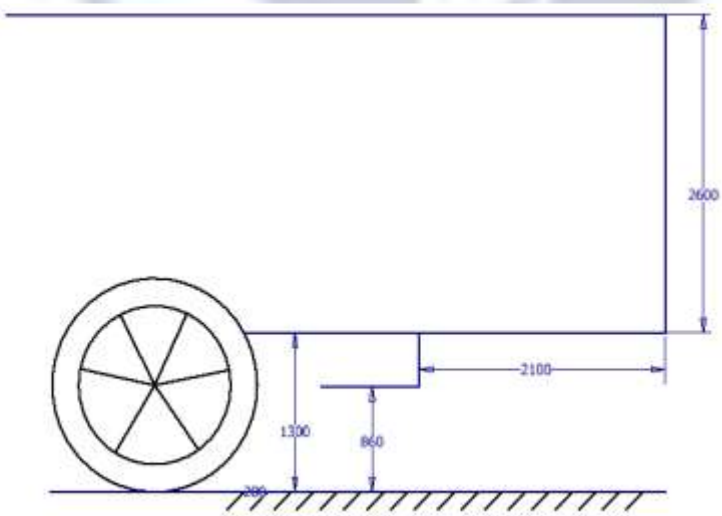


Fig. 3.2: Diagrammatic drawing of vehicle from profile projection

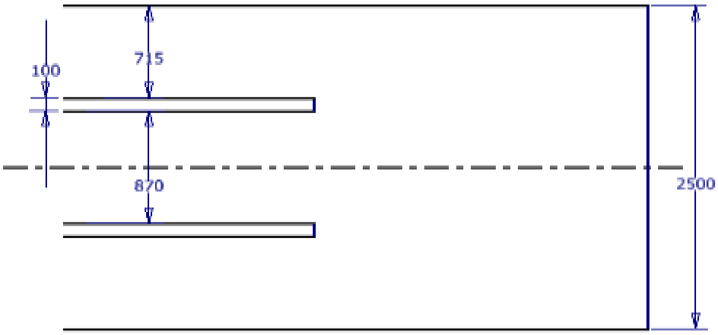


Fig. 3.3: Diagrammatic drawing of vehicle from bottom projection

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 3.4, upper and lower rods are abstracted as a parallelogram mechanism and the both extreme positions are shown on it and $H_0$is the height of the rear 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 βwhich is normally approximately 40^0 to 50^0.

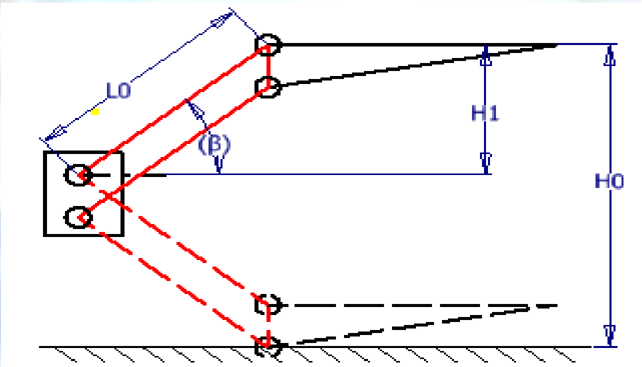


Fig. 3.4: Determination of theoretical lengths

When the vehicle is selected, the height is determined which is H_0= 1300 mm, from upon Figure 3.2, theoretical length $L_0$is,

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

$H_1$is the vertical distance from cabin floor plant to the bracket hinge point of the upper rod; β 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 β. Then

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

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

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

Then

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

When β is $40^0 \sim 50^0$, sin β = 0.64~0.77, H_0= 1300mm, then

L_0= (0.70~0.59) H_0 = 910~767[mm]

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

L_0 = 850 [mm], then C = $0.14L_0$=120[mm]

International Journal of Enhanced Research in Science Technology & Engineering, ISSN: 2319-7463
Vol. 4 Issue 2, Feb.-2015, pp: (121-134), Impact Factor: 1.252, Available online at: www.erpublications.com

4. MODELING IN SOLIDWORK 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 rear 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 rerating 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 rearboard 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.

