

DESIGN OF HYDRAULIC CYLINDER AND BODY MODIFICATION FOR MAXIMA CARGO BAJAJ



Gashaw Hussein Ali

A Thesis Submitted to

The Department of Mechanical Engineering

School of Mechanical, Chemical and Materials Engineering

Presented in Partial Fulfilment of the Requirement for the degree of Master of
Science in Automotive Engineering

Office of Graduate Studies

Adama Science and Technology University

January, 2024 G.C

Adama, Ethiopia

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Advisor: Amana Wako (PhD)

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CANDIDATE'S DECLARATION

I hereby declare that this Master Thesis entitled “Design of Hydraulic Cylinder and Body Modification for Maxima Cargo Bajaj” is my original work. That is, it has not been submitted for the award of any degree, diploma or certificate in any other university. All sources of material that are used for this thesis have been duly acknowledged through citation.

Gashaw Hussein Ali

Name of the student

Signature

Date

RECOMMENDATION

I, the advisors of this thesis, hereby certify that I have read the revised version of the thesis entitled “Design of Hydraulic Cylinder and Body Modification for Maxima Cargo Bajaj” prepared under our guidance by Gashaw Hussein submitted in partial fulfilment of the requirements for the degree of Master’s of Science in Automotive Engineering. Therefore, I recommend the submission of revised version of the thesis to the department following the applicable procedures.

Amana Wako (PhD)
Advisor

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APPROVAL OF BOARD OF EXAMINERS

I, the advisor of the thesis entitled “Design of Hydraulic Cylinder and Body Modification for Maxima Cargo Bajaj” and developed by Gashaw Hussein, hereby certify that the recommendation and suggestions made by the board of examiners are appropriately incorporated into the final version of the thesis.

Amana Wako (PhD)

Advisor

Signature

Date

We, the undersigned, members of the Board of Examiners of the thesis by Gashaw Hussein have read and evaluated the thesis entitled “Design of Hydraulic Cylinder and Body Modification for Maxima Cargo Bajaj” and examined the candidate during open defense. This is, therefore, to certify that the thesis is accepted for partial fulfilment of the requirement of the degree of Master of Science in Automotive Engineering.

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ACKNOWLEDGEMENT

Thanks to God, I managed to complete the study after gone through so many challenges. All praise and many- many thanks to God who gave the strength and confident to me for completing this thesis work. I would like to express my genuine thanks to my advisor Dr. Aman Wako for his valuable advices, constant motivation and guidance's during my study. I would like to express my appreciation to Hora Bajaj Adama Although, I would like to express my gratitude to Ato Getinet Abebayehu (mechanical engineering) at hora factory, for providing all the work and necessary information to produced and use hydraulic cylinder on a vehicle. Generally, I would like to extend my thanks for all the above people and those who are not mentioned here but contribute their part a lot to towards the success of this thesis.

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ACRONYMS

CAD	Computer-Aided Design
CAE	Computer-Aided Engineering
CAM	Computer-Aided Manufacturing
CATIA	Computer-Aided Three-Dimensional Interactive application
FEA	Finite Element Analysis
EN8	Unalloyed Medium Carbon

ABSTRACT

Three-wheelers, commonly known as Bajaj's, can be found all over the world, specifically in Asia and Africa. And they are used for transporting people or freight. In our Ethiopia, three-wheeler vehicles are used as a mode of transportation and freight sector, particularly in most of all the cities. The objective of this thesis was focused on designing cylinders and body modification for maxima cargo Bajaj. Mainly to design the hydraulic cylinder for lifting purposes during the unloading time and also to develop a hydraulic cylinder for 3D maxima cargo Bajaj model using CATIA software. Those cargo Bajaj were used to carry construction product, perishable goods, and all farm products. Almost all cargo Bajaj that are assembled in Hora Adama do not have a hydraulic lifter for unloading materials. Due to this, they cannot give a fast service during unloading time. During the study, lifting analysis maximum Von Misses stress and total deformation was also carried out by ANSYS Explicit Dynamic Analysis for modelling and analysis of the hydraulic cylinder. The model was analyzed by explicit dynamic analysis using ANSYS Workbench 16. To fulfil the purpose of this thesis, work to improve the performance of the hydraulic cylinder in terms of load capacity and service life. Modeling of the mechanism using Catia, making stress analysis by ANSYS and finally making simulation using Catia V5 software. Finally, a double acting hydraulic cylinder was successfully designed and analyzed. Therefore, the design of a double acting hydraulic cylinder was achieved and is ready for manufacturing.

Keywords: *Bajaj, deformation, finite element analysis, hydraulic cylinder, stress.*

CHAPTER ONE

INTRODUCTION

1.1. Background of the study

Three-wheelers, commonly known as Bajaj's, can be found all over the world, specifically in Asia and Africa. And they are used for transporting persons or freight. In Ethiopia, three-wheeler vehicles are used in transportation and freight sector particularly in cities of regional states, among others, Dire Dawa, Harar, Jigjiga, Bahir Dar, Hawassa, Adama and Mekele. Most of the three-wheeler's cargo Bajaj in Ethiopia are assembled in Hora Adama. These vehicles are providing transport services, mainly for public transport, and at a time for goods. In state cities in Ethiopia, they are becoming more and more common in the transportation and freight sector.

A Fluid pressurized is a designed that produces high-pressure hydraulic fluid. The mechanism is very simple. Fluid (oil) entering the system is pressurized and acts on a confined piston that drives a smaller diameter plunger or piston to deliver a fixed volume of fluid. This increases the outlet pressure, which is significantly higher than the inlet pressure. The cylinder is designed for use with compressed air, hydraulic oil, and refractory fluids. In some cases, special seals may be required. Its bodies follow strict quality control standards, and precision manufacturing ensures that all tubes meet exacting standards for straightness, roundness, and surface finish. Steel tubes have a surface finish that minimizes internal friction and increases seal life (Khan, Design and Manufacturing of Hydraulic Presses).

Hydraulic driven used composed connected systems used aligned used linear force and movement. Hydraulic cylinders are driven by externally pressurized hydraulic fluid. It is widely used in equipment and machinery such as construction machinery such as excavators and bulldozers, and transportation machinery such as transportation cranes and telescopic carts, and is mainly composed of cylinder tubes, pistons, and cylinder rods. The piston, located inside the cylinder, is connected to the cylinder rod by a screw or nut. The piston moves back and forth, and the pushing and pulling forces are perfectly aligned along the axis of the rod, producing a linear force. In most applications where bolts and nuts are used, the clamping force provided by bolt/nut torque plays a critical role and is a critical factor in the structural integrity of the connection.

Hydraulic cylinder

In the thesis project a double acting hydraulic cylinder was used and not a hydraulic motor. For this reason, the focus of the theory is on the formula required for designing systems with hydraulic cylinders.

1.2. Statement of the problem

In Ethiopia, three-wheeler cargo Bajaj is used for the transportation of light and medium weight cargos. This cargo Bajaj is used to carry construction product, perishable goods, farm products, and vegetable product and so on. Almost, all cargo Bajaj that assembled in Ethiopia do not have hydraulic lifter for unloading of materials due to this they cannot give fast service during unloading time.

In order to reduce this wastage of time and labour cost it is necessary to design a hydraulic cylinder which can be used during the unloading of any product.

1.3. Objectives of the study

1.3.1. General objective

The general objective of this thesis work is to design of Hydraulic Cylinder and Body modification for Maxima Cargo Bajaj.

1.3.2. The specific objectives

The main objectives of this thesis were achieved through the following specific objectives.

- Develop hydraulic cylinder and 3D maxima cargo Bajaj model using Catia software.
- To analyse the parts for strength using software.
- To simulate hydraulic cylinder and body modification for Maxima Cargo Bajaj..
- To analyse working performance.

1.4. Significance of study

Since sufficient studies have not been conducted on technology transfer in Ethiopia. So, the outcome of this study would give input for Bajaj manufacturers especially during assembling process. Therefore, this study will help the country at large and freight sector particularly when the outcome become evaluated and implemented.

Economic benefit

- ❖ Saving labour cost

- ❖ Optimization of automotive industry usage and
- ❖ improving market share of the experiment Bajaj type

1.5. Scope of the study

Basically, three-wheeler maxima cargo Bajaj's are found all over the country and they are playing a significant role in providing transportation of goods. This thesis work covers design and analysis of hydraulic cylinder. Lifting impact and analysis of preferred shape of hydraulic cylinder for maxima cargo Bajaj were done with the help of CATIA software in order to get best design for maxima cargo Bajaj.

1.6. Limitation of the study

There was a limitation to find out the literature has been done on this title

1.7. Organization of the thesis

This study consists of five chapters. The first chapter is an introduction part of the study. It includes: General Background on the study area, Problem of the statement, Aim of the study, Significance of the study, Scope of the study, limitation of the study, and Organization of the thesis. Chapter two focuses on the literature review based on the previous related works. Chapter three describes materials and methods that followed to fulfil this thesis work. Chapter four consists of results and discussion. Chapter five consists of conclusions and recommendations for future work areas.

CHAPTER TWO

LITERATURE REVIEW

Previous related works to hydraulic cylinder were reviewed in this chapter. Different researches were done on different hydraulic cylinder analysis in different material, method and experiment in the world in terms of cost effective and reasonable for uses. This was done by using different approaches

2.1. Previous Works

The researcher reported utilizing a variety of materials to examine the three-stage hydraulic cylinder. The ideal materials and stainless steel, the conventional material, are assessed under the same load, and the outcomes are compared. The hydraulic cylinder Which is found in dump trucks, there may be effectively designed using Pro/E and ANSYS and rapidly modelled and analysed. The investigator reached the findings in his investigation. Article stating that EN8 exhibits low deformation values in contrast to other enhanced and standard materials. The digital three-dimensional model of the multistage hydraulic cylinder was produced. This technique is crucial for reducing the cost of development and the design cycle, and improving the calibre of design (Vanka, 2015).

To design a concept prototype of a hand-held tool and to use components in the products that has designed by the company, in order to compare, the two basic designs were chosen those are: - an electro-mechanical and electro hydraulic solution. The result showed that, Large amount of information was gathered for comparing solutions and choosing the best one. For this purpose, a morphological table was done where different solution concepts were ranked with weighted ratings. Based on these rating an electro-mechanical and an electro-hydraulic solution were compared. The electro-hydraulic solution was chosen for further development (Mikkola, 2014).

(Montero, 2009). Clinical studies and tests of hydraulic cylinders have shown that their load capacities differ from those obtained from simple buckling analysis of ideal systems. However, increasing the friction coefficient at the tip of the tip will change the tip of the actuator; an increase in maximum deflection (primary misalignment) will reduce the ultimate load.

(E., 2009). Sand design models currently prepared in research articles do not include the main ideas in the use of hydraulic cylinders. The reliability of frictionless models (including models)

is a question when comparing both in experimental and theoretical result. The two pin ends of the column support the friction moment of the tight connection between the ends of the bushing. Experimental results also confirm this behaviour. In his case, the researcher can be concluded that all rigidly supported columns can withstand more axial load than that generated by the current sand model.

(I. Marczevska, 2006). At the same time, a numerical algorithm is presented that allows different loading patterns of a specific structure (e.g. hydraulic cylinder) to be applied to a specific Wöhler curve characterizing the fatigue resistance of a specific material. The model used in this paper is based on the specific application of a constitutive model, a mechanical model described for mechanical fatigue analysis. A solution to the cracking problem in the oil port area has been proposed. Calculations must be performed for cylinders subjected to loads different from the experimental data presented in the Wöhler curve. After some design changes, oil penetration into the oil drain connection area can be eliminated.

(Beretta, 2012). The manufacturing processes, operating conditions, and material properties discussed are all requirements for good design of tubular mechanical components subjected to fatigue. The most common approach during development has been the deterministic approach, which sets fixed NDT thresholds associated with fault tolerance limits. However, many uncertainties remain, such as the probability of failure associated with fatigue strength under applied load. This article describes the reliability assessment of tubular machine components with some defects and periodic changes in internal pressure.

(Sh. Molaei, 2015). They are carried out failure analysis of the fractured connecting bolts of a filter press. They investigated failure causes of sixteen connecting bolts on rod and piston of a filter and found probably low tightening torques were used for the assembly and reason for failure of the joints.

(Marin, 2015). The heavy hydraulic cylinder failed during use. A combination of break mechanics and the use of finite element methods shows that the failure was caused by a combination of the fatigue energy life estimate of the connection between the cylinder and the end cap and the life estimate before the accident.

(Fernando, 2015). Analysed two mechanisms of vibration loosening of bolts and identified critical parameters to prevent loosening of bolted joints. Finally, based on the results, it was

determined that a percent of 65 preload the fastener resistance should prevent vibration loosening in all vibration environments.

(John D,2005). They developed procedures for calculating Used to calculate friction coefficients of bolt/nut assemblies and torque characteristics, including when fastener torque is dominant. calculating torque specifications that include cases where the clamping torque is dominant. The new equations developed essentially resulted in a 15 percent improvement in the accuracy of the torque specification calculations.

Based on optimization of a 250-ton four pillar type hydraulic press considering constraints like design, weight and cost. focused on design and optimization of top plate of the press machine, validated the result by using Finite Element method with proper boundary conditions (Akshay Vaishnav, 2016).

(Saleh, 2022), investigated and talked about the machine's theoretical and experimental models in order to determine the precise optimal design analysis and to continue developing the current machine in the shortest amount of time and for the least amount of money. The theoretically predicted and jointly experimental results have been compared with the press structure's design objective. and came to the conclusion that while there is a 43% decline in agreement between these models and the experimental model, the conventional analytical model and the plane frame FEA model agree.

(Ankit H. Parmar, 2014). The main objective of their study was the structural optimization in order to decrease the total mass of hydraulic press while assuring adequate stiffness. By using structural optimization for hydraulic press were proposed in order to reduce mass while assuring adequate stiffness during the study period. By comparing result of old bottom platen and modified final bottom platen they get weight reduced that 2263 Kg to 1303 Kg, Deflection increases 0.055 mm to 0.22 mm but, it was in permissible limit. It was not affect the components. Von-mises stresses increases 104 Mpa to 141 Mpa and it doesn't cross permissible limit and their bottom plate was safe under working condition.

(ThankGod E. Boye, 2017). Briefly discussed about design and analysis of hydraulic cylinders by forming combined equations of analytical and finite element analysis methodology. Longitudinal stresses, burst pressures.

(Monville, 2016). Investigated that incorrect tightening of bolt joint is one of most common reasons of ductile breakage and also reason for most of frequent cause of fatigue failures. Paper focuses on 'hydraulic tensioned' to show how tightening load can be found based on applied pretension load and to suggest a new way for optimization of bolt torque procedures.

(Walaszek, 2006). Studied that ultrasonic method of bolt tightening gives information of stress generated in bolts due to preload. The pre-load provides right measuring of tension generated on bolt joint, independent of manufacturing tolerances. It uses 'bi-wave method' using electromagnetically coupled transducers (EMAT) and data of applied stress on bolts with 5% of uncertainty.

(Eccles, 2014). Presented a new approach to check tightness of bolted connection. This shows a new method to find the torque required to rotate and tighten the joint and then instantaneously doing in opposite direction which allows load available on bolted connection to be ascertained correctly.

(Copeland, 2006) The "Dynamic Tightening of Bolted Connections" study developed a new bolted connection model that enables "dynamic tightening" of bolts in threaded holes through a physical model that behaves nonlinearly in a helical thread pattern. Evaluation of the clamp load on the original connection showed a deviation of 0.15 percent between the original and FEA screws. The average plastic deformation of the joint bearing surface is correlated to the original connection.

(C. Tzeng, 1996). Studied on the dynamic response of composite cylinders subjected to a moving internal pressure.

(J.A, 1998). Also analysed the resonance of stress waves in the cylinder at the instant and location of the pressure front passage, when the velocity of the moving load approaches to the critical velocity.

(B. Faria De, 2004). Use the finite element method to study the vibration of a cylindrical plate due to a moving force or mass. The effects of plate curvature, moving load velocity, and moving mass on the main structure on the dynamic response are analysed. He observed that the dynamic behaviour of the cylinder was strongly influenced by high moving speeds and large moving masses.

(D. Beltman, 2001). Suggested that the flexural waves can result in much higher strain and stress than static loading with the same loading pressure. The wave propagation in a hollow cylinder is analysed for pressure and velocity prescribed at its inner boundary

(P.Moulali, 2014). Investigated on the study of hydraulic and pneumatic system components are modelled with the computer aided static and modal analysis performed by using finite element methods. These studies are addressed to the inverse problem in which the forces acting on the structure are identified by using the vibration responses without the knowledge of the vehicle characteristic.

2.1.1 Types of Hydraulic Cylinders:

Basically, Hydraulic cylinders could be classified into two broad categories.

- i. Single action cylinders.
- ii. Double action cylinders

Single action cylinder: can be defined as "Cylinder in which displacement in one direction is by working fluid pressure and in the other direction by external force. The single-acting cylinder is pressurized at one end only, with the opposite end vented to atmosphere through a breather filter (air cylinder) or vented to a reservoir. The return stroke of the cylinder is accomplished by some external means.

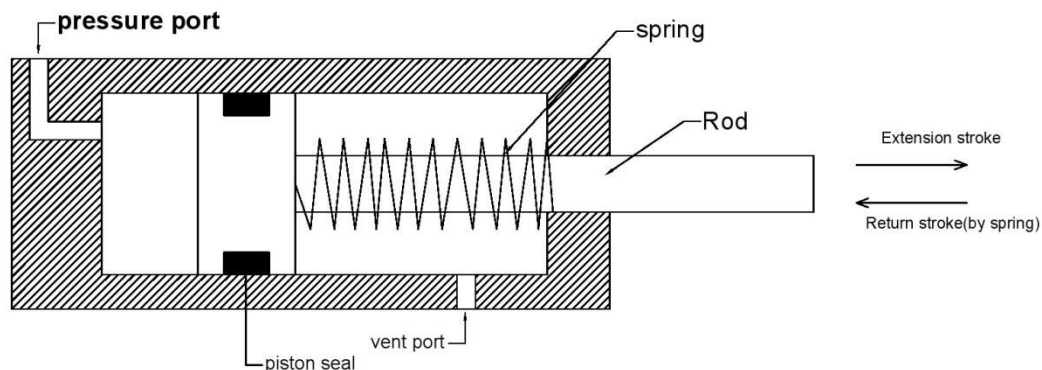


Figure 2. 1. Single action cylinder

Single action cylinder can take power-stroke only in single direction. That is either it can develop necessary force in forward stroke of cylinder or return stroke of cylinder, depending on its construction. The non-productive direction of cylinder stroke is achieved by various means such as self-weight (gravity), spring, auxiliary cylinder etc (Saleh, 2022).

Double Acting Cylinder: A cylinder that operates forward and backward strokes using fluid pressure. The most common double acting cylinder is a single rod end. This type of cylinder has pressure ports on both ends, providing power in both directions. Single rod end cylinders exert more force when extended than when retracted because the piston area at the blind end is larger than the piston area at the rod end due to the area covered by the piston rod (Saleh, 2022).

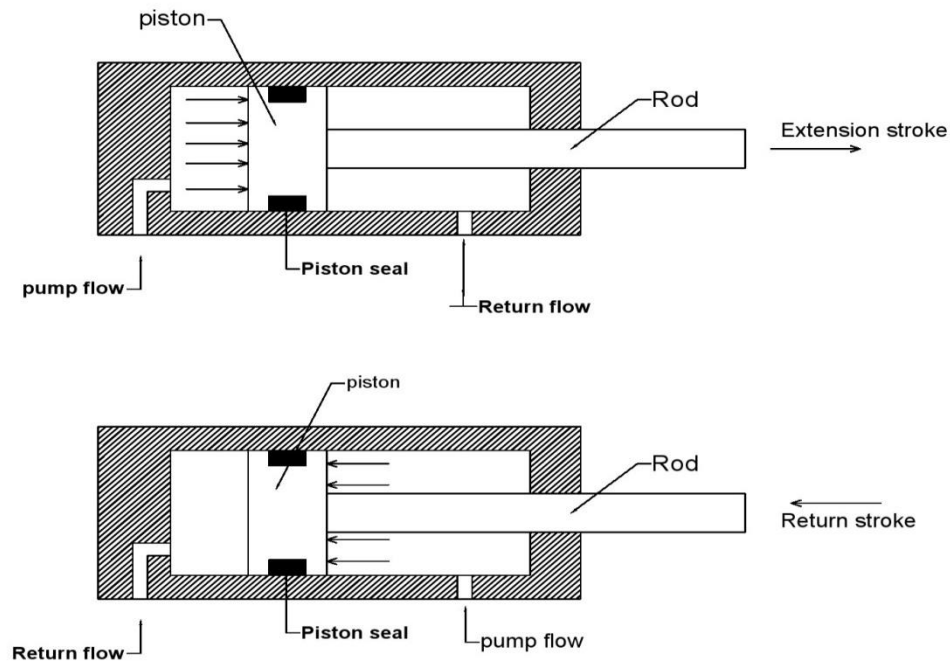


Figure 2. 2. Double action hydraulic cylinders

Double action cylinder can develop power-stroke in both forward and reverse direction.

2.1.2. Classification based on Body Construction of Hydraulic Cylinder

On construction basis hydraulic cylinders could be divided in to five categories.

1. Tie - Rod Construction.
2. Threaded Construction.
3. Bolted Construction.
4. One Piece welded construction.
5. Costume Build Cylinder with combination of above-mentioned constructions

Tie - Rod Construction

This type of construction is most widely used in industry. ISI standard also generally refers to one of this type of construction. As all the components are only machined and assembled

together and not welded. Hence planning manufacturing, quality control, assembly, and maintenance are more convenient than other types of construction. As long tie rods are used to hold the entire component together hence special care required to tighten them, and safe guard against loosening in operation. Like standard valves and pumps, these types of cylinders are also manufactured as standard hydraulic component, and used for low to medium pressure and low to medium duty operation for general purpose, and machine tool industry (Ankit H. Parmar, 2014).

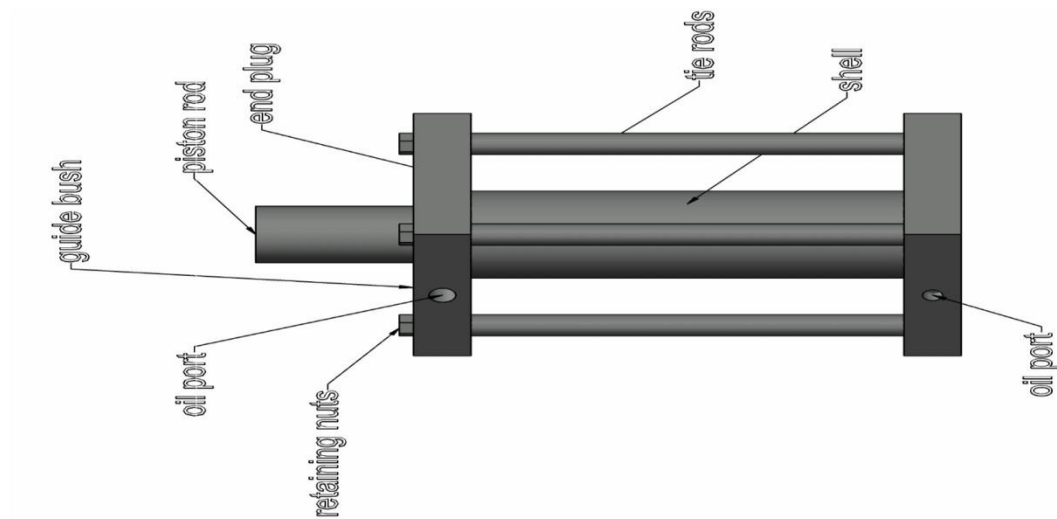


Figure 2. 3. Tie - Rod Construction

Threaded Construction

This construction is similar to tie-rod construction, but more compact, stronger, and requires more accuracy and care in manufacturing and quality control. In this design both ends are assembled with cylinder tube by threading, as shown in following design. These are used for medium to heavy-duty operation, and widely used in earth moving machinery (Ankit H. Parmar, 2014).