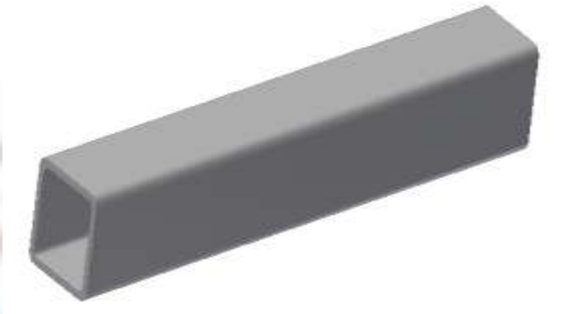
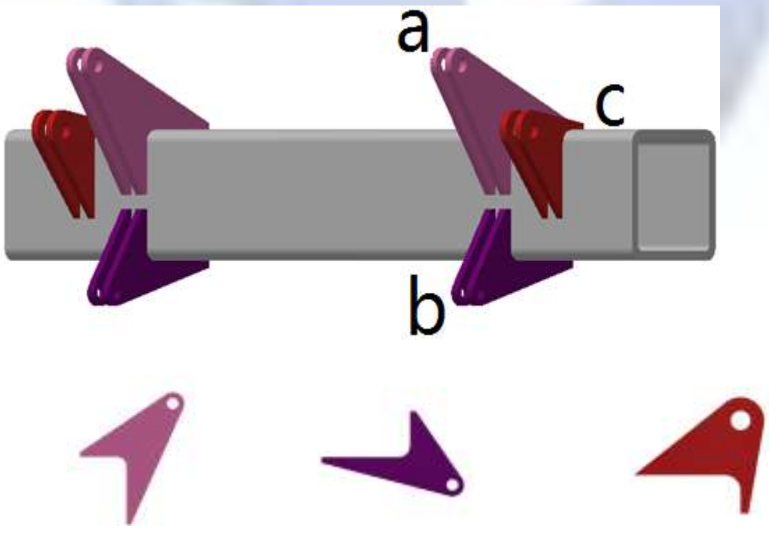


Fig. 4.1: Structure of frame

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;



a) b) c)

Fig. 4.2: Ear Plate a) upper ear plate b) lifting ear plate c) rotating ear plate

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.

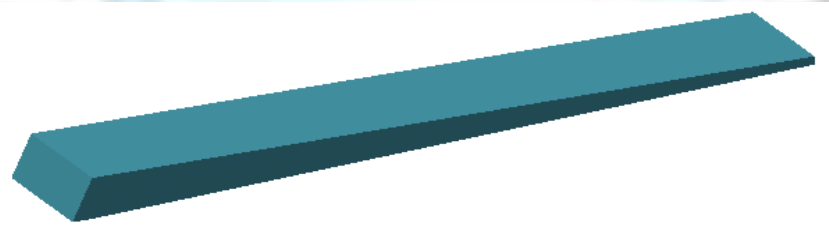
Fig. 4.3: Upper rocker

4.5 Modeling of Ear plate of board

Fig. 4.4: Ear plate of board1 **Fig.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.

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



Fig, 4.6: 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.

4.7 Modeling of board
The board was designed not only a work plate but also functioning as a rear gate. Therefore the parameters of board are collected from container, the size of board was set as 2500*1900*20[mm^3]. 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. Its mass (257.45 kg) is generated by solid work software.

Fig. 4.7: Board

In order to further reduce its mass, the hollow structure was designed. Then, the mass was decreased to 162.5 kg.

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 rear lift, Steel AISI 4130 (R683/ Ⅱ2) was chosen as material of pistons and hydraulic cylinders. Because Steel AISI 4130 (R683/ Ⅱ2) 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 harden ability, and 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.

Name		Steel AISI 1020	Steel AISI 4130	Aluminum 6061
General	Mass Density	7.858 g/cm^3	7.799 g/cm^3	2.71 g/cm^3
	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/(mK)	21.6173 W/(mK)	167 W/(mK)
	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 SOLIDWORK SOFTWARE

5.1 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 analyses as an indicator of success or failure in analyses.