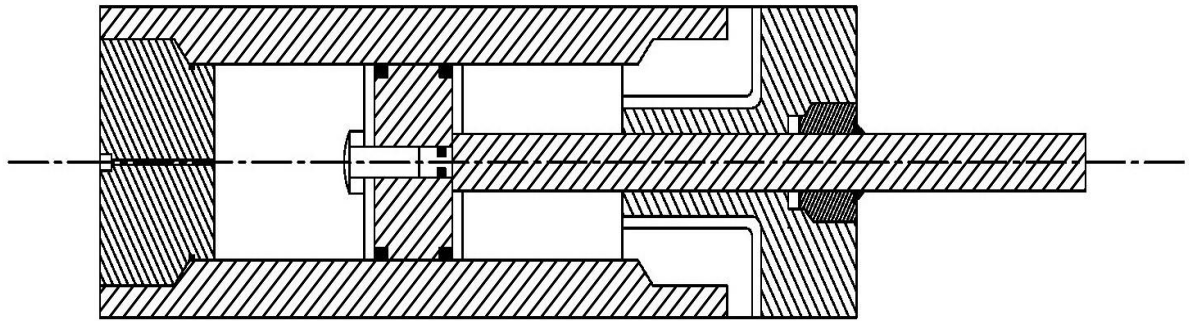


Figure 2. 4. Threaded Constructions

Bolted Construction

This type of construction involves welding of flanges to cylinder tube, and bolting of end cover to the welded flange. Similar to tie rod construction these are also designed and manufactured as standard hydraulic component and widely used in industry (Ankit H. Parmar, 2014).

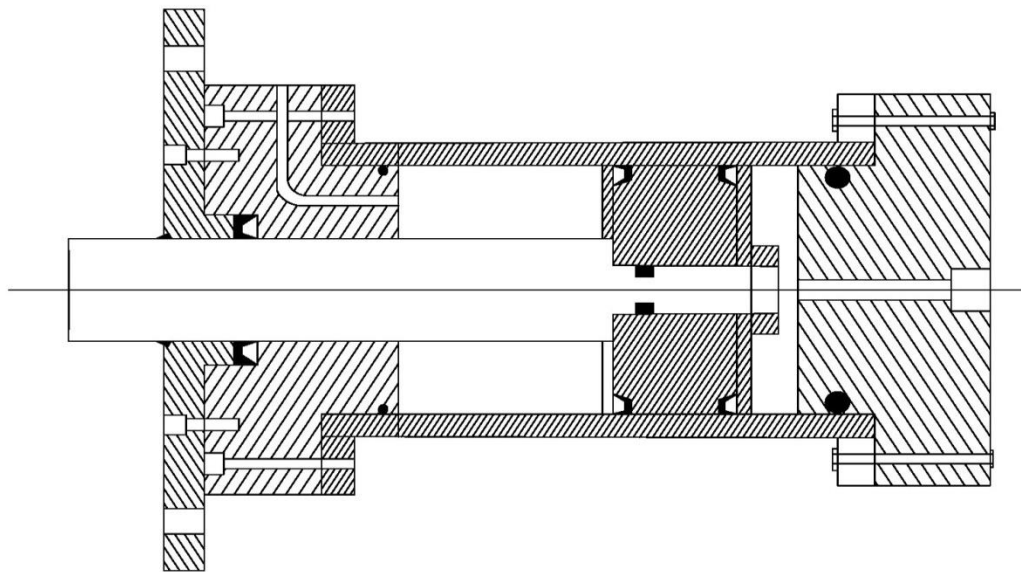


Figure 2. 5. Bolted Constructions

One Piece - Welded Cylinder

Similar to shock - absorber, in this design the end covers and cylinder tube are welded together. These are economical but cannot be repaired. They are used for low pressure; agriculture machinery application (Ankit H. Parmar, 2014).

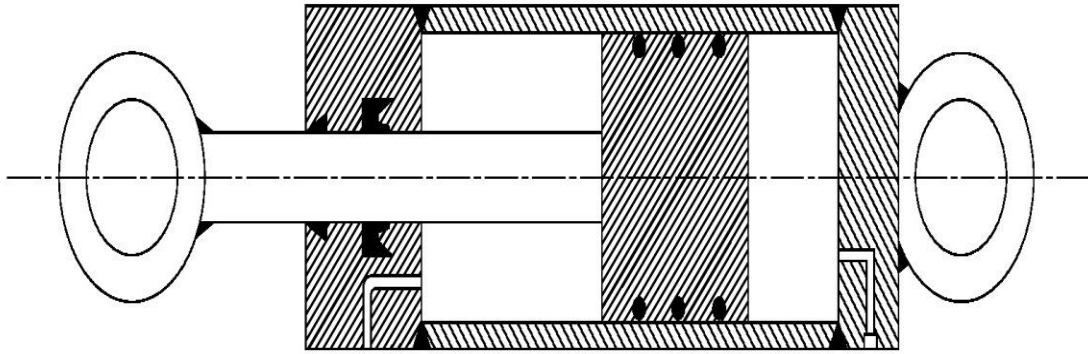


Figure 2. 6. One Piece – Welded cylinder

Custom - Build Cylinder

In this type of cylinder, various type of construction are mixed together to suit the requirement. One of the most widely used combinations is welded cap-end cover, bolted head-end cover, with front tube flange mounting. In case of high-capacity cylinder when it is steel cast or machined from solid steel forging, then end cover and front flange may be integral part of cylinder tube. Cylinders with these types of constructions are widely used in hydraulic press (Ankit H. Parmar, 2014).

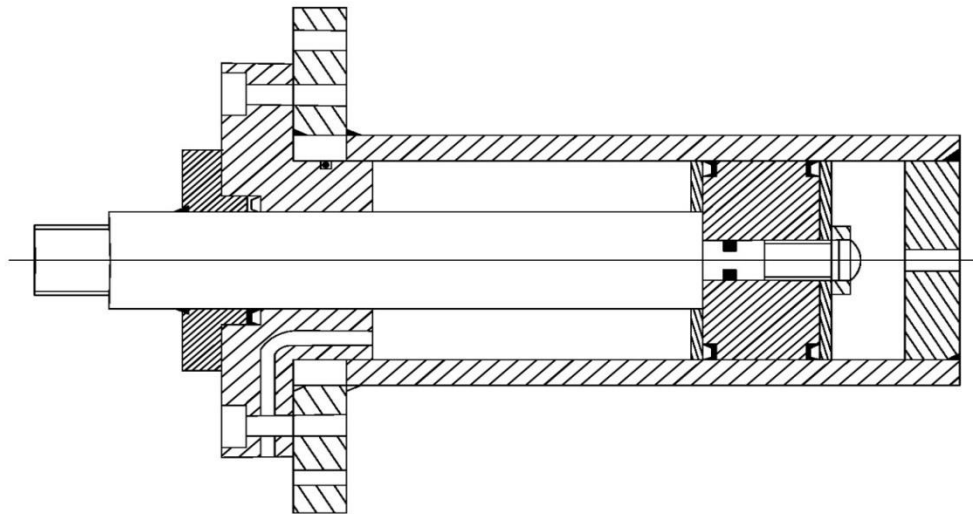


Figure 2. 7. Customs - Build Cylinder

2.1.3. General summary of literature review and research gap

Fatigue fracture mechanics concepts supported by finite element analysis were used to demonstrate the in-service failure could have been predicted. Imperfection angle and a wear on the guide ring reduce the cylinder load capacity for the tested actuator. Fatigue fracture

mechanics concepts supported by finite element analysis were used to demonstrate the in-service failure could have been predicted. From the literature review we conclude that EN8 has low values of deformation compared to conventional materials like stainless steel and other materials. Also, carbon fiber can be considered as it has the deformation value nearer to EN8 by changing the design and effective diameter of hydraulic cylinder we can improve the lifting load capacity and also increase hydraulic cylinder life. Finite Element analysis can be used as a tool to redesign the component if it is already designed by classical design theory. From the above literature review it is observed that most of the researchers use different types of finite element approach, others use analytical and experimental investigation for the analysis of hydraulic cylinder, and have done their research on other different vehicle like Sino truck and excavator rather than maxima cargo Bajaj and this will make this proposed work unique and also this proposed thesis uses both analytical and numerical approach throughout the process for comparing the results on the output. Literature survey about hydraulic piston is the key methodology of this thesis that will help to about what others done and identified, where the limitations does exist so that this work was based on the research gap.

CHAPTER THREE

MATERIALS AND METHODS

3.1. Introduction

The required materials and methodology to be followed are given in this chapter.

3.2. Materials

Materials required for execution of this thesis work are given here. Catia, CAD, and ANSYS software's expected to be used for designing, modelling and analysis are also given here.

3.3. Catia

CATIA software is a complete cross-platform developed as a CAD, CAM, CAE, PLM, and 3D solution. Software from France's Dassault Systems is ideal for creating solids, surfaces, assemblies, drawings, manufacturing, and analysis. Computer-aided design platform software suites are also used in a many of industries, aerospace to defines to packaging design (Aysha, 2020).

3.4. Methodology of the study

In order to address the general and specific objectives of this work a well-designed methodology were followed. Various steps included in the methodologies are listed below.

Some methods that were used in order to fulfil the basic objective of the hydraulic cylinder design analysis for lifting load are:

- ❖ Data collection of the maxima cargo Bajaj through surveying and documentation
- ❖ Identifying material for lifting load
- ❖ Mathematical modelling of the hydraulic cylinder for geometrical analysis
- ❖ Stress analysis of the model using ANSYS software under operating condition
- ❖ Identifying the equivalent design parameters and stress for the hydraulic cylinder under different loading conditions

The above methods are applied at a specified conditions and constraints with appropriate methodologies to achieve the most suitable and reliable result.

Stresses in Cylinder Shells subject to Internal Pressure: -

In thin shell, we could reasonably assume that the tensile stresses are uniformly distributed over the section of the wall.

When designing a thick cylinder, the following equation is usually used:

- i. Lamé equation: Assume equal stresses are applied to the longitudinal fibers of a cylindrical shell.
- ii. Barney's equation: Allowable stress theory of failure for open cylinders made of ductile material alloys (pump cylinders, tappets, gun barrels, etc.)
- iii. Clavarinos equation: Although the equation is based on the minimum strain theory of failure, applies to closed cylinders (or cylinders with heads) ductile materials made.
- iv. Barlow Equations: The equations are typically used for high pressure oil and gas pipelines

Design and analysis: under this we design and analysis different part of hydraulic cylinder

Part Modelling and Assembling: The 3D modelling of hydraulic cylinder and then assembling of maxima cargo Bajaj was done by CATIA V5

Validation: After modelling the modified parts then assemble those parts and finally, check for improvement.

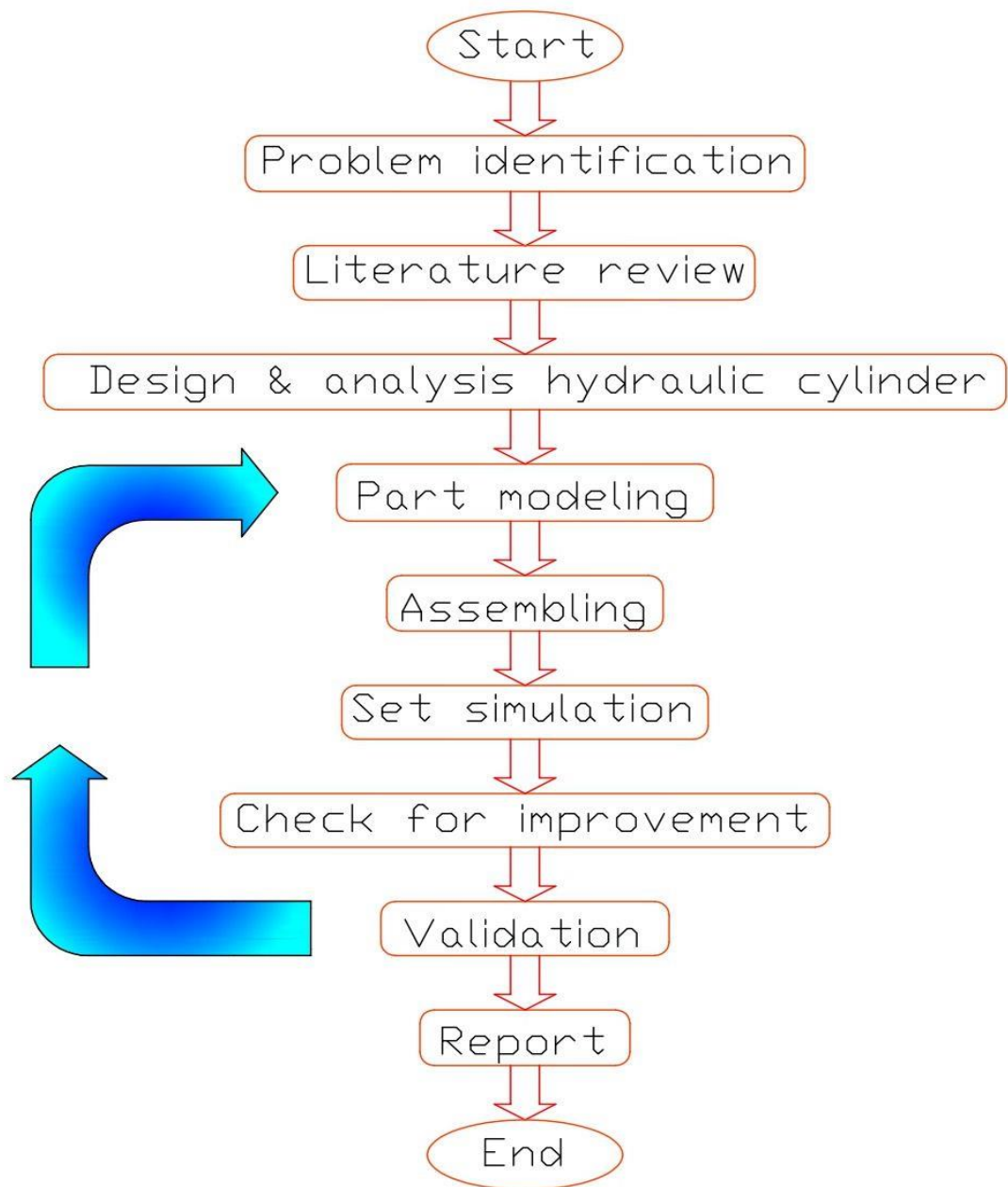


Figure 3. 1. Conceptual frame work

Table 3. 1. Maxima cargo Bajaj specification

Specification		
Seating Capacity		1 and driver
Max Power		6.61KW at3400rpm
Max Torque		2323@2000rpm
Kerb weight		518kg
Gearbox		4-speed
Ground Clearance		193mm
Over all Dimension (L,W,H)	Length	3230mm
	Width	1493mm
	Height	1818mm
Max speed		50 km/hr
Chassis Type		Monocoque
Suspension Type		



Figure 3. 2. Maxima cargo Bajaj (photo)

3.5. Hydraulics

A hydraulic system is a power transmission chain that converts mechanical energy into pressure and flow and return it to machine movement. Typically, the initial mechanical energy is rotary motion produced using ICE or an electric motor. Pressure and flow are transmitted hydraulically. The oil and final action can be either rotary or linear (Kauranne H. K., 2002).

There are two basic types of hydraulic systems: hydrodynamic systems and hydrostatic systems. The difference b/n the two is that in hydrostatic systems, energy is transferred and actuated by hydraulic pressure, whereas in hydrodynamic power transmission, it is transferred by 1 the kinetic energy of the hydraulic fluid.

3.5.1. Control of double acting hydraulic cylinder

The operation is described as follows:

- ❖ When the four-way valve is in its spring-cantered position (tandem design), the cylinder is hydraulically locked. Also, the pump is unloaded back to the tank at essentially atmospheric pressure.
- ❖ When the four-way valve is actuated into the flow path configuration of the left envelope, the cylinder is extended against its load force, Fluid, as oil flows from port P through port A. Also, oil in the rod end of the cylinder is free to flow back to the tank via the four-way valve from port B through port T.

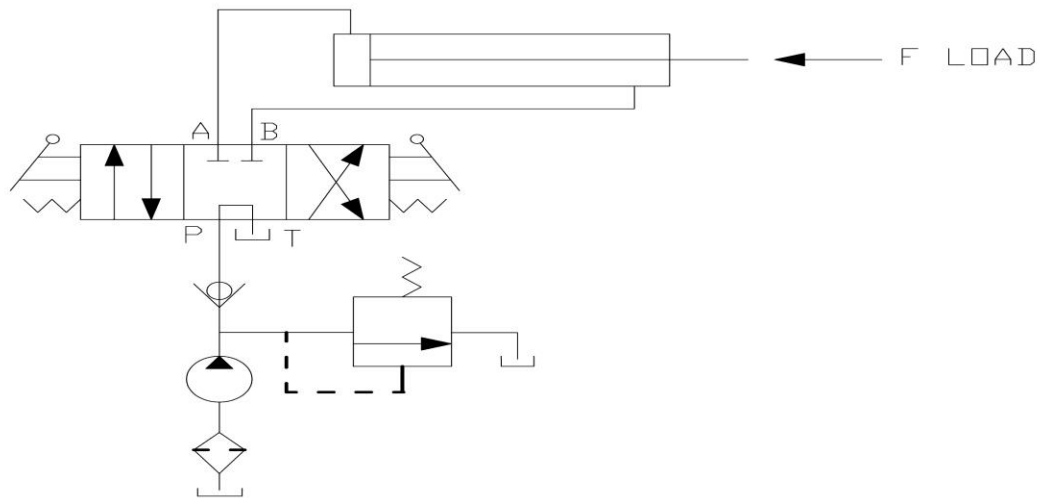


Figure 3. 3. Control of double acting hydraulic cylinder

- ❖ When the four-way valve is deactivated, the spring-cantered envelope prevails, and the cylinder is once again hydraulically locked.
- ❖ When the four-way valve is actuated into the right envelope configuration, the cylinder retracts as oil flows from port P through port B. Oil in the blank end is returned to the tank via the flow path from port A to port T.

- ❖ At the ends of the stroke, there is no system demand for oil. Thus, the pump flow goes through the relief valve at its pressure-level setting unless the four-way valve is deactivated. In any event, the system is protected from any cylinder overloads.
- ❖ The check valve prevents the load (if it becomes excessive) from retracting the cylinder while it is being extended using the left envelope flow path configuration.

3.5.2. Regenerative Cylinder Circuit

Used to accelerate the extension speed of double-acting hydraulic cylinders. Note that the piping to both ends of the hydraulic cylinder is connected in parallel, and one port of the four-way valve is blocked. The cylinder movement during the backward stroke is the same as a normal double-acting cylinder. Once retracted, fluid flows through the DCV through the right sleeve. In this manner, the fluid bypasses DCV and enters into the end of the cylinder rod from the pump. (Rola, 2014).

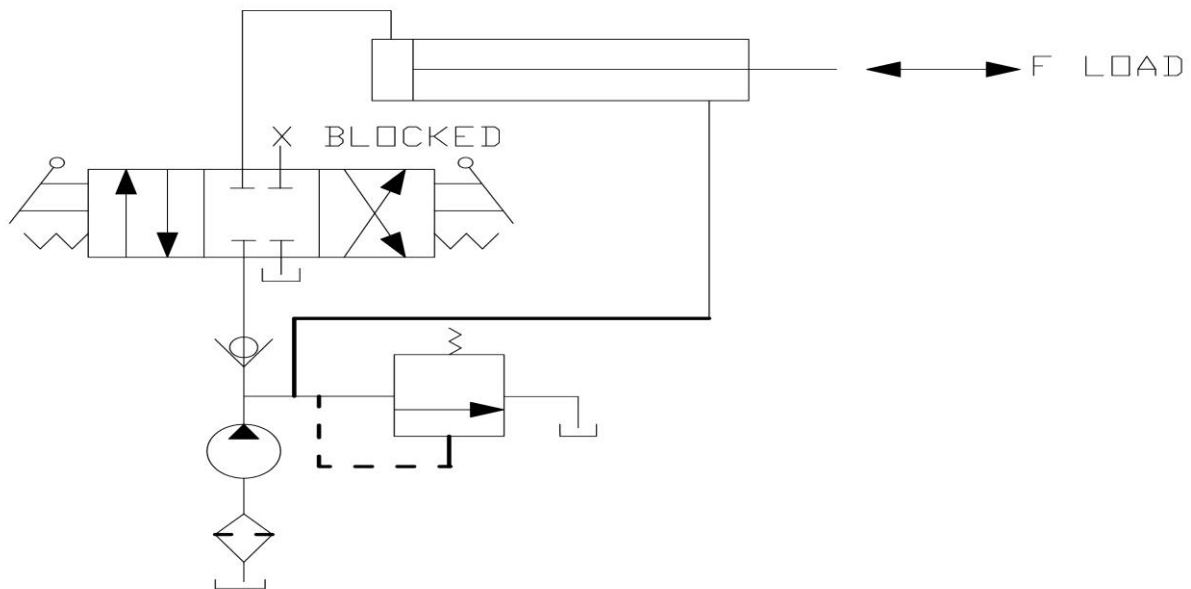


Figure 3. 4. Regenerative Cylinder Circuit

When the cylinder retracts, the fluid in the blind end returns into the tank through the DCV. As the DCV moves to the left envelope configuration, the cylinder extends. The (Q_R) is regenerated by (Q_p) , and (Q_r) to the empty end is more than the pump flow rate, so the expansion speed is faster than a normal double-acting cylinder (Rola, 2014).

Cylinder Extending Speed

The equation for the extending speed can be obtained as follows: The total flow rate entering the blank end of the cylinder equals the pump flow rate plus the regenerative flow rate coming from the rod end of the cylinder: $Q_T = Q_P + Q_R$ $Q_P = Q_T - Q_R$ We know that the total flow rate equals the piston area multiplied by the extending speed of the piston (V_{pext}). Similarly, the regenerative flow rate equals the difference of the piston and rod areas ($A_p - A_r$) multiplied by the extending speed of the piston. Substituting these two relationships into the preceding equation yields $Q_p = A_p V_{pext} - (A_p - A_r) V_{pext}$ Solving for the extending speed of the piston, we have:

$$V_{pext} = Q_p / A_r \dots \dots \dots (3.1)$$

Where: V_{pext} = extending speed of the piston
 A_r = retracting Area
 Q_p = flow pump

From the previous equation, we see that the extending speed equals the pump flow divided by the area of the rod. Thus, a small rod area (which produces a large regenerative flow) provides a large extending speed. In fact, the extending speed can be greater than the retracting speed if the rod area is made small enough.

Ratio of extending and retracting Speeds

Let's find the ratio of extending and retracting speeds to determine under what conditions the extending and retracting speeds are equal. We know that the retracting speed (V_{pret}) equals the pump flow divided by the difference of the piston and rod areas.

$$V_{pret} = Q_p / (A_p - A_r) \dots \dots \dots (3.2)$$

Dividing two equations, we have

$$V_{pext} / V_{pret} = A_p / A_r - 1 \dots \dots \dots (3.3)$$

Where: V_{pext} = extending speed of the piston
 V_{pret} = retracting speed, A_p = area of piston
 A_r = retracting Area, Q_p = flow pump
 Q_T = total flow rate

The standard double-acting cylinder with piston rod out one end is used in the majority of applications. It develops force in both directions of piston travel. The double-end-rod type is a

variation of the standard cylinder but having a piston rod extending out both end caps. It is occasionally used where it is necessary to have equal area on both sides of the piston, such as a steering application, and where one of rod extensions is to be used for mounting a cam for actuation of a limit switch, or for mounting a stroke limiting stop. The single-acting cylinder develops force in one direction, and is retracted by the reactive force from the load or an internal or external spring. The single-acting ram is a construction often used on fork lift mast raise, or a refuse body tailgate raise, or a high tonnage press cylinder. The telescoping cylinder is built in both single-acting and double-acting types. Its purpose is to provide a long.

Force Produced by a Cylinder: A standard double-acting cylinder has three significant internal areas. The full piston area when exposed to fluid pressure produces force to extend the piston rod. The amount of this force, in pounds, is calculated by multiplying piston square inch area times gauge pressure, in PSI (Rola, 2014).

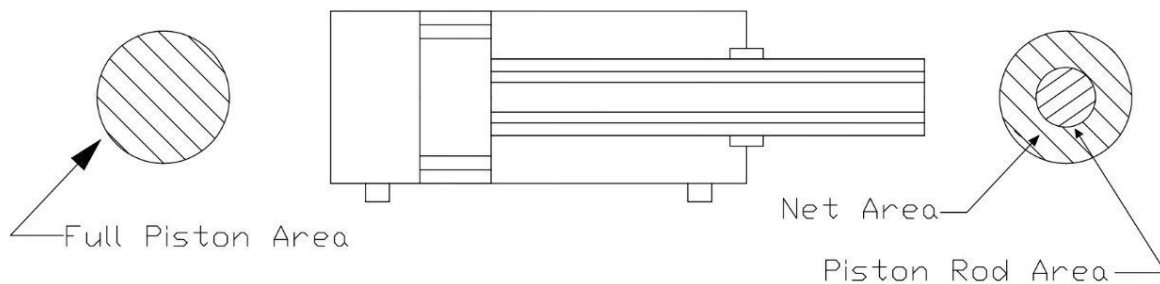


Figure 3. 5. Piston Cylinder

The “net” area on the front side of the piston is less than full piston area because part of the piston surface is covered by the rod. Net area is calculated by subtracting rod area from full piston area. Because net area is always less than piston area, cylinder force for rod retraction is always less than can be developed for extension when working at the same pressure.

Cylinder Force against a Load: The force which a cylinder can exert against a load is determined by making two calculations. First, extension force is calculated according to piston area and PSI pressure against it. Then, the opposing force on the opposite side of the piston is calculated the same way. Net force against a load is the difference between the two.

3.6. Hydraulic cylinder

In this thesis, a double acting hydraulic cylinder was used and not a hydraulic motor. For this reason, the focus of the theory is on the formulae required for designing systems of with hydraulic cylinders.

Hydraulic cylinders convert hydraulic power into linear mechanical movement. Depending on the cylinder construction, movement can be hydraulically powered in one direction and the return movement done with a spring or by the load. In this case the cylinder is called single acting cylinder. If the cylinder is hydraulically powered in both directions, it is a double acting cylinder (Figure 3.6).

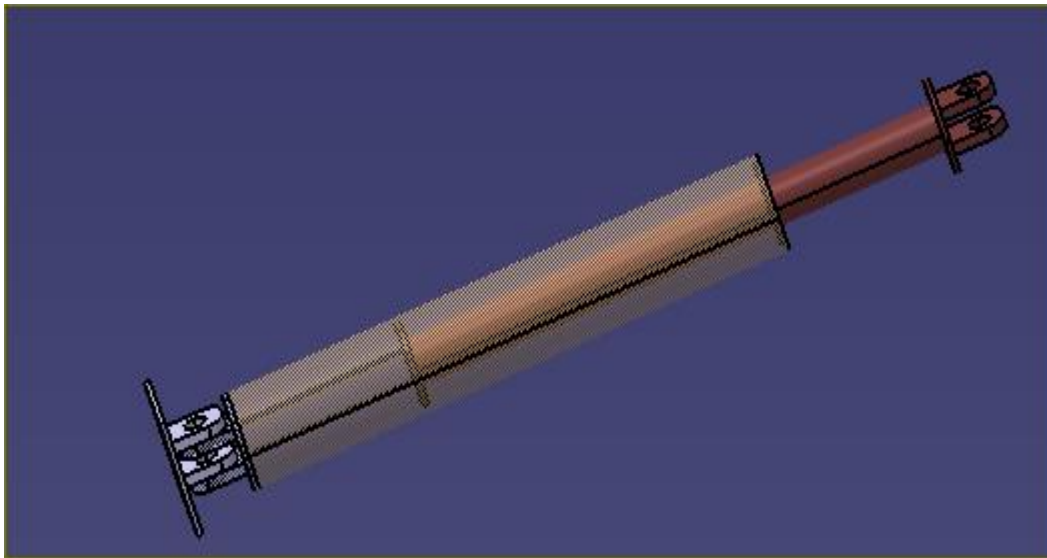


Figure 3. 6. Double acting cylinder

Cylinders which are built to be truly single acting are rare. Usually in their place is a cylinder with double acting construction but it is used as a single acting cylinder and the return movement is done with an external load. This allows the same cylinder to be used as a pulling or a pushing cylinder depending on how the hydraulic hoses are connected.

The movement speed of the cylinder depends on the volumetric flow and the effective piston area. Equation 1 shows the formula from calculating the movement speed of a hydraulic cylinder. The volumetric efficiency of the cylinder can lower the movement speed but in general the leaks are so small that calculations are done with η_{vol} .

$$v = Q\eta_{vol}/A \dots\dots\dots(3.4)$$

Where: v = movement speed (m/s)
 Q = volumetric flow (m³/s)
 η_{vol} = volumetric efficiency
 A = effective piston area (m²)

The maximum force generated by hydraulic cylinders is the product of the system pressure, the effective piston area, and the hydro mechanical efficiency as shown in Equation 3.5.

$$F = P \times A \times \eta_{hm} \dots \dots \dots (3.5)$$

Where: F = cylinder output force (N)
 P = pressure inside the cylinder (Pa)
 A = effective piston area (m²)
 η_{hm} = hydro mechanical efficiency

Double acting cylinders with a piston rod on one side have different movement speed and force in plus- and minus-directions due to the effective piston area being different. The piston rod creates an annular area which reduces effective piston area and results in less force but faster movement, in minus direction. When the piston area and annular area have a large ratio, the cylinders are called differential cylinders. In general, the ratio is 2, so that a differential coupling can be used to have same movement speed in both directions.

3.7. Hydraulic pumps

The most common way of producing hydrostatic power in industry is with a positive displacement pump. The basic working principle is the same in all the positive displacement pumps. They use a prime mover to power the pump which then moves the hydraulic fluid in the system. Movement of the hydraulic fluid is done by trapping the fluid in chambers which are open to intake and discharge lines alternately.

The pumps move the hydraulic oil in the system but they do not create pressure. The pressure is created only when the movement of the hydraulic fluid is resisted, for example by the load on a cylinder. The limit to the maximum pressure comes from the sealing of the system and from maximum torque driving the pump. One major component of the system sealing is the pump itself which acts as a seal between high pressure actuator side and the low-pressure suction side. When the maximum pressure goes over the limit of the sealing, oil leaks happen inside the

system or out of the system, and cause degraded performance. When the maximum pressure goes over the limit set by maximum output torque of the main mover, the pump stalls.

The most common pumps can be divided by their design into following categories: gear pumps, screw pumps, vane pumps and piston pumps. They all have the same basic principle outlined earlier but they have different characteristics. Some of the features that change are: fixed displacement and variable displacement, single and double direction flow, maximum pressure and efficiency.

When choosing a pump, the required volumetric flow and pressure need to be known. Based on these values the torque, rotation speed and power required of the prime mover can be calculated. The volumetric flow produced by pump depends on the rotation speed, displacement per revolution and volumetric efficiency.

$$Q = \eta_{vol} \times n \times V_k \dots \dots \dots (3.6)$$

Where: Q = Volumetric flow (m³/s)

V_k = displacement per revolution (m³/r)

n = rotation speed (r/s)

η_{vol} = volumetric efficiency

Torque required to drive the pump depends on the displacement, pressure and the hydro mechanical efficiency of the pump (Equation 3.7). For continuous use it is best to size the pump so that it can run at the pressure which gives the best total efficiency.

$$T = (\Delta p \times V_k) / (2 \times \pi \times \eta_{hm}) \dots \dots \dots (3.7)$$

Where: M = torque required from motor (Nm)

Δp = pressure difference over the pump (Pa)

V_k = displacement per revolution (m³/r)

η_{hm} = coefficient of hydro mechanical efficiency

Finally, the total power required from the motor has to be calculated (Equation 3.8). This is especially important when choosing an electric motor so that they do not overheat in continuous use.

$$P = Q \Delta P / \eta_{k \text{ ok}} \dots \dots \dots (3.8)$$

Where: P = power required from the motor (W)

Q = volumetric flow (m^3/s)

Δp = pressure difference over the pump (Pa)

η_{kok} = total efficiency of the pump

3.7.1. Importance of safe design

Hydraulic cylinder is actuated by hydraulic fluid at very high pressure. Any failure of cylinder may cause an explosion or ejection of a high velocity jet of oil, which may cause extensive damage to property and fatal injury to human being. Hence most precaution should be taken while designing a hydraulic cylinder.

3.7.2. Design of Cylinder Tube

In designing cylinder tube, the researcher basically calculates its inside diameter and wall thickness. Inside diameter and wall thickness depends upon capacity of cylinder and working pressure.

3.7.3. Capacity of Hydraulic Cylinder

Capacity of a hydraulic cylinder were decided for the purpose of which cylinder is going to be used. For example, if a cylinder is to be designed for deep drawing press to draw a particular utensil. Then the capacity of cylinder will depend upon the size of utensil and thickness and mechanical property of material to be drawn. Hence before designing a cylinder its capacity is determined and considering the operation it has to perform.

3.7.4. Theoretical Design of Main Shell OR Cylinder Tube

Cylinder tubes are divided in to two categories.

- 1) Thin Cylinder Tube
- 2) Thick Cylinder Tube

When ratio of cylinder bore to wall thickness is more than 10 then it is called thin cylinder. And when it is equal to or less than 10 then it is called thick cylinder. Equations to calculate wall thickness, and assumption made to used them are different,

3.7.5. Design of Thin Cylinder

The analysis of stress induced in thin cylinder is made on the following assumptions the effect of curvature of the cylinder wall is neglected. The tensile stresses are uniformly distributed over the section of the walls. The effect of the restraining action of the heads at the end of the cylinder tube is neglected. Whenever a cylinder is subjected to an internal pressure, it is likely to fail

either by splitting up into two cylinders (splitting circumferentially) or splitting it up into two troughs (splitting longitudinally). As shown in figures. No. Hence the wall of thin cylinder when subjected to internal pressure has to withstand following two types of tensile stress (Khan, Design and Manufacturing of Hydraulic Presses).

a) *Circumferential or hoop stress.*

b) *Longitudinal stress*

a) Circumferential or hoop stress: Consider thin cylinder subjected to internal pressure. Tensile stress acting in a direction tangential to the circumference is called circumferential stress or hoop stress. It is a tensile stress in longitudinal section. And expressed as

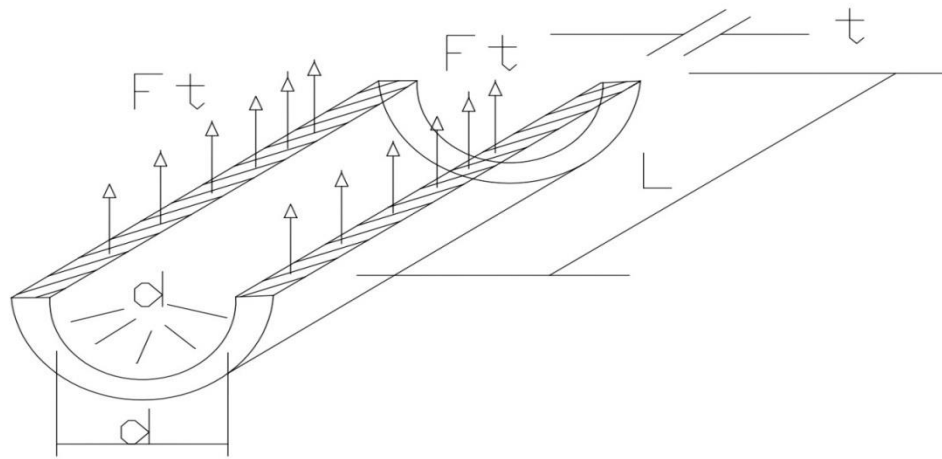


Figure 3. 7. Stress acting on direction of tangential circumference

Where: P = Intensity of Pressure (kg/cm^2)
 d = Inside diameter of cylinder (cm)
 t = Thickness of cylinder (cm)
 f_t = Circumferential stress or hoop stress in the cylinder (Kg/cm^2)

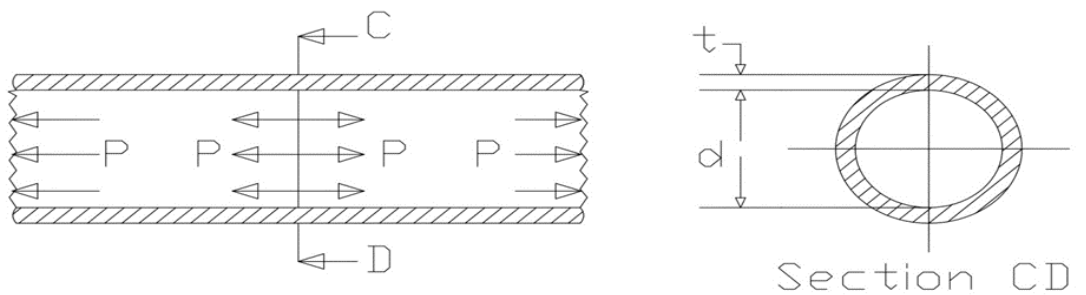


Figure 3. 8. Stress acting in longitudinal direction

In a closed vessel, tensile stress acting in the direction of the axis is the called longitudinal Stress. It is stress acting on the circumferential section CD as shown in following figure above.

3.7.6. Thin Spherical Shells

Sometime the end cover of casted cylinder is in spherical shape. When thin spherical shell subjected internal pressure tensile stress get developed in its wall, the thickness is calculated by following equation:

$$f_t = Pd/4t \dots \dots \dots (3.9)$$

Where:

P = Internal Pressure (kg/cm²)

d = Inside diameter of shell (cm)

t = Thickness of shell (cm)

f_t = permissible Tensile stress in shell material (kg/cm²)

3.7.7. Design of Thick Cylinder

In case of thin cylinder, stress assumed to be uniformly distributed over the section of wall, but in case of thick cylinder same assumption cannot be made. In case of thick cylinder stress distribution are as follow.

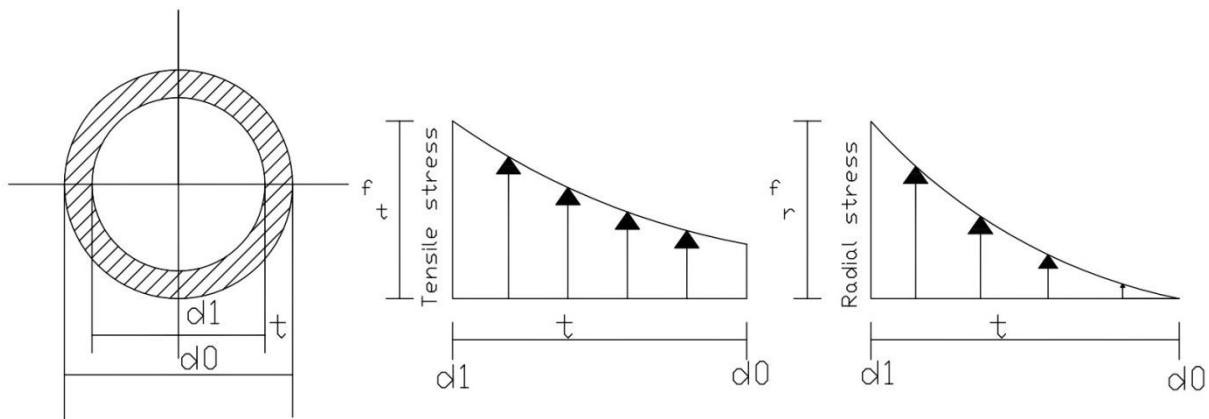


Figure 3. 9. Stress distribution in case of thick cylinder

Maximum radial stresses are generally equal to the internal pressure, and it is maximum at inner surface of cylinder.

$f_r (max) = -pr$ calculating tangential stress the following four equations are used.

Lame's equation:

$$f_t(\max) = p \frac{d_1^2 + d_2^2}{d_1^2 - d_2^2} \frac{1}{t} \frac{d}{2} \left(\sqrt{\frac{(f_t + p)}{(f_t - p)}} - 1 \right) \dots \dots \dots 3.10$$

where: t= thickness of metal in the shell

d= internal diameter

p= internal pressure

f_t= tensile stress of material

Lame's equation is used for designing cylinder of brittle material and it depends on maximum-stress theory of failure, and could be used for open as well as closed cylinder.

2) Brinie's equation:

Brinie's equation depends upon the maximum strain theory of failure. That is failure will occur when the strain reaches a limiting value. According to this theory the wall thickness of cylinder is

$$t = \frac{d}{2} \left(\sqrt{\frac{(f_t + (1 - \mu)p)}{(f_t + (1 + \mu)p)}} - 1 \right) \dots \dots \dots (3.11)$$

This equation is generally used for open-end cylinder made of ductile material, such as gun-barrels

3) Clavarino's equation:

This equation is similar to Birnie's equation, but applies to closed-end cylinder made of ductile material. According to this equation the thickness of a cylinder is.

$$t = \frac{pd}{2} \left(\sqrt{\frac{(f_t + (1 - \mu)p)}{(f_t + (1 + \mu)p)}} - 1 \right) \dots \dots \dots (3.12)$$

Where: p = internal pressure d= internal diameter

f_t = tensile stress of material t= thickness of metal

μ =Poisson's ratio

4) Barlow's equation

This equation is generally used for high pressure oil and gas pipes. According to this equation, the thickness of a cylinder is:

$$t = \frac{pd}{2f_t} \dots \dots \dots (3.13)$$

3.8. Parts design consideration

The following assumptions were taken into the consideration of the design of the cylinder, piston, piston rod and seals in the hydraulic cylinder.

- Working fluid is mineral oil.

- Available pressured $P_a = 25 \times 10^6 \text{ Pa}$
- Atmospheric pressure $= 1.0135 \times 10^5 \text{ Pa}$
- Stroke length $= 0.8 \text{ m}$
- Cylinder output force $= 10 \text{ kN} = 10000 \text{ N}$
- Material for cylinder calculation unalloyed medium carbon steel EN8 (080M40)
- Tensile stress of material is $= 550 \text{ MPa}$
- Yield stress of material is $= 280 \text{ MPa}$
- Factor of safety $= 3$
- Young modulus of the material used is $= 190 \text{ GPa}$, for BS 970 080M40 (unalloyed medium carbon steel)
- End fixing factor $= K = 0.7$ is chosen because of maintenance purpose, in a case of adjustment i.e., in the case of increasing stroke length.

3.8.1. Material for the cylinder

Unalloyed medium carbon steel EN8 (080M40) chosen as material for the cylinder because it combines the required lightness with very high yield strength, good corrosion resistance and good surface hardness for wear resistance. It has minimum yield strength of 280 MPa, minimum tensile strength 550 MPa and minimum surface hardness of 152 HB.

3.8.2. Design of piston rod

The piston rod of a hydraulic cylinder is highly stressed, and therefore it should be able to resist the bending, tensile and compressive forces that it may encounter during the operation without buckling. In practice, the rod is more likely to fail by buckling under the compressive load than by bending. In this case, the rod behaves like a column and is subjected to buckling. The rod diameter can be related to critical load. Therefore, Euler's formula in the equation below for long column can be used to obtain the piston rod diameter.

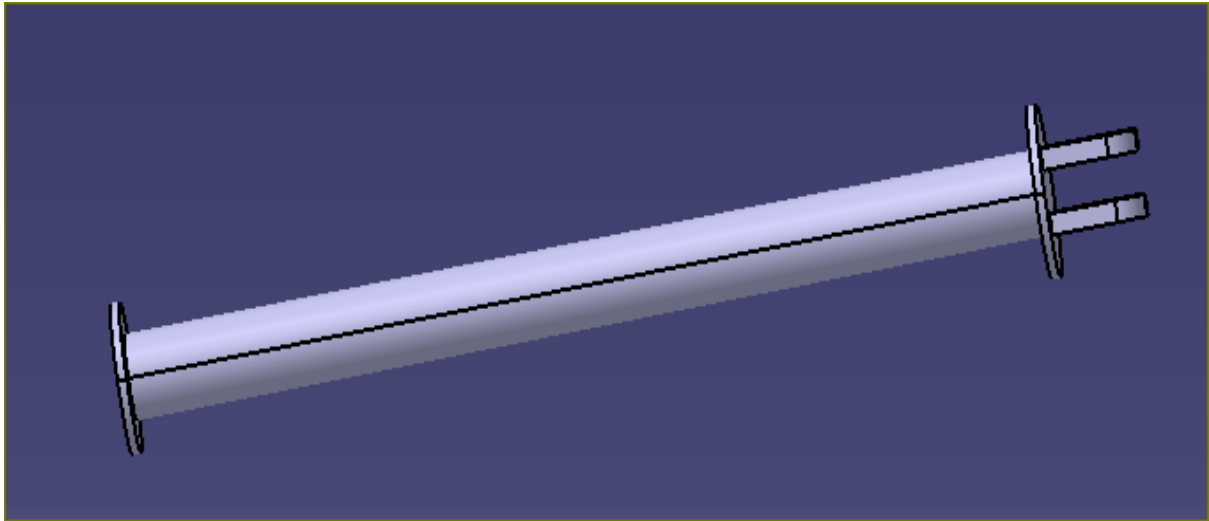


Figure 3. 10. Piston rod

$$P = (\pi^2 * E * I) / (L^2 * K^2) \dots\dots\dots(3.14)$$

Where: P = Buckling load (N)

L = the column length (m)

I = Moment of inertia (m⁴)

E = Young's Modulus of Elasticity for the column material (Pa)

K = the end fixing factor

E = Young's modulus of the material used in this design calculation is 190 (GPa) for BS 970 080M40 (unalloyed medium carbon steel).