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

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 analyses and hence FOS will more commonly be based on yield limit. [13]

5.2 Stress analysis workflow

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

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5.3 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 stress on each board base is:

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

5.4 Stress analysis of the whole rear lift

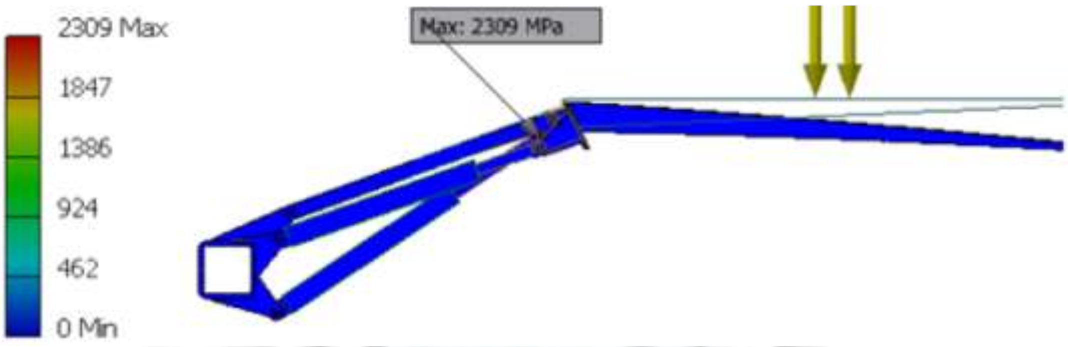

Fig. 5.1: Stress Result

By operating stress analysis in program, the Stress result shows, in Figure 5.1, 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, the maximum stress will occur in the hinge which joins the rotating hydraulic cylinder and the board ear plate as shown in Figure 5.1. 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 rear 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 rear 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 rear Lift

6.1 Modification of rear 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 rear 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.

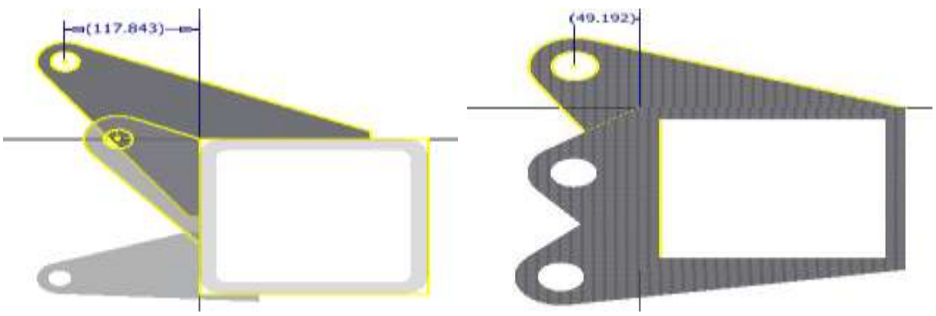

Fig. 6.1: The comparison of original and modified frame

2) For all joint pins the diameter was changed from 25 mm to 35 mm in order to de-crease 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.

Fig. 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.

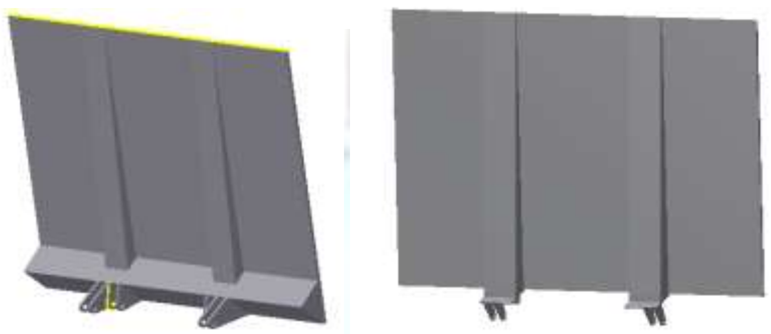

Fig. 6.3: The comparison of original and modified board

6.2 Stress Analysis of modified rear lift
Figures 6.4 and 6.5, they show the constraint, applying loads and the results of stress and displacement respectively at upper horizontal position.

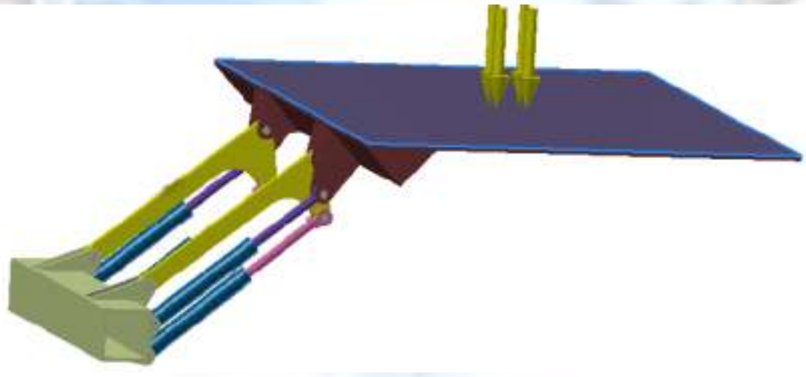

Fig. 6.4: Constraint face at upper loading position, applying loads

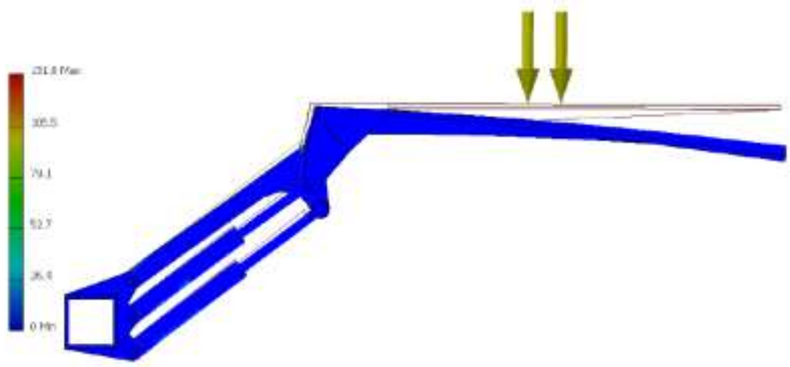

Fig. 6.5: Stress Result

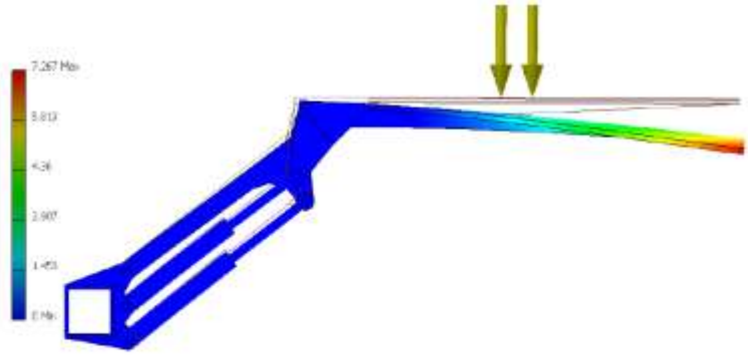

Fig. 6.6: Result of displacement at upper loading position

From Figure 6.7 to Figure 6.10, they show the meshed model and mesh settings, the constraint, applying loads and the results of stress and displacement respectively at lower horizontal position.