P = cylinder force x factor of safety = 10000 'x' 3 = 30 'x' 10³N

L = total stroke length = 0.8m

K = 0.7, Reason for choosing k = 0.7 is for maintenance purposes in case of adjustment i.e., in case of increase in the stroke length of the rod.

$$30 \times 10^3 = (\pi^2 \times 190 \times 10^9 \times I) / ((0.8)^2 \times (0.7)^2)$$

$$30 \times 10^3 = (1.8732 \times 10^{11} \times I) / (3.136 \times 10^{-1})$$

$$1.8752 \times 10^{11} \times I = 9408$$

$$I = 9408 / (1.8732 \times 10^{11})$$

$$I = 5.02 \times 10^{-9} \text{m}^4$$

The moment of inertia and the maximum permissible stress to avoid buckling is dependent on the type of end fixing of the cylinder. The moment of inertia (I) can be found from the formula below

$$I = (\pi d^4)/64 \dots \dots \dots (3.15)$$

$$[5.02 \times 10]^{-9} = \pi d^4/64$$

$$d^4 = [32.12 \times 10]^{-8}/\pi$$

$$d^4 = 10.23 \times 10^{-8}$$

$$d = \sqrt[4]{[10.23 \times 10]^{-8}}$$

$$d = 17.8 \text{ mm}$$

Hence diameter of the piston rod required, $d = 17.8 \text{ mm}$. Finally, for other calculations and in a case of construction, the diameter of the piston rod used will be 20 mm , for safety and because from Baym Hydraulics Corporation catalogue of metric rod wipers and piston seals the nearest standard rod seal diameter is 20 mm .

Chromium Plating on Piston Rod

A piston rod must have a high tensile strength, it must have a hard surface, it must be corrosion resistant and it must be smooth. It must resist wear from the side load forces that a hydraulic cylinder will see in service. It must provide a low friction surface as it moves back and forth in the hydraulic cylinder passing over seals, wipers and bearing surfaces.

Thickness of chromium plating = $50\text{-}80 \text{ micron}$

Force on both side of piston:

Tensile strength of EN8 = 550 mpa

Dia of piston rod = 20 mm

$$F_1 = P \times A_1 \dots \dots \dots (3.16)$$

$$F_1 = ([25 \times 10]^6 \times \pi \times 52^2)/[4 \times 10]^6 = 53.1 \text{ KN}$$

$$F_2 = P \times A_2$$

$$F_2 = ([25 \times 10]^6 \times \pi \times (52 - 20)^2)/[4 \times 10]^6 = 20.1 \text{ KN}$$

3.8.3. Design of the piston

Piston is a sliding part of cylinder, which is attached to piston rod. It accommodates hydraulic seals to avoid leakage between cylinder and piston. It is also fitted with guide-ring, wear rings, or piston itself may be made from bearing material to protect the honed inside surface of cylinder. Piston withstands the full load developed by pressurized oil and transfers it to the piston rod. It is the first member which bears and transfers the load; hence it should be designed accordingly for its satisfactory performance

According to the attachment with piston rod, piston can be classified in two categories.

- a) Piston as integral part of piston rod.
- b) Piston simply supported by piston rod. Integral type of piston rod

Integral type of piston rod:

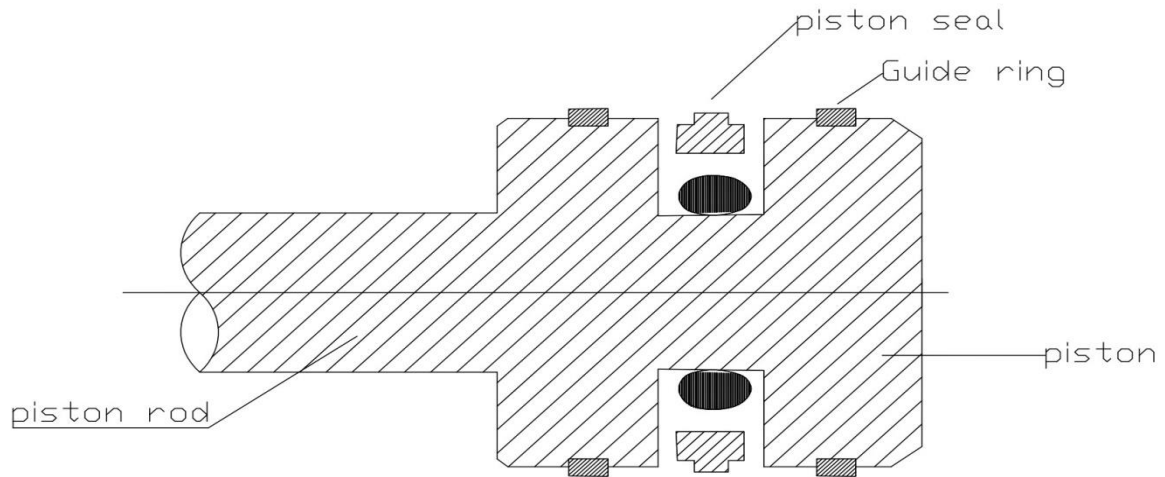


Figure 3. 11. Piston as integral part of Piston-rod

Integral type of piston and piston rod arrangement is generally used for large size of piston rod, and high-capacity cylinder. And in any cylinder this arrangement will be more economical where length of piston rod is short and diameter of piston rod is very close to inside diameter of cylinder. This arrangement was sturdier than supported type.

Perfect alignment is also always remaining between piston, piston rod, cylinder tube, and guide-bush, hence more life to seal and less wear and tear to guides.

Simply supported type of piston or assembled type of piston: This type piston is attached to piston rod by means of threads or simply assembled on end of piston rod and retained by check-nut. This type of assembly is most-widely used in industry, as it is convenient for mass-production of piston of standard dimension. Piston rod is also mass produce as straight ground and hard chrome plated rods. As Inside Diameter of cylinder, Outside Diameter of piston rod and seals etc. are all as per international standard, so whenever a cylinder is to be made then tube and piston rod are cut as per required stroke, and piston and other component which are same for a particular Inside diameter of cylinder are assemble together (Khan, Design and Manufacturing of Hydraulic Presses).

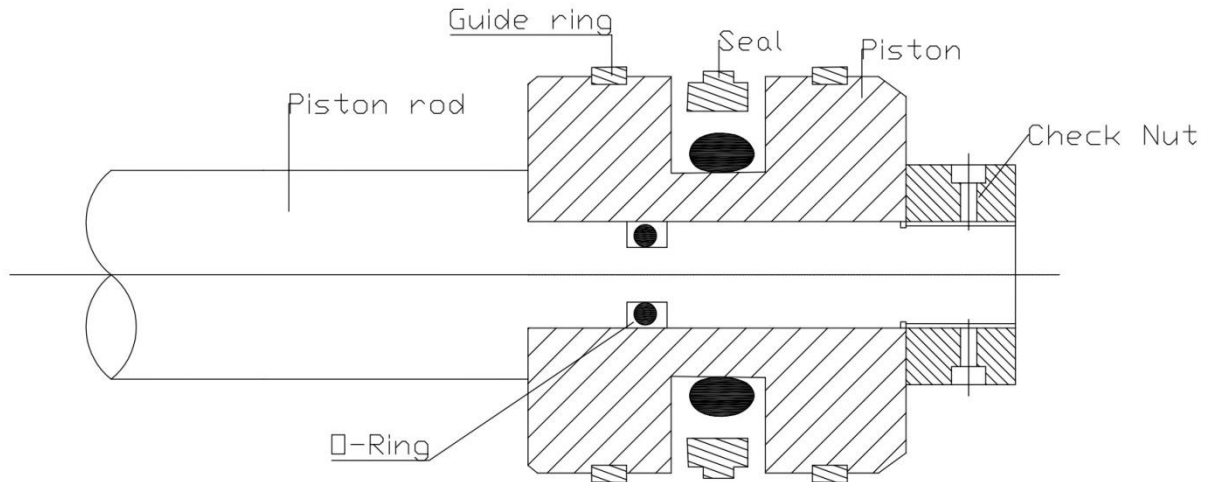


Figure 3. 12. Piston assembled on Piston Rod

At 25 MPa the piston diameter required for creating the 10000N piston force is calculated with
 $A_p = F / (P \times \eta_{hm})$(3.17)

Where: F = cylinder output force

N = assume force $\times$ factor of safety (3) = $10000 \times 3 = 30\,000\text{N}$

P = pressure inside the cylinder (Pa) 25×10^6

A_p = effective piston area (m^2)

η_{hm} = hydro mechanical efficiency = 0.9

$$A = \frac{[30 \times 10]^3}{[0.9 \times 25 \times 10]^6}$$

$$A_p = 1.333 \times 10^{-3} \text{ m}^2$$

Now piston diameter is calculated from the basic equation for area of a circle

$$A_p = \pi ((D_p)^2) / 4$$
.....(3.18)

Where: A_p = piston area (m^2)

D_p = piston diameter (m),

$$D_p = \sqrt{4A_p / \pi}$$

$$D_p = \sqrt{(4 * 1.333 \times 10^{-3}) / \pi}$$

$$D_p = 41.2 \text{ mm}$$

Result for minimum area of $1.333 \times 10^{-3} \text{ m}^2$ corresponds to piston diameter of 41.2 mm. This was rounded up to 45 mm and it was accepted since standard seals could be found in suppliers' catalogue for this size.

When the piston diameter is 45 mm and piston rod diameter is 20 mm, the piston annular area is calculated with

$$A_{\text{annular}} = (\pi(d_{\text{piston}}/2)^2) - (\pi(d_{\text{pistonrod}}/2)^2) \dots\dots\dots(3.19)$$

$$A_{\text{annular}} = (\pi(0.045/2)^2) - (\pi(0.02/2)^2)$$

Where: A_{annular} = piston annular area (m^2)

d_{piston} = piston diameter (m) 0,045 m

$d_{\text{pistonrod}}$ = piston rod diameter (m) 0,02 m

Result for annular area is = $12.5 \times 10^{-4} \text{mm}^2$

3.8.4. Design of the cylinder

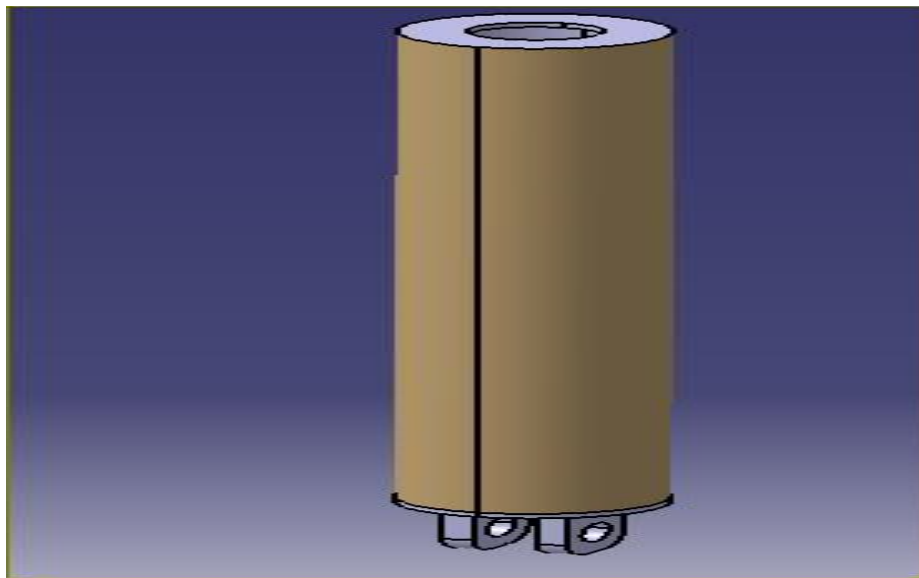


Figure 3. 13. Cylinders

Let OD = outside diameter of the cylinder

Tensile stress of BS 970 080M40 = 550MPa

Factor of safety (N) = 3

Determine the maximum working stress (σ_m) is given as

$$(\sigma_m) = \text{tensile stress of material} / \text{FOS} = \sigma_t / N$$

$$\Sigma m = \sigma_t / (N) \dots\dots\dots(3.20)$$

$$\begin{aligned} \Sigma m &= (550\text{MPa}) / (3) \\ &= 183.3 \times 10^6 \text{ pa} \end{aligned}$$

Applying lame's equation to determine OD

$$OD^2 = D^2 ((\sigma + P))/((\sigma - P)) \dots\dots\dots(3.21)$$

Where: σ_m = working stress

p = pressure

$$OD^2 = \frac{45^2 (183.3 \times 10^6 + 25 \times 10^6)}{(183.3 \times 10^6 - 25 \times 10^6)}$$

$$OD = 51.6 = 52\text{mm}$$

3.8.5. Cylinder Tube thicknesses

The tube thickness of a cylinder barrel is a very important factor in the design of a hydraulic cylinder. The strength of the cylinder tube is proportional to its wall. If a cylinder is too thick or too thin may pose serious safety and operational problems and hence the tube thickness of the cylinder has to be carefully chosen. The wall thickness required for the cylinder can be calculated from the formula in equation

$$T = (OD - d)/2 \dots\dots\dots(3.22)$$

Where: OD = the cylinder external diameter, 52mm and small

d = the piston seal diameter (cylinder internal diameter), 45mm.

$$t = (52 - 45)/2$$

$$\text{Thickness } (t) = 3.5\text{mm}$$

3.8.6. Bursting stress

In this design calculation, the hoop stress must be lower than a tensile strength of the material which the cylinder is made from to ensure the safety of the cylinder and personnel during actual operation. Material of the cylinder BS 970 080M40 (unalloyed medium carbon steel) Tensile stress = 550×10^6 pa. The hoop stress (σ_H) of a cylinder can be determined from the Barlow formula as shown in the equation below

$$\sigma_h = P \frac{Do^2 + Di^2}{Do^2 - Di^2} \dots\dots\dots(3.23)$$

Where: p = oil pressure, = 25×10^6 Pa

do = outer diameter of cylinder, 52mm

di = inner diameter of cylinder, 45mm

$$\sigma_h = \frac{25 \times 10^6 (52^2 + 45^2)}{(52^2 - 45^2)}$$

$$\sigma_h = 174.1 \text{ MPa}$$

Also, the longitudinal stress given by

$$\Sigma l = \frac{P_1 R_1^2 - P_2 R_2^2}{R_1^2 - R_2^2} \dots \dots \dots (3.24)$$

Where: P_1 = Internal pressure (25×10^6)

P_2 = External pressure (atmospheric pressure = 1.0135×10^5 Pa)

R_1 = Internal radius

R_2 = External radius

$$\Sigma l = \frac{25 \times 10^6 \times 22.5^2 - 1.0135 \times 10^5 \times 26^2}{26^2 - 22.5^2}$$

$$\sigma l = 74.15 \text{ MPa}$$

The pump chosen is PARKER H B S-519 miniature piston pump (Appendix 2). It has the maximum intermittent pressure of 27,6 MPa and displacement of $0.519 \text{ cm}^3/\text{r}$. The maximum rotation speed after gear is 4000 rpm and torque 2.6 Nm. Volumetric flow can be calculated with equation...

$$Q = \eta_{\text{vol}} \times n \times V_k \dots \dots \dots (3.25)$$

Where: Q = volumetric flow (m^3/s)

V_k = displacement per revolution (m^3/r), $5.19 \times 10^{-7} \text{ m}^3/\text{r}$

n = rotation speed (r/s), 66.67 r/s

η_{vol} = volumetric efficiency, 0.75

$$Q = 0.93 \times 30 \text{ r/s} \times 2.5 \times 10^{-6} \text{ cm}^3/\text{r}.$$

$$2.6 \times 10^{-5} \text{ m}^3/\text{s}$$

At 25 MPa pressure the pump gives volumetric flow of $2.6 \cdot 10^{-5} \text{ m}^3/\text{s}$. This value is used to calculate the plus movement. For minus movement there is no back pressure and so the volumetric efficiency goes up to 0.93. For minus movement the volumetric flow is $3.2 \cdot 10^{-5} \text{ m}^3/\text{s}$. The torque required to run the pump at maximum pressure is calculated with

$$T = (\Delta p \times V_k) / (2 \times \pi \times \eta_{\text{hm}}) \dots \dots \dots (3.26)$$

Where: T = torque required from motor (Nm)

Δp = pressure difference over the pump (Pa), $25 \cdot 10^6$ Pa

V_k = displacement per revolution (m^3/r), $5.19 \cdot 10^{-7} \text{ m}^3/\text{r}$

η_{hm} = coefficient of hydromechanical efficiency, 0.865

$$T = ((25 \times 10)^6 \times (5.19 \times 10)^{-7}) / (2 \times \pi \times 0.865)$$

$$T = 2.4 \text{ Nm}$$

The torque required at maximum pressure is 2.4 Nm. This value is under the maximum allowed torque of 2.6Nm and so this pump can be used.

The next step is to examine if this pump and cylinder combination fulfils the cycle speed requirement. The piston movement length used to calculate it is 0.8m. Equations 3.27 and 3.28 are used to calculate the theoretical movement speed of the piston. Equation 3.27 and 3.28 are derived from the equation for cylinder movement speed (Equation 3.4).

$$t_{\text{plus}} = S/V \dots\dots\dots 3.27$$

Where:

V= speed of piston movement (m/s)

S = movement length (m),

Q25 = volumetric flow of the pump at 25 MPa, $2.6 \times 10^{-5} \text{ m}^3/\text{s}$

$$V = Q/A \dots\dots\dots 3.28$$

$$V = \frac{[2.6 \times 10]^{-5}}{[12.7 \times 10]^{-4}}$$

$$V = 0.2047 \text{ m/s}$$

So, time plus movement will be

$$T_{\text{plus}} = 0.8/0.2047$$

$$t_{\text{plus}} = 3.9 \text{ s}$$

$$t_{\text{minus}} = (S \times \text{annulus}) / (Q \eta_{\text{volcyl}}) \dots\dots\dots 3.29$$

Where: t_{minus} = time for minus movement (s)

s = movement length (m), 0.8

Aannulus = piston annulus area (m^2) $12.7 \times 10^{-4} \text{ m}^2$

η_{volcyl} = cylinder volumetric efficiency, 0.93

Q0 = volumetric flow of the pump with no back pressure, $3.2 \times 10^{-5} \text{ m}^3/\text{s}$

$$t_{\text{minus}} = (0.8 \times [12.7 \times 10]^{-4}) / (0.93 \times [3.2 \times 10]^{-5})$$

$$t_{\text{min}} = 0.604 \text{ s}$$

With these values the plus movement time is 3.9 seconds and a minus movement time 0.604 seconds. This makes the total cycle time approximately 4.504 seconds and is better for the system that we require.

3.8.7. End-plug

End-plug It also is called as "Cap-End" "cover-End" or rear-end of cylinder. This is a cylinder end enclosure, which completely covers the cylinder-bore area. In addition, it could be used for

mounting cylinder, providing oil port and air-bleed-off-port, accommodating cushioning arrangement etc.

Design of end-plug

$$T = pd/4ft \dots \dots \dots (3.30)$$

Cylinder end-plug may be, threaded, bolted, welded or integral part of cylinder shell. Spherical type end-plug is only made in case of steel-casted cylinder. When end plug is flats in shape, then its minimum thickness can be calculated by following empirical formula

In case of flat-end-plug thickness is calculated as

$$T = di(\sqrt{(kp/ft)}) \dots \dots \dots (3.31)$$

So, the thickness of the end plug is

$$t = 45(\sqrt{((0.162 \times [25 \times 10]^6)/[550 \times 10]^6)})$$

$$t = 45(\sqrt{(4.05/550)})$$

$$45\sqrt{0.00736}$$

$$t = 5\text{mm}$$

Where: di = Inside diameter of cylinder (mm)

k = an empirical coefficient and equal to 0.162

P = Pressure inside cylinder (kg/cm²)

Ft = Permissible tensile stress for the material of the plate (kg/cm²)

Important guide-lines:

- ✓ In general, most of the structural failure of cylinder occurs at end-plug because.
- Cylinder expands under pressure.
- Cylinder expanders at high temperature
- Micro cracks present in welding propagate and get fully developed in long period. · Because of loose thread fitting between cylinder and plug, end-plug tend to move with every pressure cycle, in long run this leads to cracking of welding.

3.8.8. Flanges bolted to cylinder tube

The purpose of the front end is to seal the hydraulic cylinder and to hold tool in place a pin place through the holes in the end of the part. The end with pin holes has more wall thickness to

withstand the force. The wall thicknesses were optimized with Catia simulations with minimum safety factor of 3.

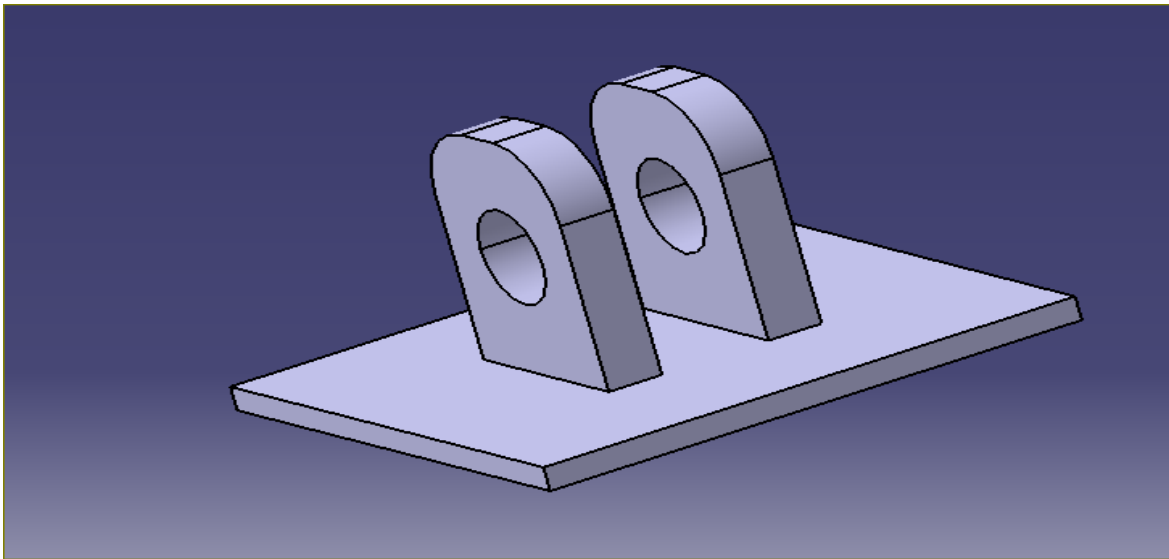


Figure 3. 14. Rectangular flanges

In normal use the front-end flange and piston should never come into contact. The cylinder tube and tool mechanism have been designed so that when the mechanism closes there is still a 3mm gap between the piston and the front-end flange. This is done so that the maximum stress is moved outside the cylinder into the main mechanism which is stronger. Front end flange is strong enough to withstand the maximum 10kN force but it only has a safety factor 3. This way the front end can handle occasional contact between it and piston but not for each cycle during the life time.