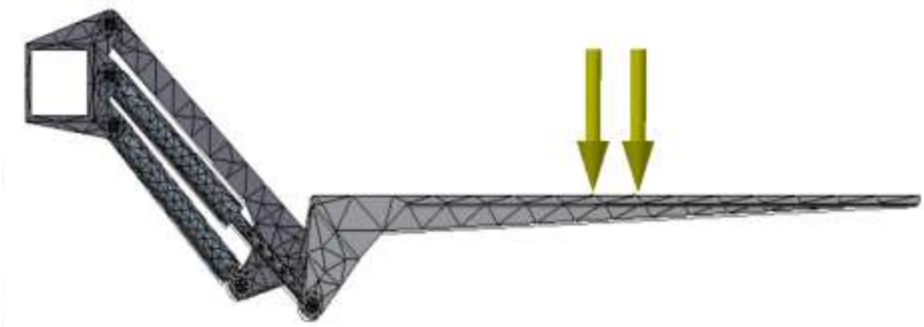

Fig. 6.7: The meshed model and mesh settings

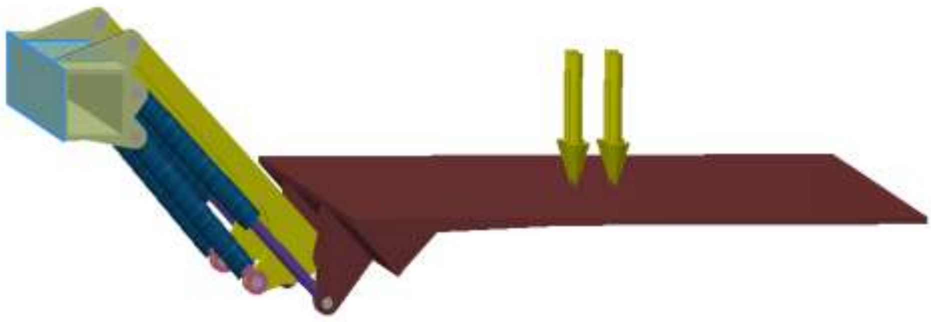

Fig. 6.8: Constraint face at lower loading position & Applying loads

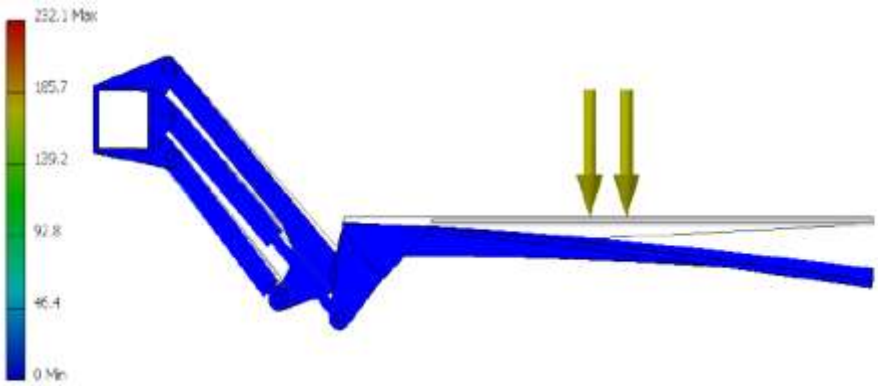

Fig. 6.9: Stress Result

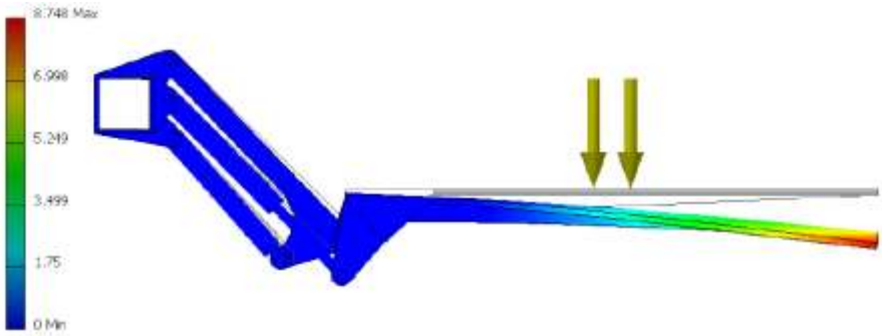

Fig. 6.10: Result of displacement at lower loading position

6.3 Conclusion of stress analysis of modified rear lift
In comparison with the first set of rear lift, the modified rear lift has less stress in its structure. The maximum stress in mechanical system is 232.1 MPa. So the modified rear 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 rear lift has been accomplished well.

7. Conclusions

A mechanical system that includes the board, rocker, pin, hydraulic cylinder and the frame has been created according the dimensions of specific type vehicles. 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 solid work dynamic simulation 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 rear 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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