These are circular or rectangular rings threaded and welded to the outside diameter of cylinder tube. When this is fixed at front-end of cylinder then it is called front-tube flange and may be used for bolting of guide-bush, and cylinder mounting. When it is fixed to the rear-end of cylinder then it is called rear-tube-flange of cylinder and it may be used for bolting of end-plug and also cylinder mounting.

Flange is subjected to shear and bending load. When mounting support of cylinder is too close to cylinder-tube outside diameter then bending load is minimum, and as gap increase bending load increases.

3.9. Body modification

Today there is a demand on trucks not only on the cost and weight aspect but also on the improved and complete vehicle feature and the overall performance. In addition to this number of variants that are possible due to different types of design and modernization call for several design iterations to arrive is suitable combination. The project work deals with Maxima Bajaj body.

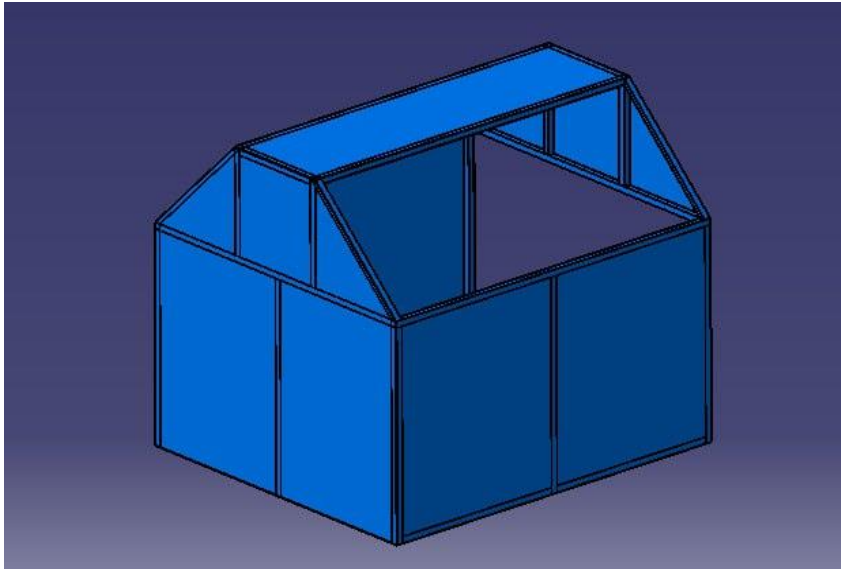


Figure 3. 15. Body modification

Loading

- The maxima cargo Bajaj body is loaded with a static force from the material it carries
- The load carrying capacity of the Bajaj is 520kg.
- Thus, the exerted force is equal to weight times gravity ($520\text{kg} \times 9.81\text{m/s}^2 = 5101.2\text{N}$)

Assuming that, 100% of force acts at the bottom, 15% on the side, and 20% on the front.

The force acting at the bottom single nodes: -

$$520 * 9.81 = 5101.2\text{N}$$

Force acting on a single node: -

$$520 * 9.81 * 15/100 = 765.18\text{N}$$

The force acting on front at single node: -

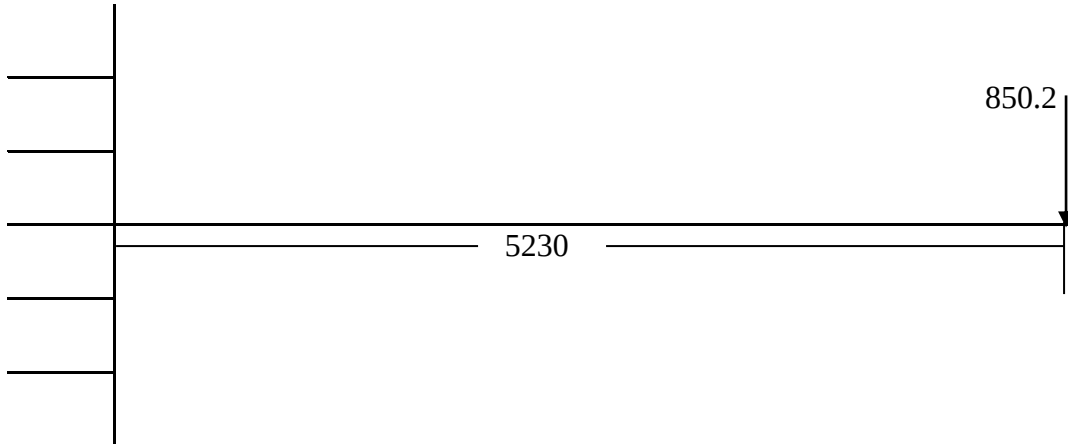
$$520 * 9.81 * 20/100 = 1020.2\text{N}$$

CASE 1: - Since there are 6 member and force acting on a single member

$$5101.2/6 = 850.2N$$

Allowable stress = (tensile strength of material)/FoS

$$\frac{758}{3} = 252\text{mpa}$$



$$\sum fy = 0$$

$$R_o = 850.2 = 0$$

$$R_a = 850.2N$$

$$\sum M_{a=0}$$

$$M_-(a = 850.2 * 3230)$$

$$2,746,146\text{Nmm}$$

By bending flexural formula

$$\frac{M}{I} = \frac{\sigma}{Y}$$

Inertia for rectangular shape:

$$I = \frac{bh^3}{12} = \frac{1493 * (1500)^3}{12}$$

$$= 50388750 \text{ mm}^4$$

$$Y = \frac{1.5m}{2} = 0.75m$$

$$\sigma = \frac{my}{I}$$

$$\sigma = \frac{(2746146 * 750)}{50688}$$

$$40.87\text{N/mm}^2$$

CHAPTER FOUR

RESULT AND DISCUSSION

4.1. Introduction

In the previous chapter, steps and input parameters that are used in the analysis of the hydraulic cylinder are discussed briefly. In this chapter, a result of the analysis in relation to input parameters was discussed. The finite element model is generated in the ANSYS 16 software and the stresses, deflections are obtained. The results are taken for the cylinder, piston rod and flange of the hydraulic cylinder model using EN8 material. The effect of weight and pressure are taken into consideration. The explicit dynamic analysis was mainly considered for analysis the hydraulic cylinder model. This was clearly explained the result of the hydraulic cylinder.

All components and parts of the double acting hydraulic cylinder have been carefully developed and modelled using Catia workspace. The 3D models of the parts and the pictorial simulation results for both the stress analysis and deformation were represented below.

4.2. Results

Table 4. 1. Result of analysis

No.	Part name	Symbol	Value	Unit
1	Inertia	I	5.02×10^{-9}	mm^4
2	Piston rod	D	20	mm
3	Effective area	A	1.33×10^{-3}	mm^2
4	Piston diameter	Dp	45	mm
5	Annual area	A annual	12.5×10^{-4}	m^2
6	Working stress	δm	183×10^6	Pa
7	Outer diameter	OD	52	mm
8	Thickness of cylinder	T	3.5	m
9	Hoop stress	δh	174.1	MPa
10	Longitudinal stress	δl	74.15	MPa
11	Volumetric flow	Q	2.6×10^{-5}	m^3/s
12	Torque	T	2.4	Nm
13	T plus	-	3.9	Sec
14	Tminus	-	0.604	Sec
15	Speed	V	0.2047	m/s
16	Thickness of end plug	T	4.8	Mm

1. Result of equivalent (von-Mises) stress distributions on the piston rod model with a 10 kN along with the direction of the impact with a fixed wall.

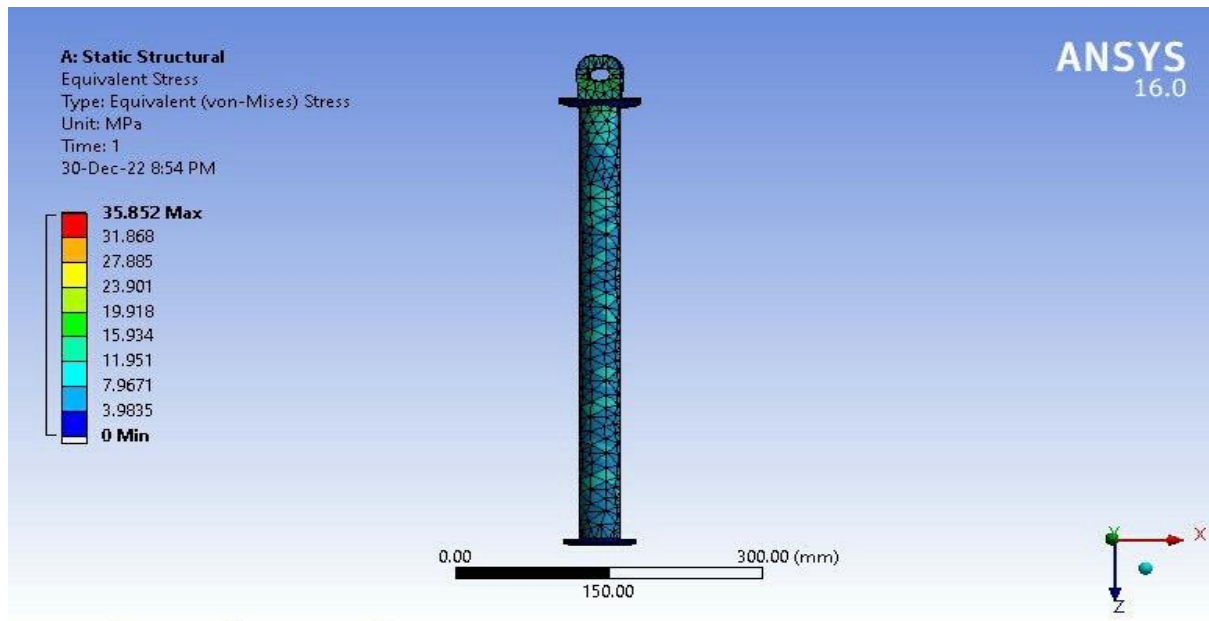


Figure 4. 1. Analysis result of the piston rod model with a mass 10kN

The total von-Mises stress value of the piston rod with an applied input weight of 10kN is shown in figure 4.1. The total maximum von-Mises stress is 35.852 Mpa.

2. Result of total deformation of the piston rod model with a 10kN along with the direction of the impact with a fixed wall.

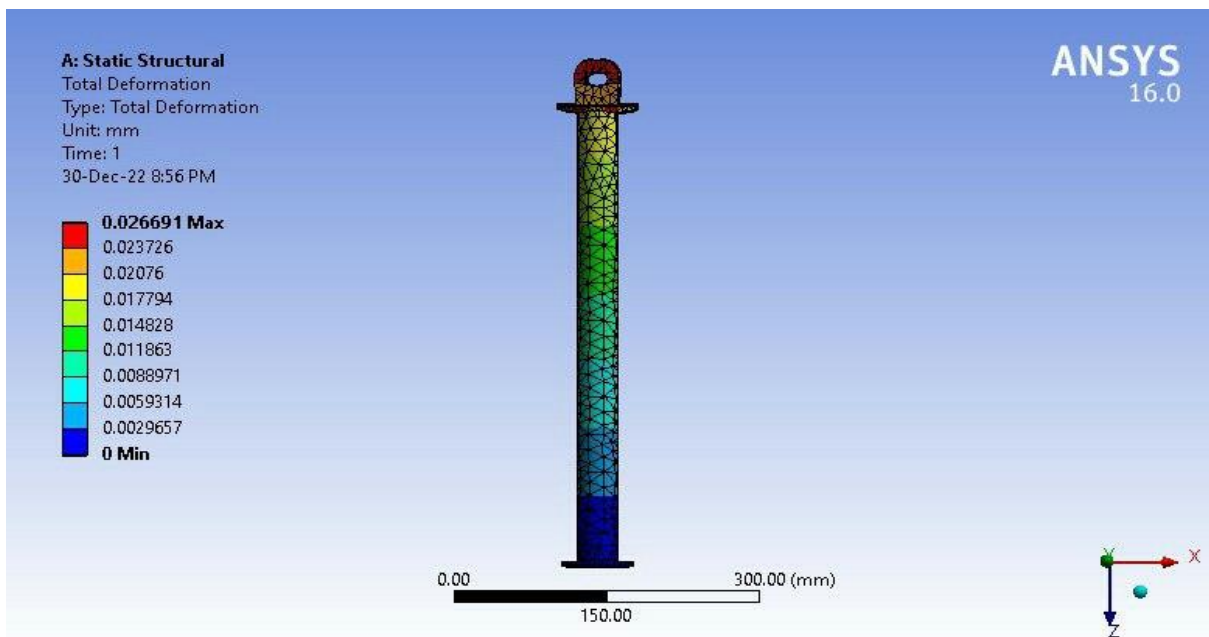


Figure 4. 2. Analysis result of the total deformation of piston rod model with a mass of 10 kN
The total deformation value of the piston rod with an applied input weight of 10kN is shown in the above figures. The total maximum deformation is 0.026691mm.

3. Result of equivalent (von-Misses) stress distributions on the flange model with a 10kN along with the direction of the impact with a fixed wall.

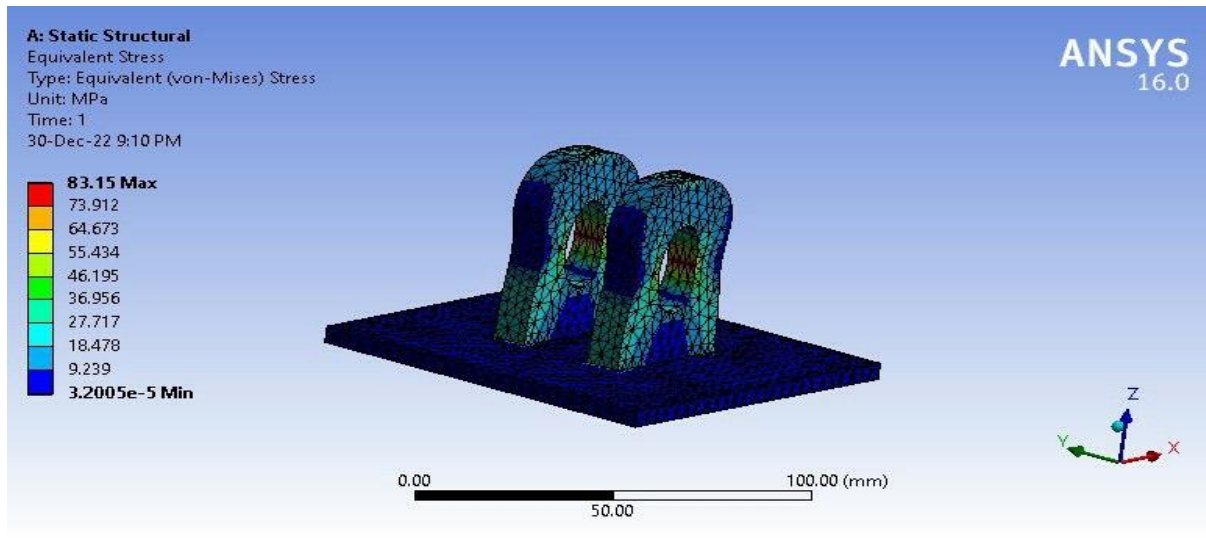


Figure 4. 3. Analysis result of the flange model with a mass 10 kN.

The total von-Misses stress value of the flange with an applied input weight of 10 kN is shown in the above figures. The total maximum von-Misses stress is 83.15 Mpa.

4. Result of total deformation of the flange with 10 kN along with the direction of the impact with a fixed wall.

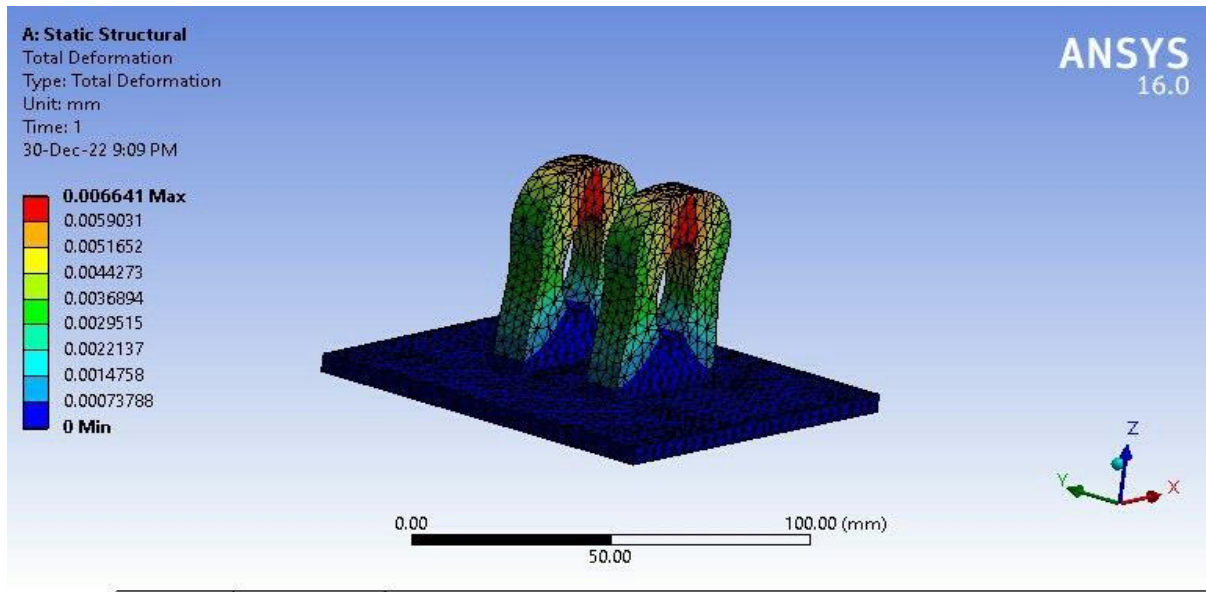


Figure 4. 4. Analysis result of the flange model with a mass 10 kN

The total deformation value of the flange with applied input weight of 10kN is shown in the above figure 4.4. The total maximum deformation is 0.006641mm.

5. Result of equivalent (von-Mises) stress distributions on the cylinder model with a 250bar along with the direction of the impact with a fixed wall.

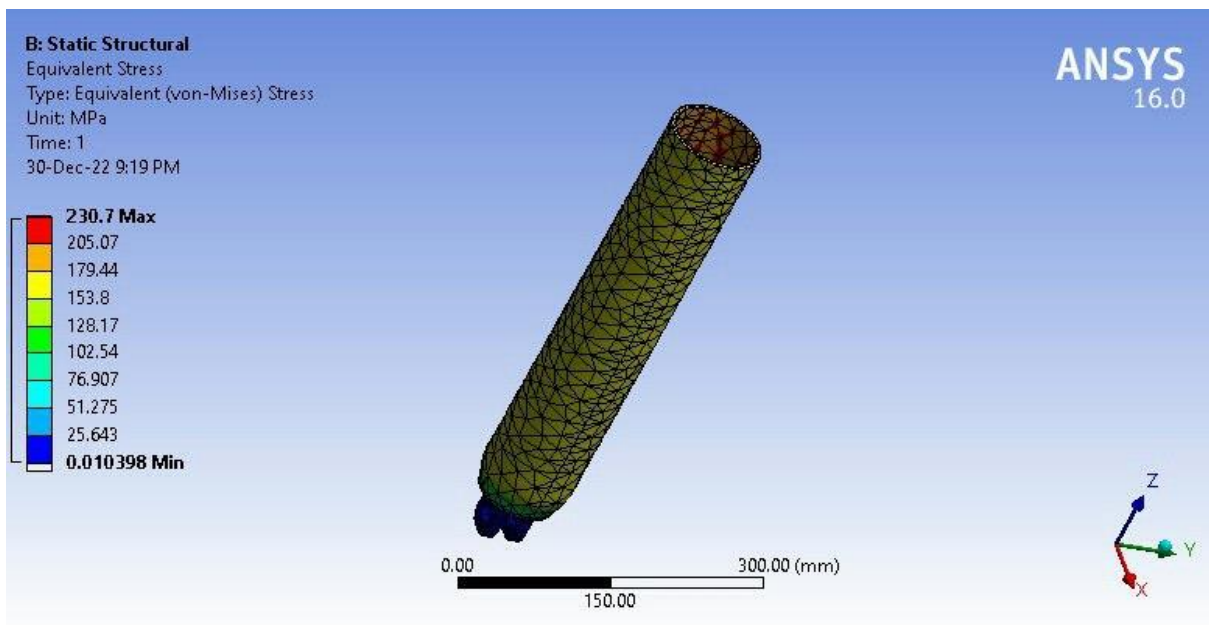


Figure 4. 5. Analysis result of the cylinder model with a pressure of 250bar.

The total von-Mises stress value of the cylinder with an applied input pressure of 250bar is shown in the above figure 4.5. The total maximum von-Mises stress is 230.7 MPa.

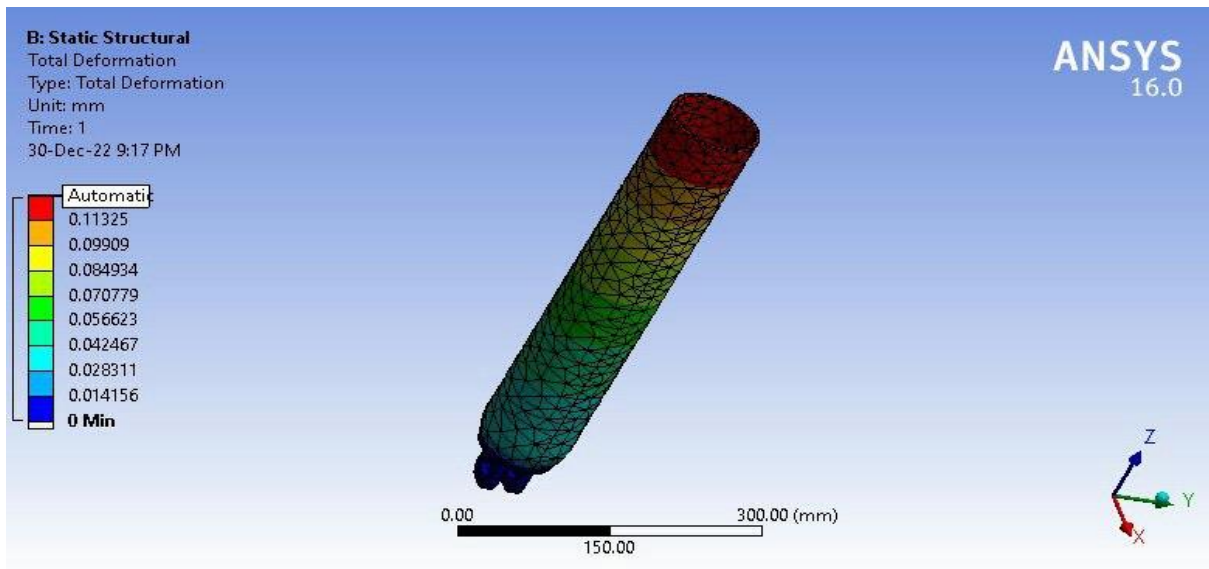


Figure 4. 6. Analysis result of the cylinder model with a pressure of 250 bar

The total deformation value of the cylinder with applied input pressure of 250 bar is shown in the above figure 4.6. The total maximum deformation is 0.11325mm.

A 3D double acting model of hydraulic cylinder was designed for BS 970 080M40 (medium carbon steel) with the specifications.

And from the simulation results as can be seen in figure 4.6 even when subjected to an internal pressure of 25MPa, the maximum stress experienced at the ends was less than the Yield strength of material. It can as well be seen from the stress analysis results that the von Moses stress on both the flange and piston and piston rod assembly did not exceed the Yield strength of BS 970 080M40 (medium carbon steel). This means this material can be used for the cylinder.

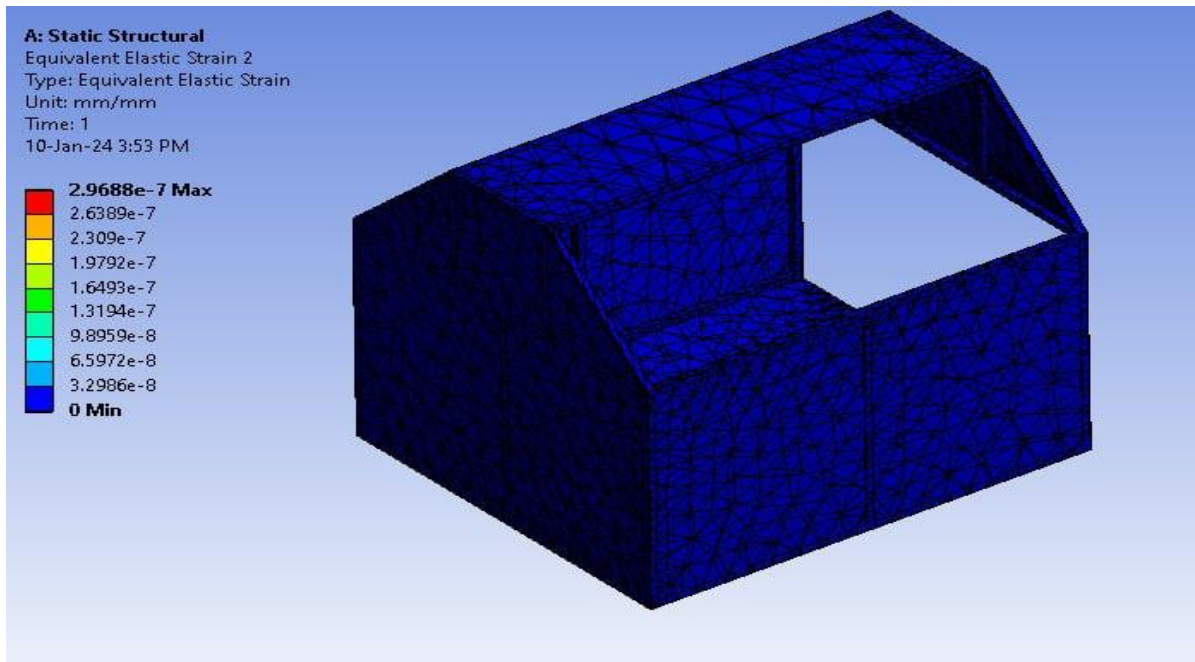


Figure 4. 7. Equivalent elastic strain

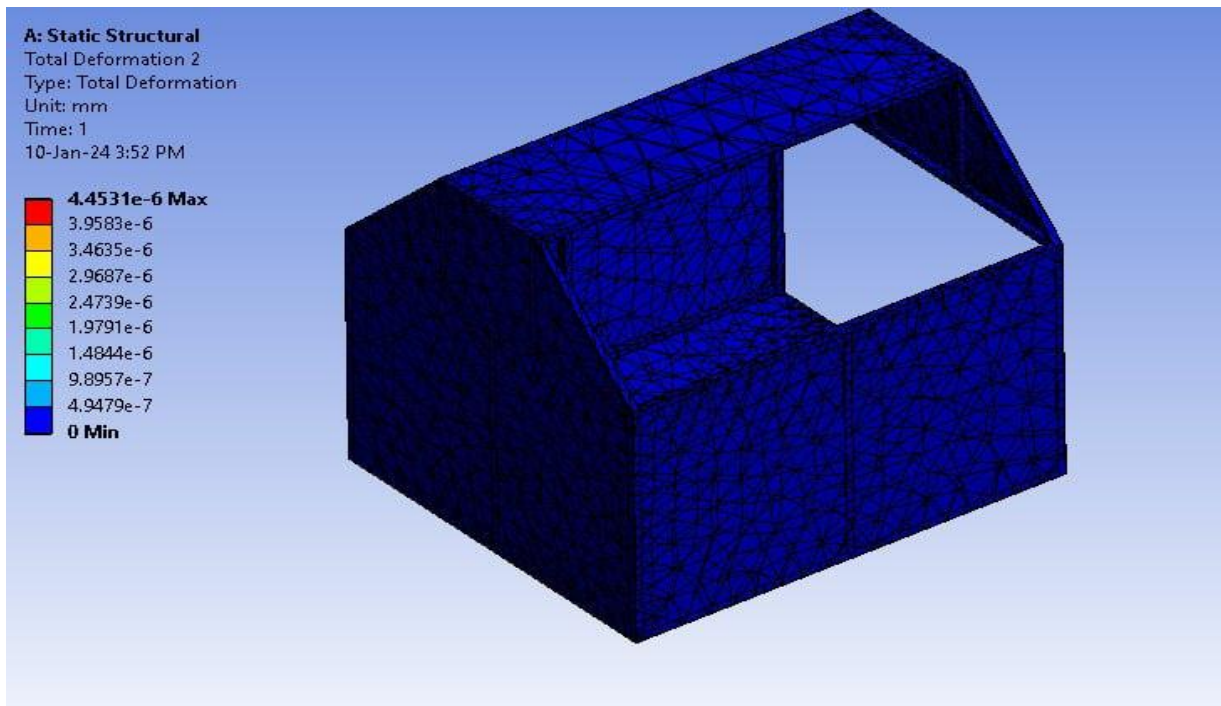


Figure 4. 8. Total deformation

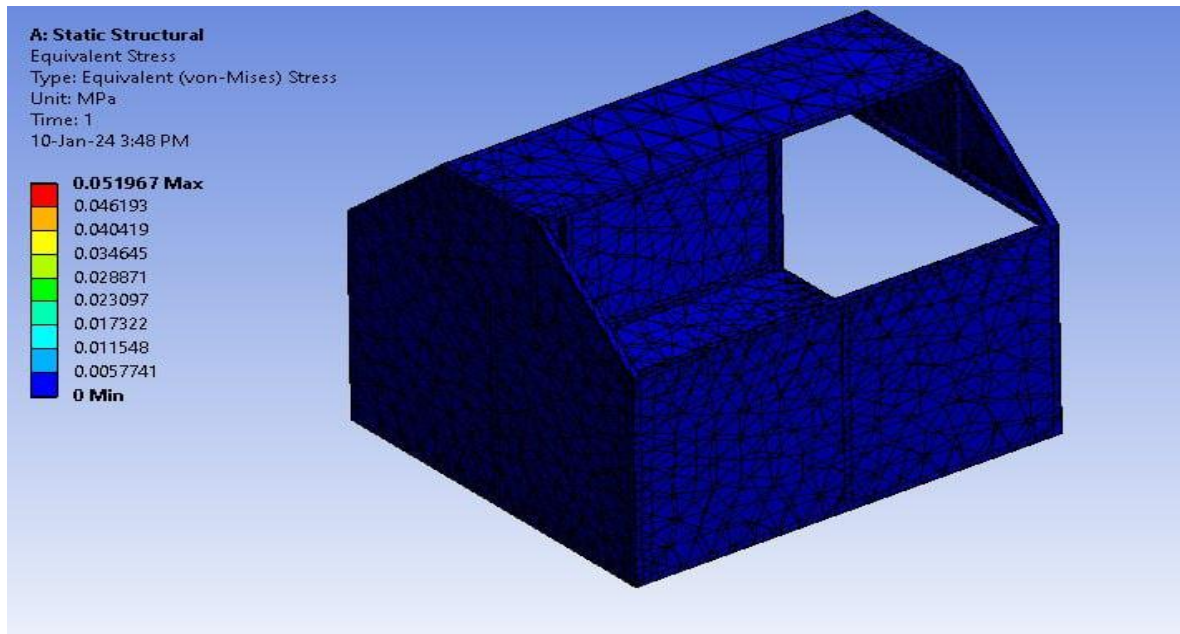


Figure 4. 9. Equivalent (von-mises) stress

His simulation of the Maxima Load Bajaj hydraulic cylinder is shown below.

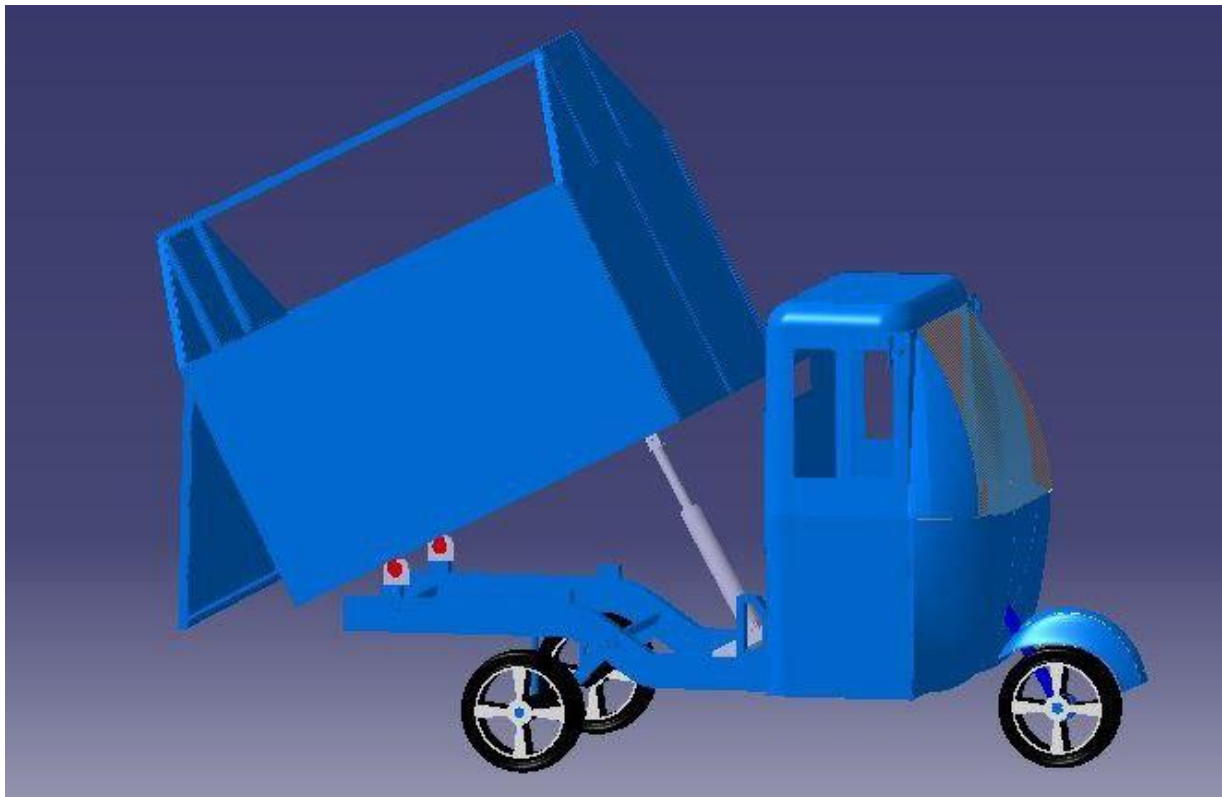


Figure 4. 10. 3D maxima cargo Bajaj

4.3. Summary

Finally, the result obtained in the analysis are given in Table 4.2.

Table 4. 2. Analysis results of von Misses stress on both the flange model, cylinder model and piston rod model

No.	Type	Unit	Min.	Max.
1	Equivalent (von-Misses) stress distributions on the piston rod model	Mpa	0	38.852
2	Total deformation of the piston rod model	Mm	0	0.026691
3	Equivalent (von-Misses) stress distributions on the flange model	Mpa	3.2005e-5	83.15
4	Total deformation of the flange model	Mm	0	0.006641
5	Equivalent (von-Misses) stress distributions on the cylinder model with a 250bar.	Mpa	0.010398	230.7
6	Total deformation of cylinder model with a pressure of 250 bar	Mm	0	0.11325

CHAPTER FIVE

CONCLUSION AND RECOMMENDATION

5.1. Conclusion

I have successfully designed and analysed a double-acting hydraulic cylinder. We selected the materials and selected the seals taking into account the relevant standards and regulations. According to findings revealed about Design of Hydraulic Cylinder and Body Modification for Maxima Cargo Bajaj in general, the following conclusions are drawn:

- The design of 1 double-acting hydraulic cylinder is completed and ready for production.
- The outer and inner cylinder diameters are set to 52mm, the piston diameter with seal is 45mm, the piston rod diameter is 20mm, and the stroke length is 800mm.
- FEA analysis and simulation of Hydraulic Cylinder 2 performed at Maxima Cargo Bajaj provided reliable confirmation of the reliability, functionality and safety of the designed hydraulic cylinder.
- Double-acting hydraulic cylinder ideal for industrial automation applications.

5.2. Recommendations

It would like to recommend the future researcher should continue to work in this direction and design system of the same title and try to manufacture (fabricate) the prototype of hydraulic cylinder and it recommends that maxima cargo companies to use this system for all vehicles.

5.3. Future Works

For the further development of the hydraulic cylinder there are things which deserve attention.

Sealing: The sealing system of the cylinder is possibly over engineered. By simplifying the sealing system costs could be driven down at the cost of performance. Also, the seals chosen now are chosen for basic mineral hydraulic oil. If more ecological hydraulic oil is used, then the compatibility of the seals and hoses must be check

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APPENDICES

Appendices I. Specification of materials

Table I. 1. Result of analysis

S/N	Part name	Symbol	Value	Unit
1	Inertia	I	5.02×10^{-9}	mm^4
2	Piston rod	D	20	Mm
3	Effective area	A	1.33×10^{-3}	mm^2
4	Piston diameter	Dp	45	Mm
5	Annual area	A annual	12.5×10^{-4}	m^2
6	Working stress	δm	183×10^6	Pa
7	Outer diameter	OD	52	Mm
8	Thickness of cylinder	T	3.5	Mm
9	Hoop stress	δh	174.1	Mpa
10	Longitudinal stress	δl	74.15	Mpa
11	Volumetric flow	Q	2.6×10^{-5}	m^3/s
12	Torque	T	2.4	Nm
13	T plus	-	3.9	Sec
14	T minus	-	0.604	Sec
15	Speed	V	0.2047	m/s
16	Thickness of end plug	T	4.8	Mm

Table I. 2. Analysis results of von Misses stress on both the flange model, cylinder model and piston rod model

No.	Type	Unit	Min.	Max.
1	Equivalent (von-Misses) stress distributions on the piston rod model	Mpa	0	38.852
2	Total deformation of the piston rod model	Mm	0	0.026691
3	Equivalent (von-Misses) stress distributions on the flange model	Mpa	3.2005e-5	83.15
4	Total deformation of the flange model	Mm	0	0.006641
5	Equivalent (von-Misses) stress distributions on the cylinder model with a 250bar.	Mpa	0.010398	230.7
6	Total deformation of cylinder model with a pressure of 250 bar	Mm	0	0.11325

Appendices II. AutoCAD Results

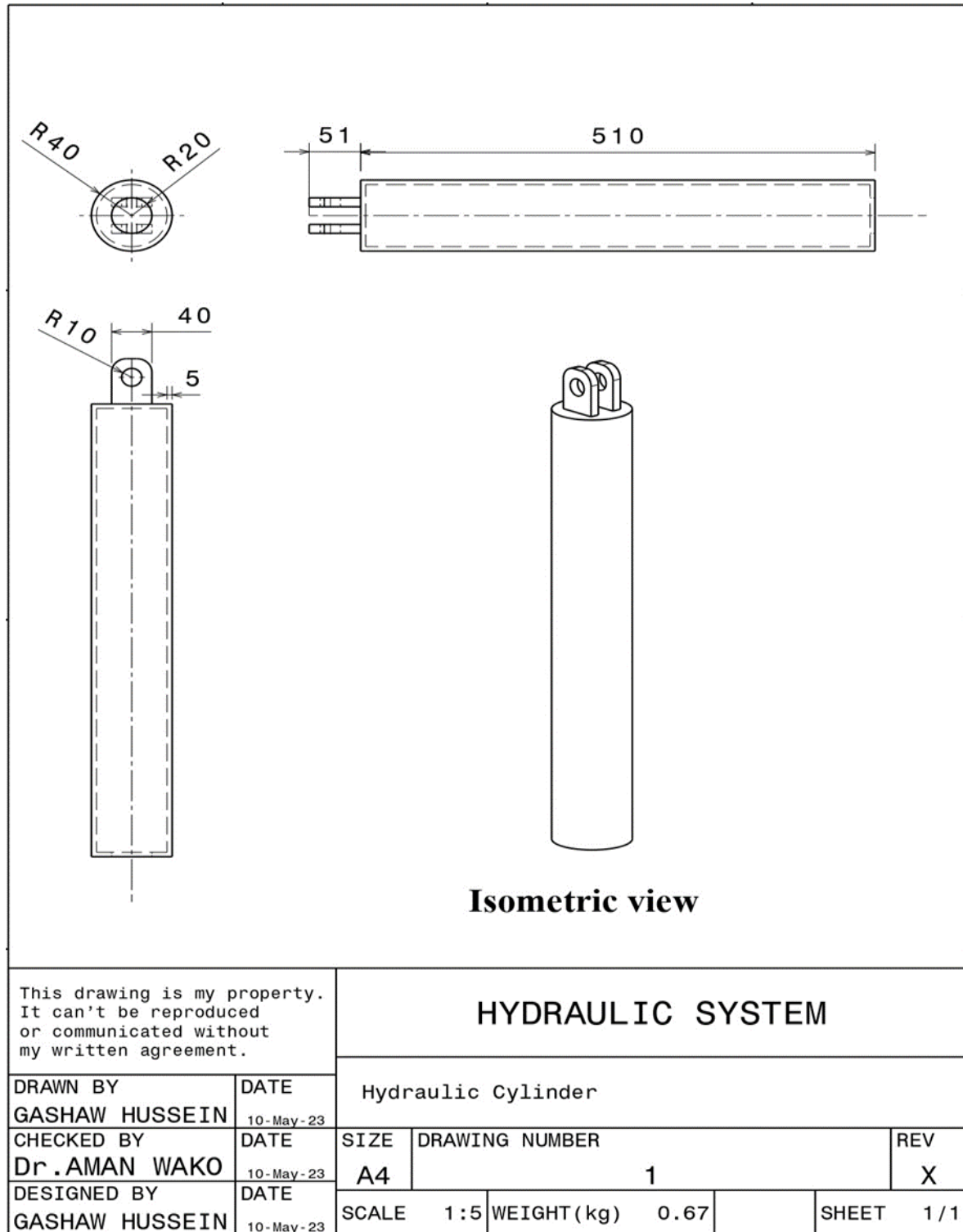


Figure II. 1. Hydraulic Cylinder

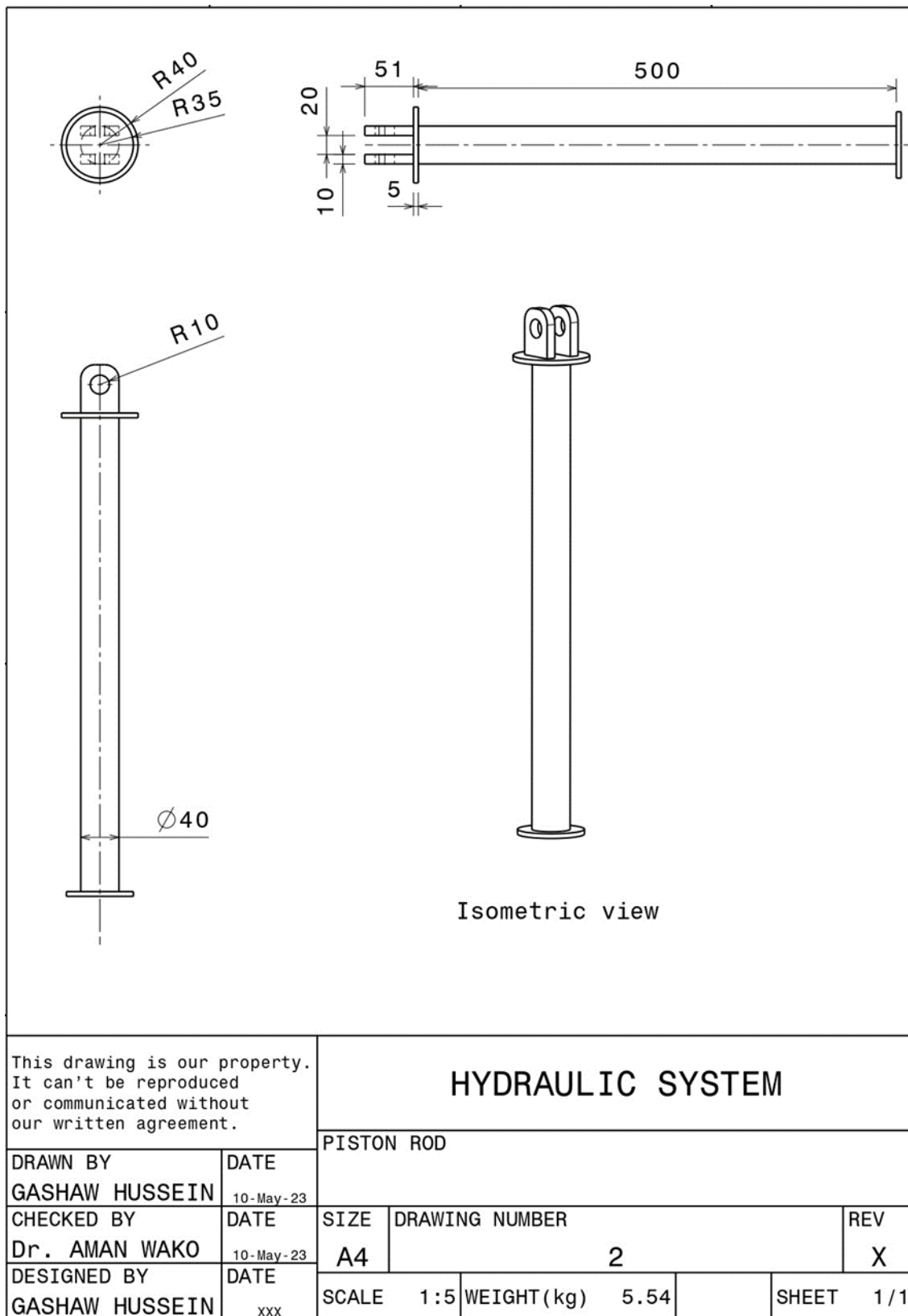
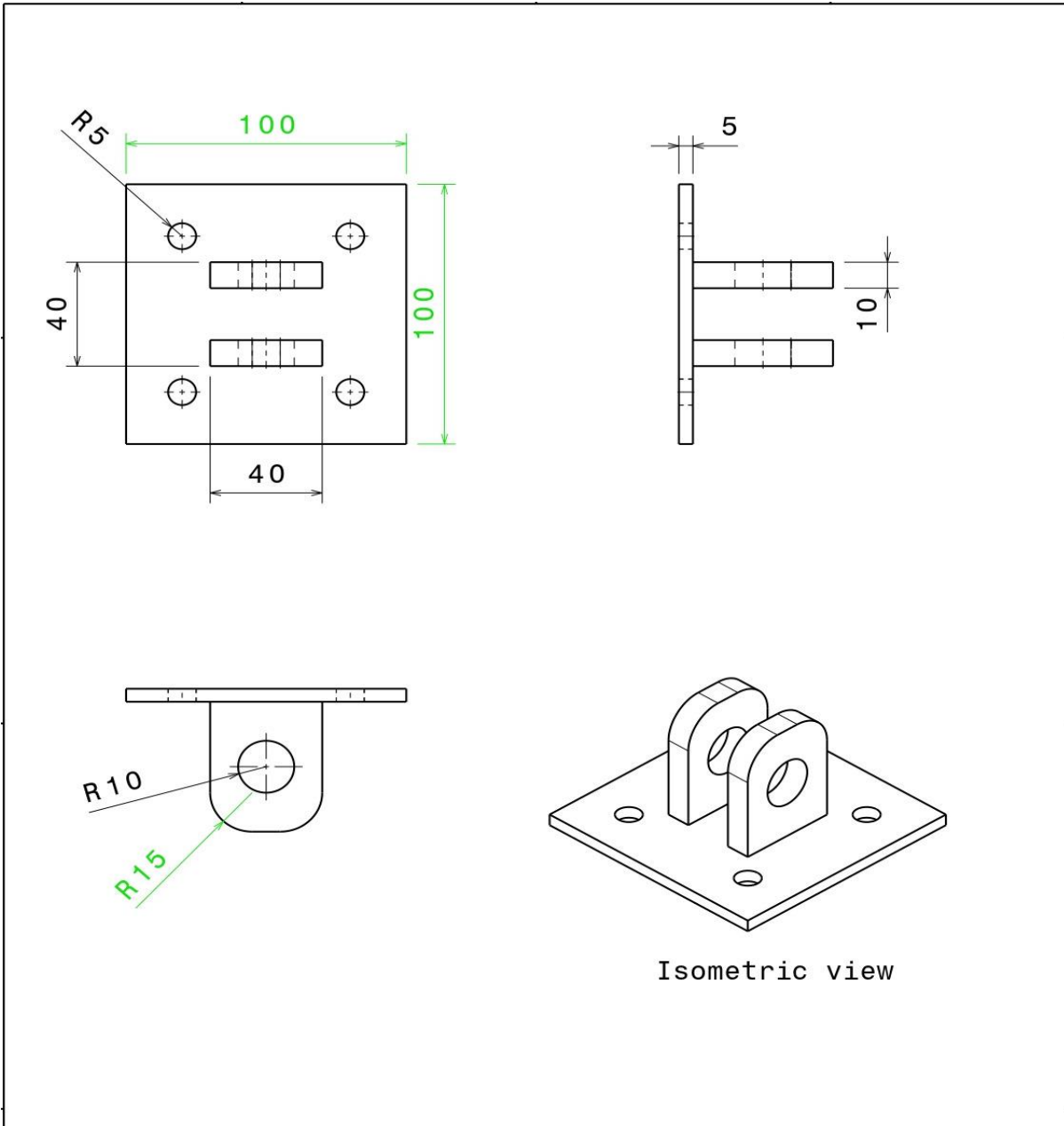
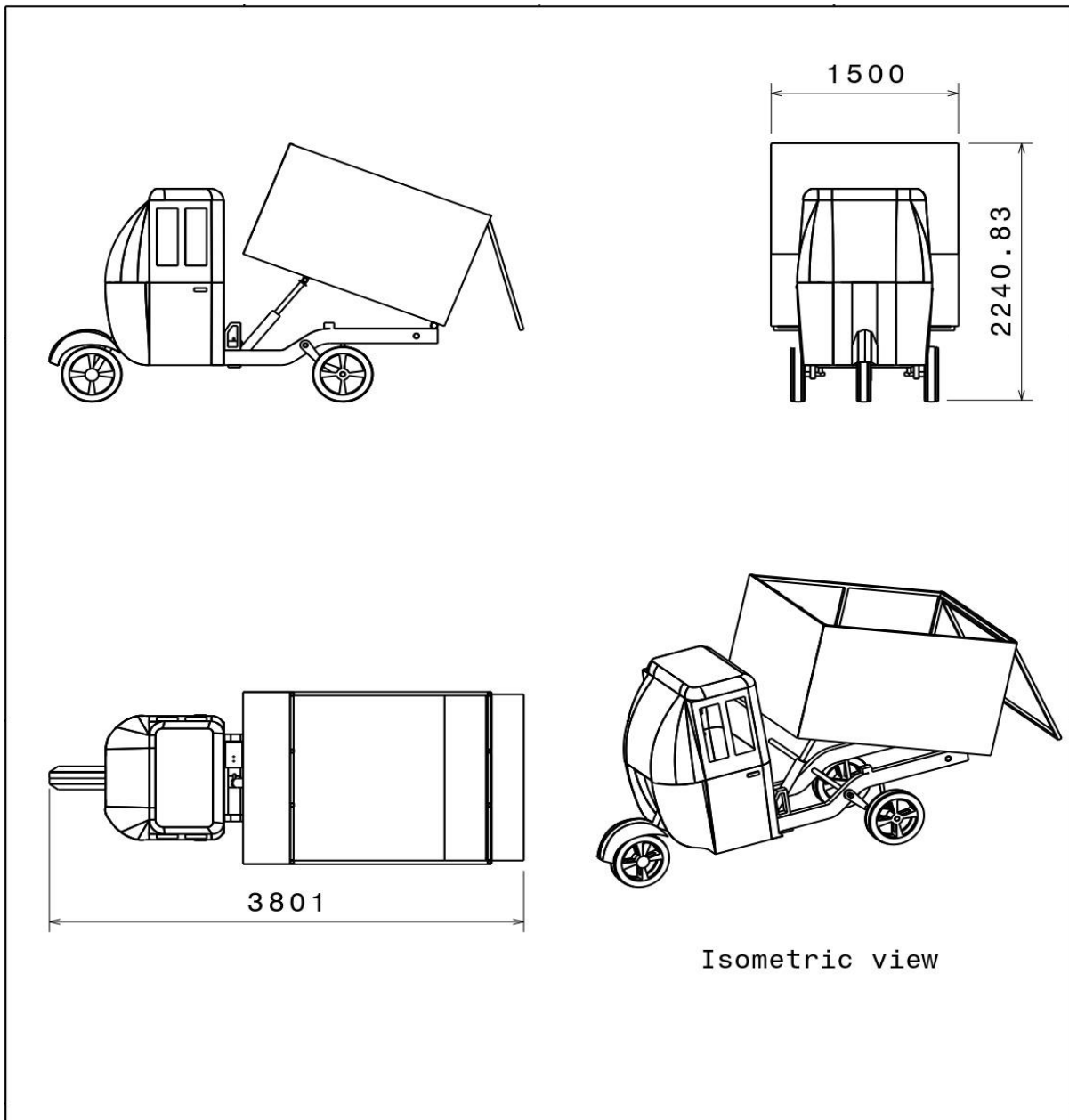


Figure II. 2. Piston rod



This drawing is my property. It can't be reproduced or communicated without my written agreement.		Hydrolic System			
		FLANGE			
DRAWN BY GASHAW HUSSEIN	DATE 11-May-23	SIZE A4	DRAWING NUMBER 3		
CHECKED BY Dr.AMAN WAKO	DATE 11-May-23	SCALE 1:2	WEIGHT (kg) 0.08	SHEET 1/3	
DESIGNED BY GASHAW HUSSEIN	DATE 11-May-23				

Figure II. 3. Flanger



This drawing is my property. It can't be reproduced or communicated without our written agreement.		MAXIMA CARGO BAJAJ			
DRAWN BY Gashaw Hussein		3D ISOMETRIC VEIW			
CHECKED BY Dr AMAN WAKO		SIZE A4	DRAWING NUMBER 4		REV X
DESIGNED BY GASHAW HUSSEIN		SCALE 1:45	WEIGHT (kg) 200.87	SHEET 1 / 1	

Figure II. 4. Isometric view

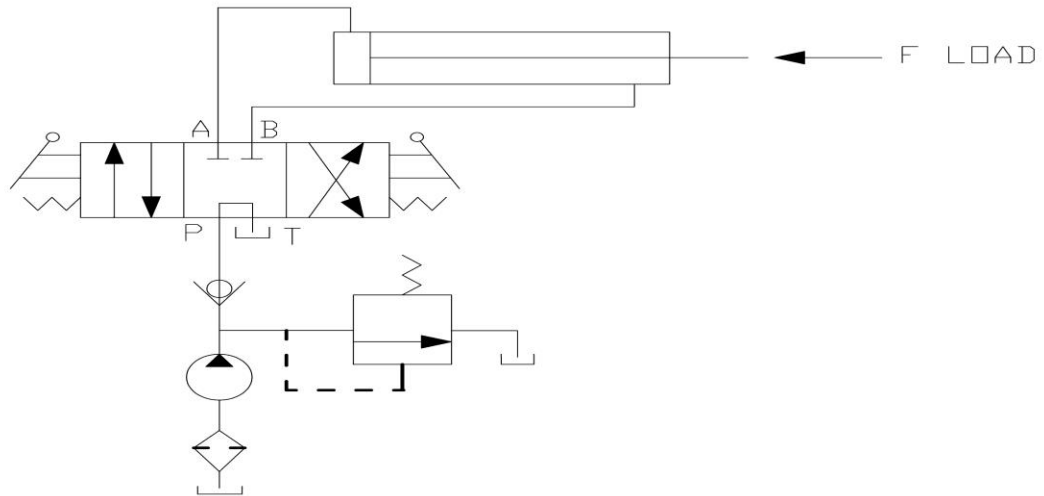


Figure II. 5. Control of double acting hydraulic cylinder

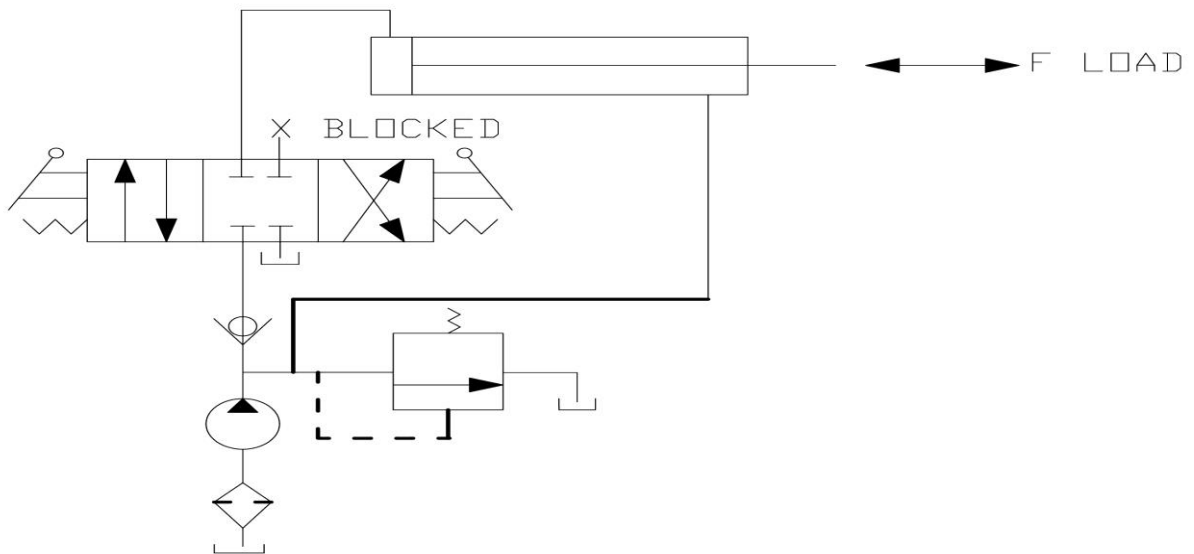


Figure II. 6. Regenerative Cylinder Circuit

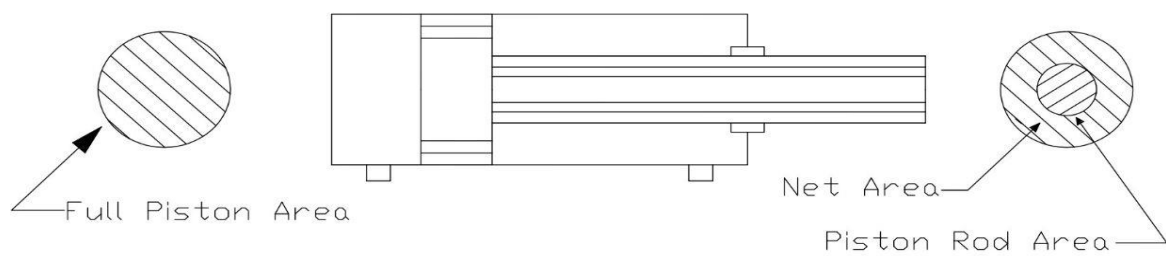


Figure II. 7. Piston Cylinder

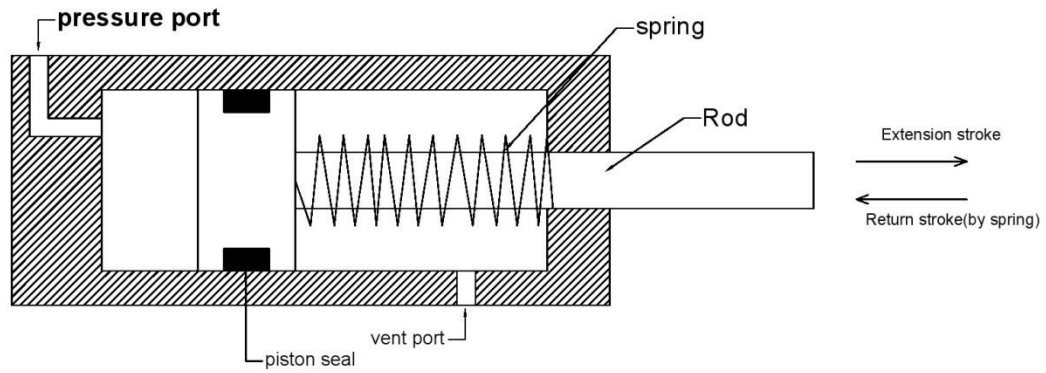


Figure II. 8. Single action cylinder

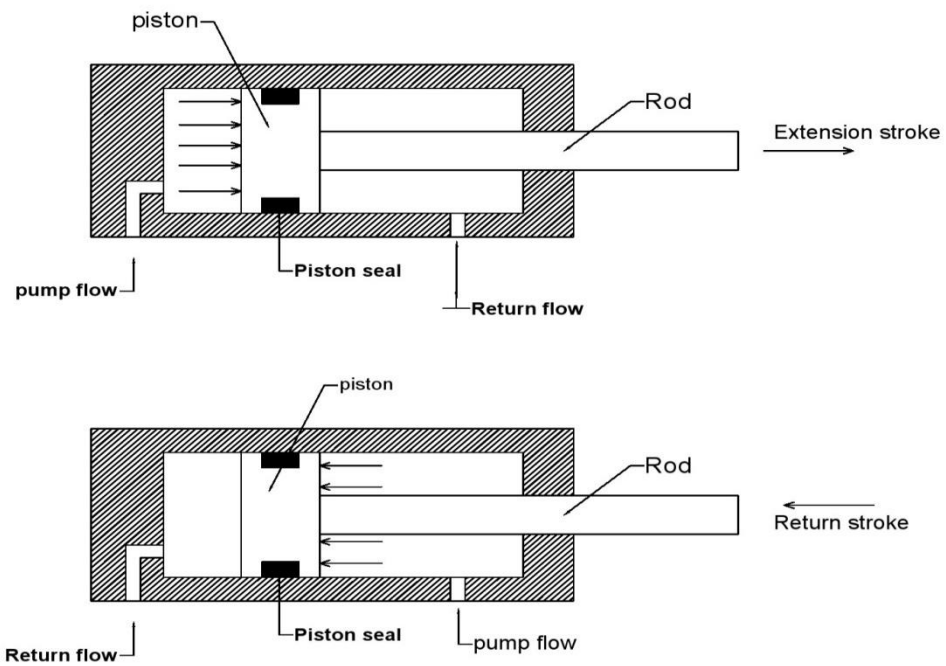


Figure II. 9. Double action hydraulic cylinders

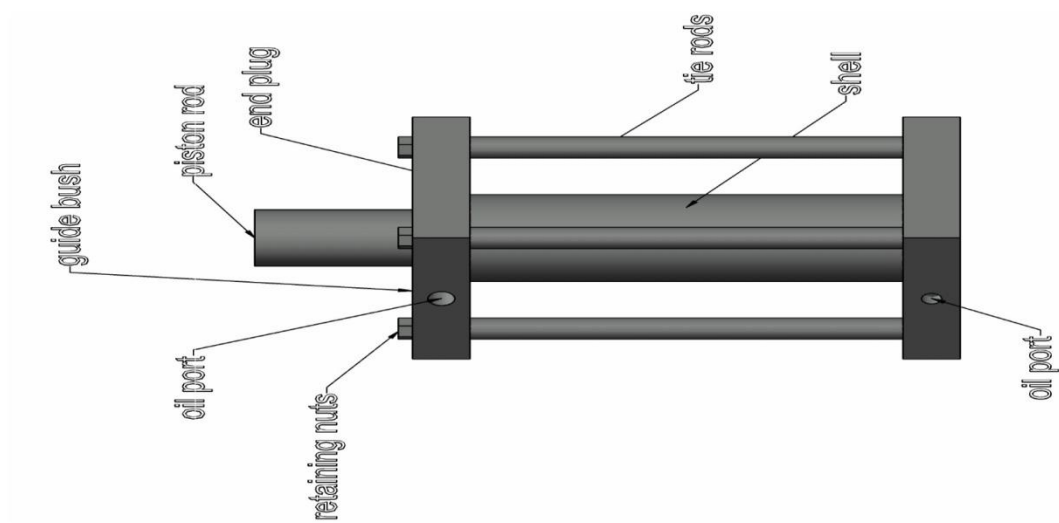


Figure II. 10. Tie - Rod Construction

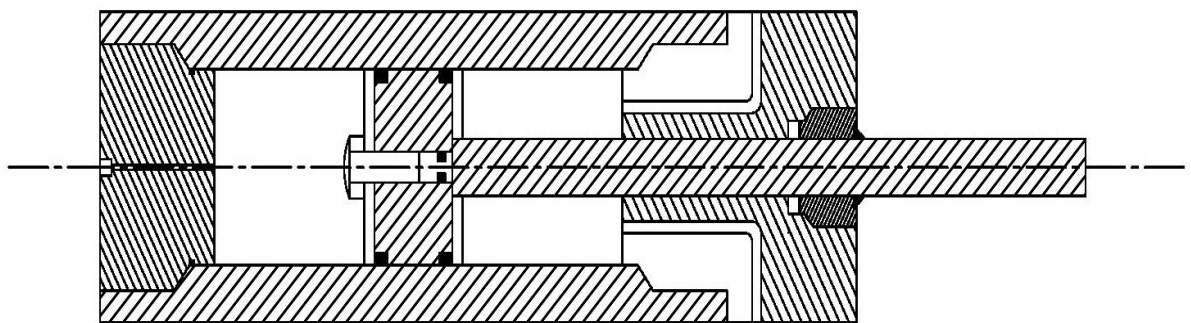


Figure II. 11. Threaded Constructions

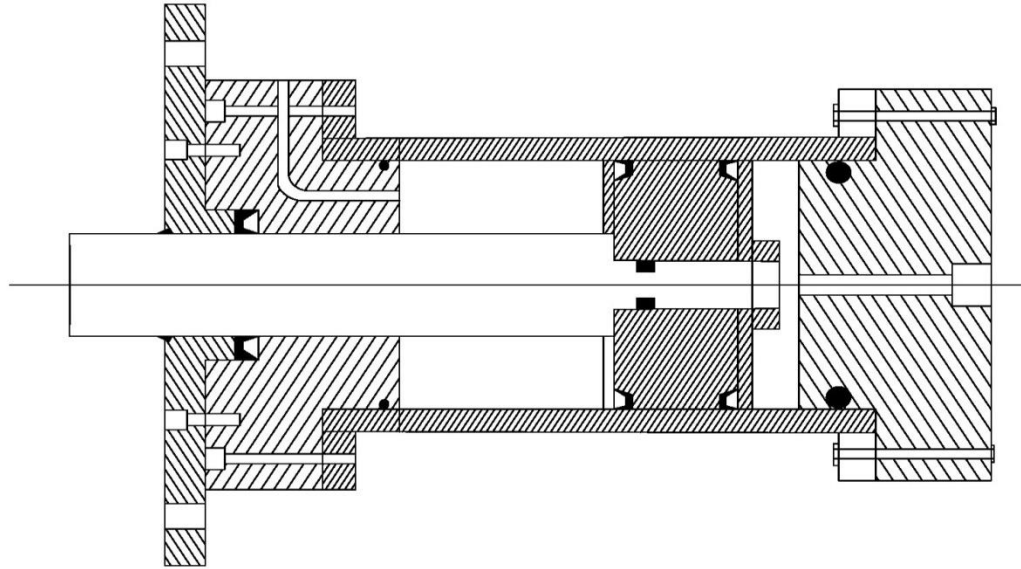


Figure II. 12. Bolted Constructions

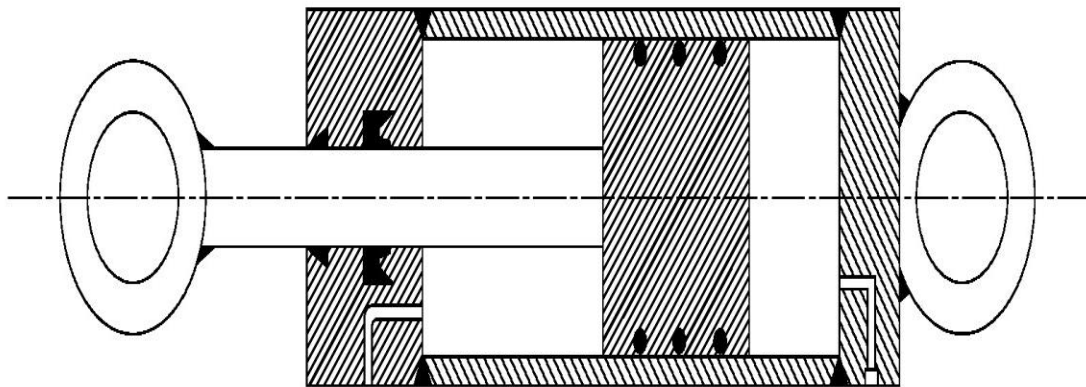


Figure II. 13. One Piece – Welded cylinder

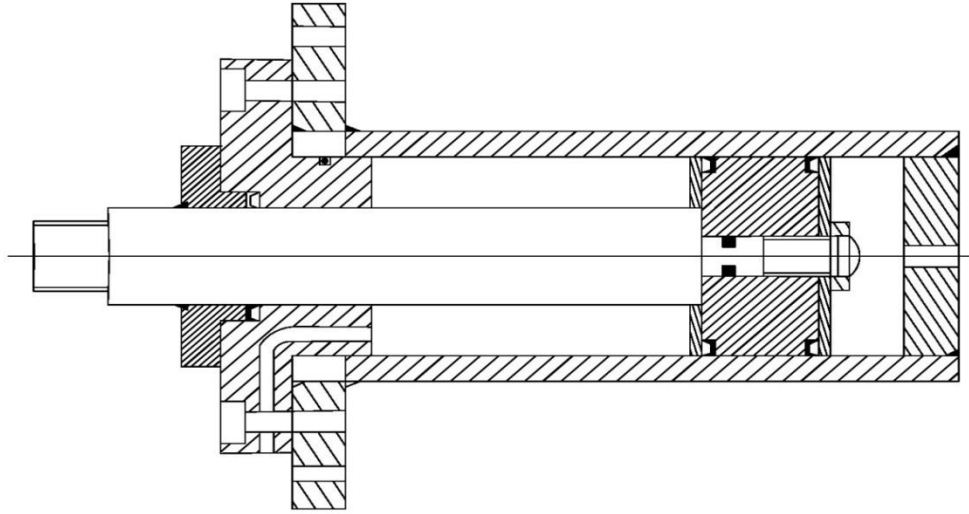


Figure II. 14. Customs - Build Cylinder

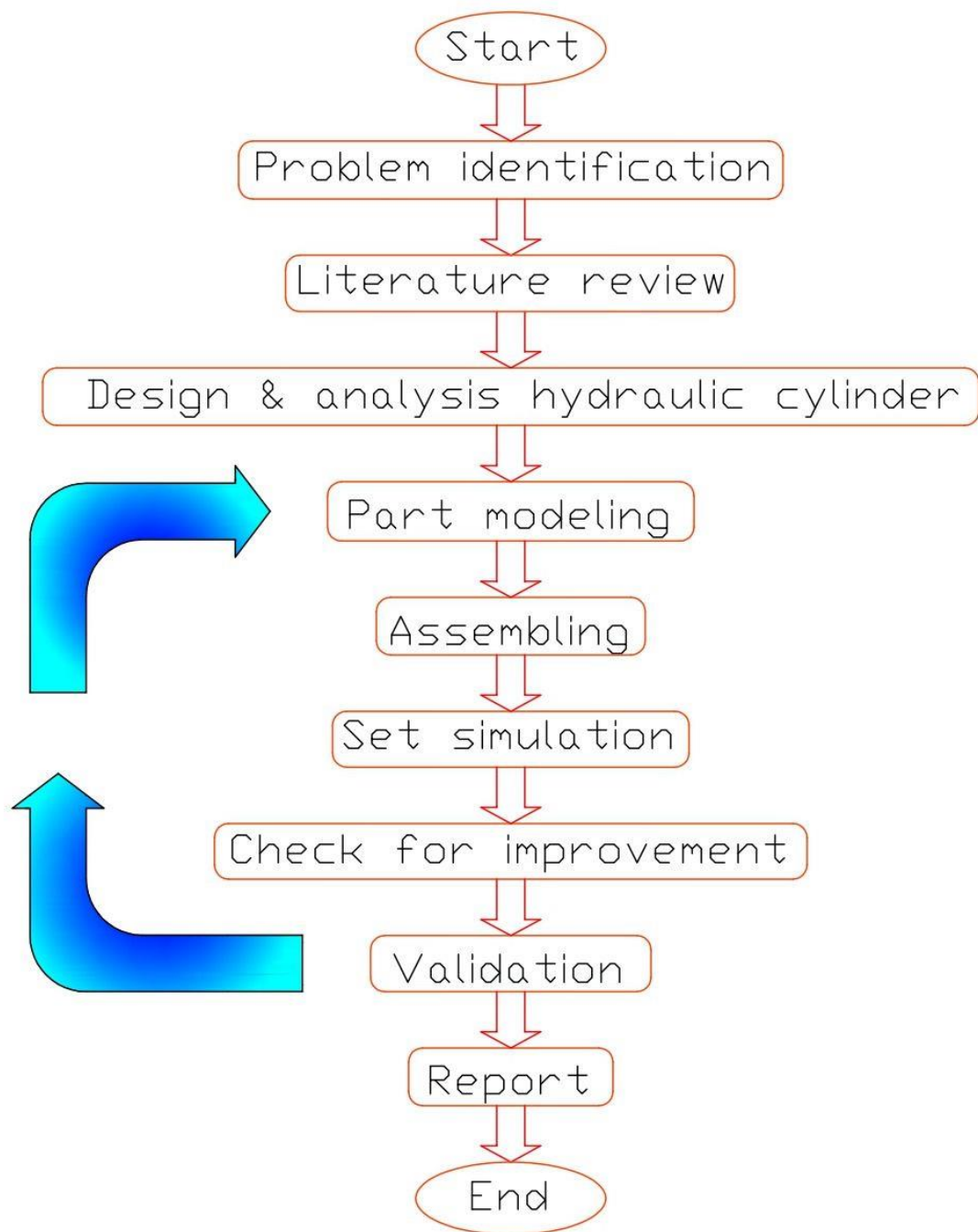


Figure II. 15. Conceptual frame work

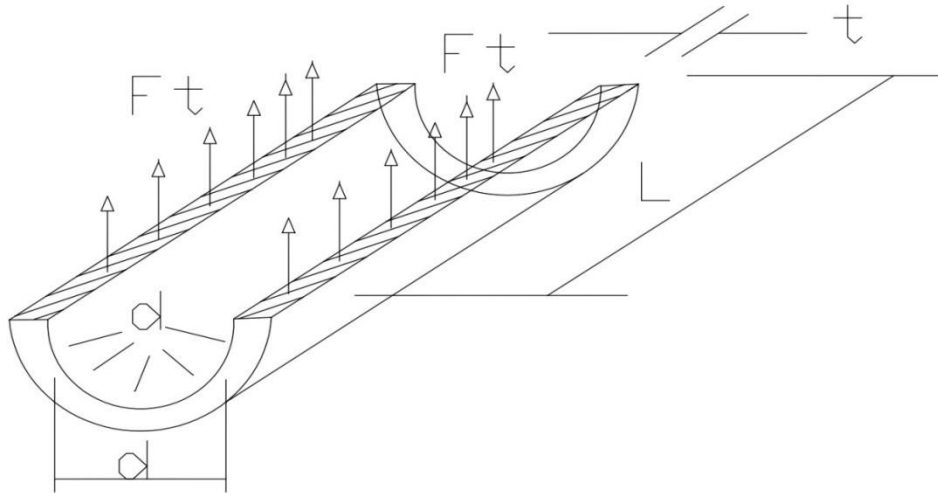


Figure II. 16. Stress acting on direction of tangential circumference

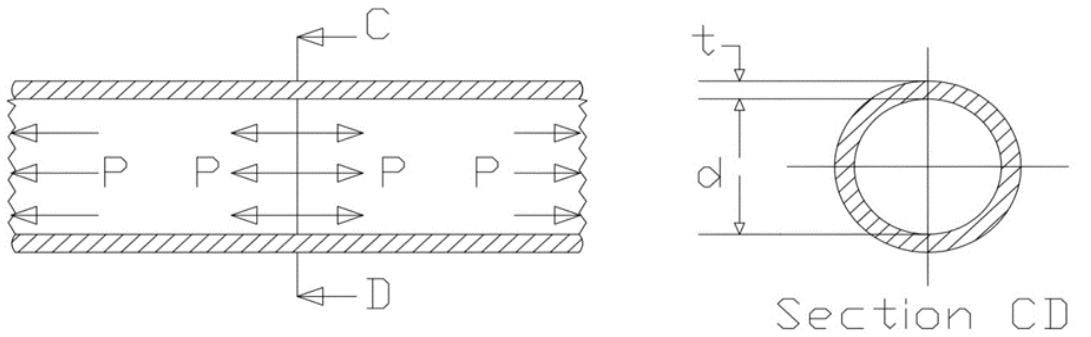


Figure II. 17. Stress acting in longitudinal direction

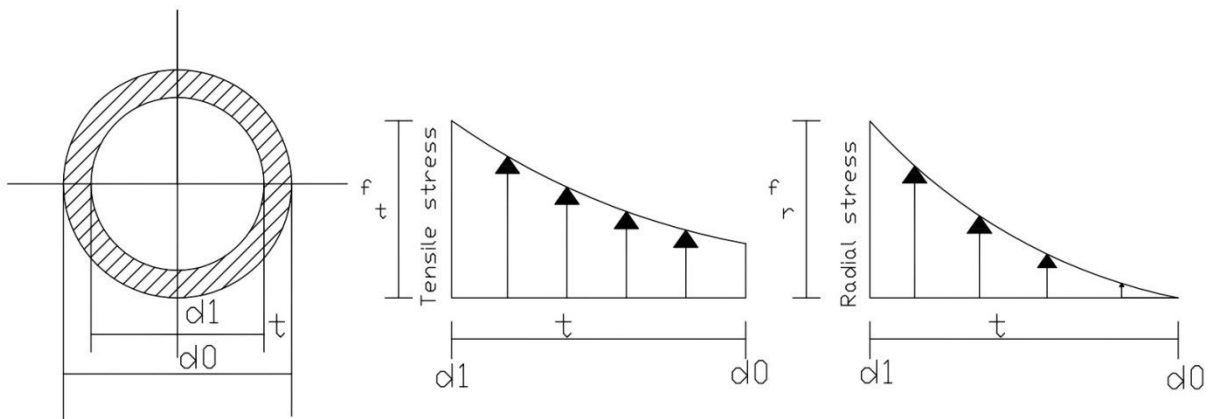


Figure II. 18. Stress distribution in case of thick cylinder

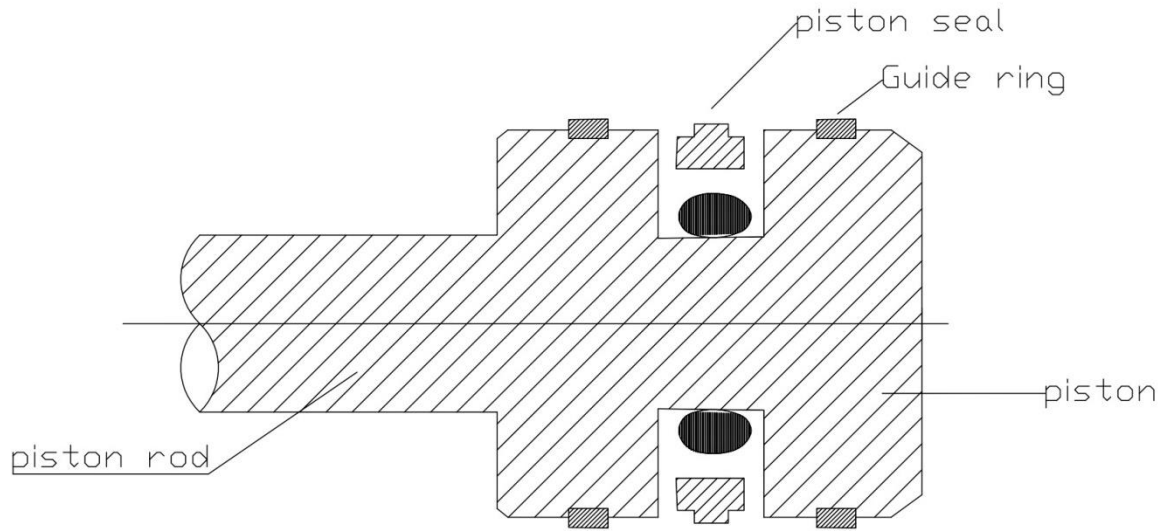


Figure II. 19. Piston as integral part of Piston-rod

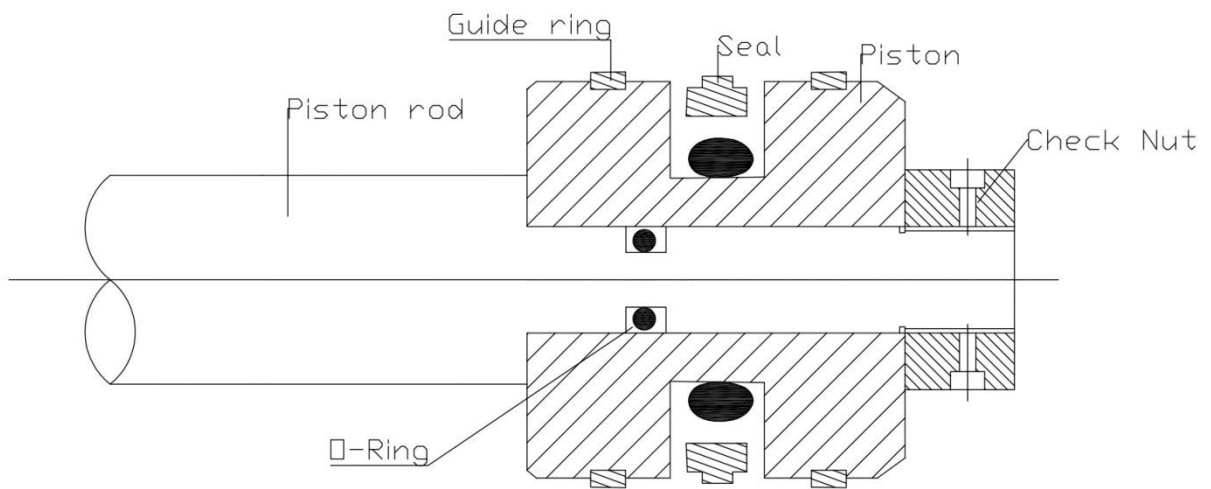


Figure II. 20. Piston assembled on Piston Rod

Appendices III. CATIA Software Results

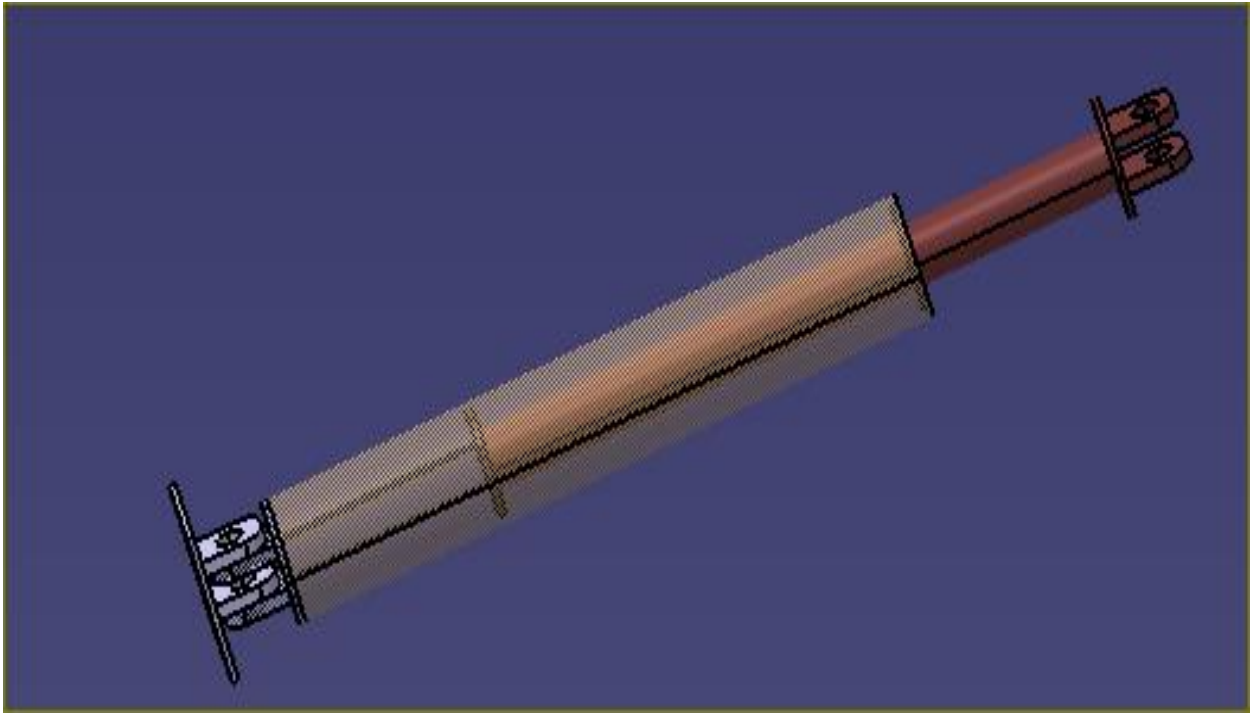


Figure III. 1. Double acting cylinder

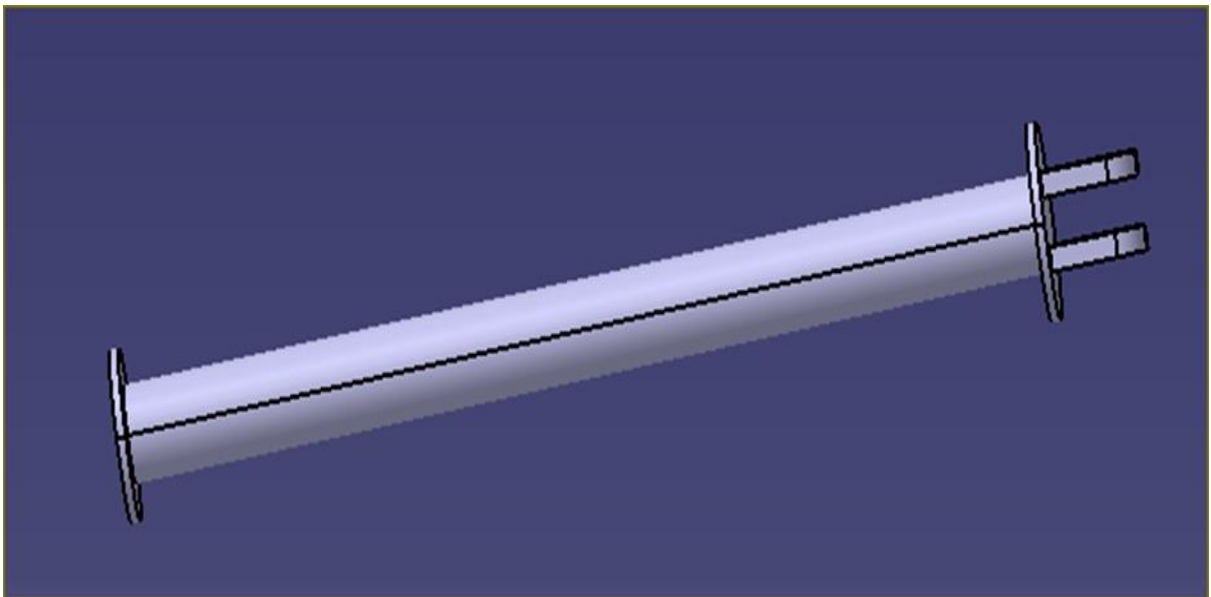


Figure III. 2. Piston rod

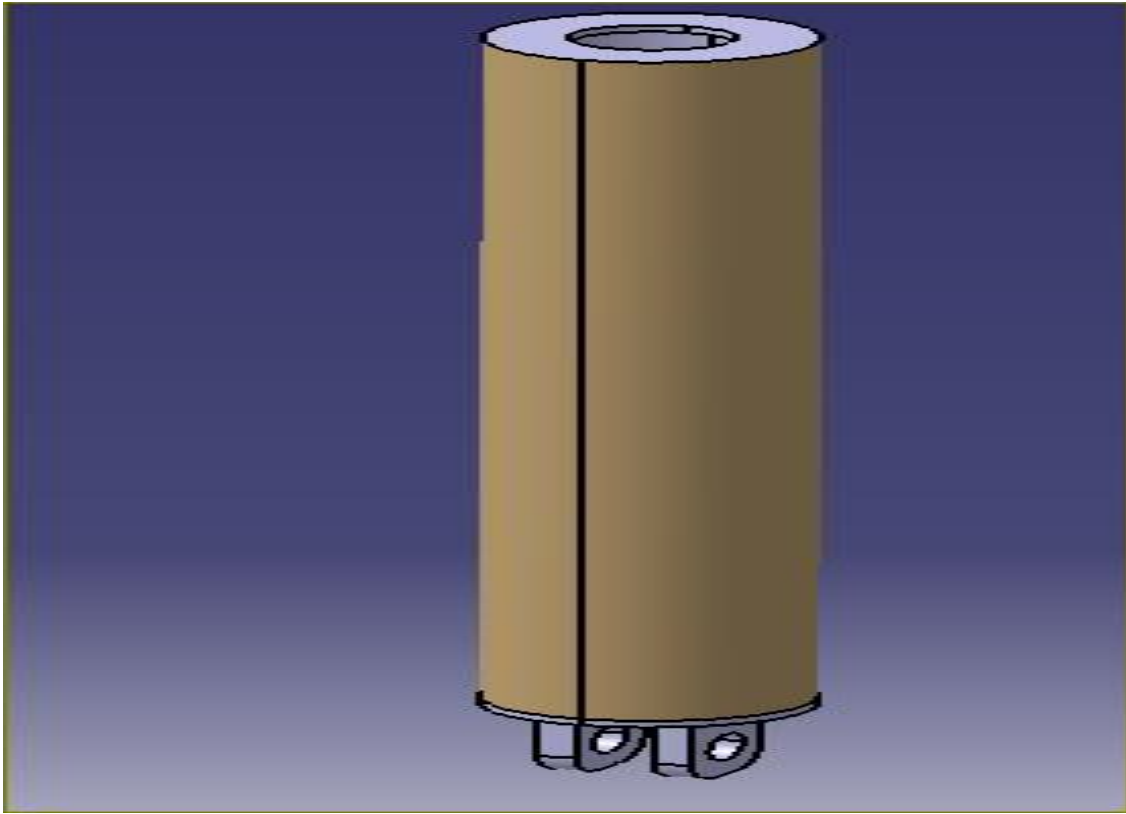


Figure III. 3. Cylinders

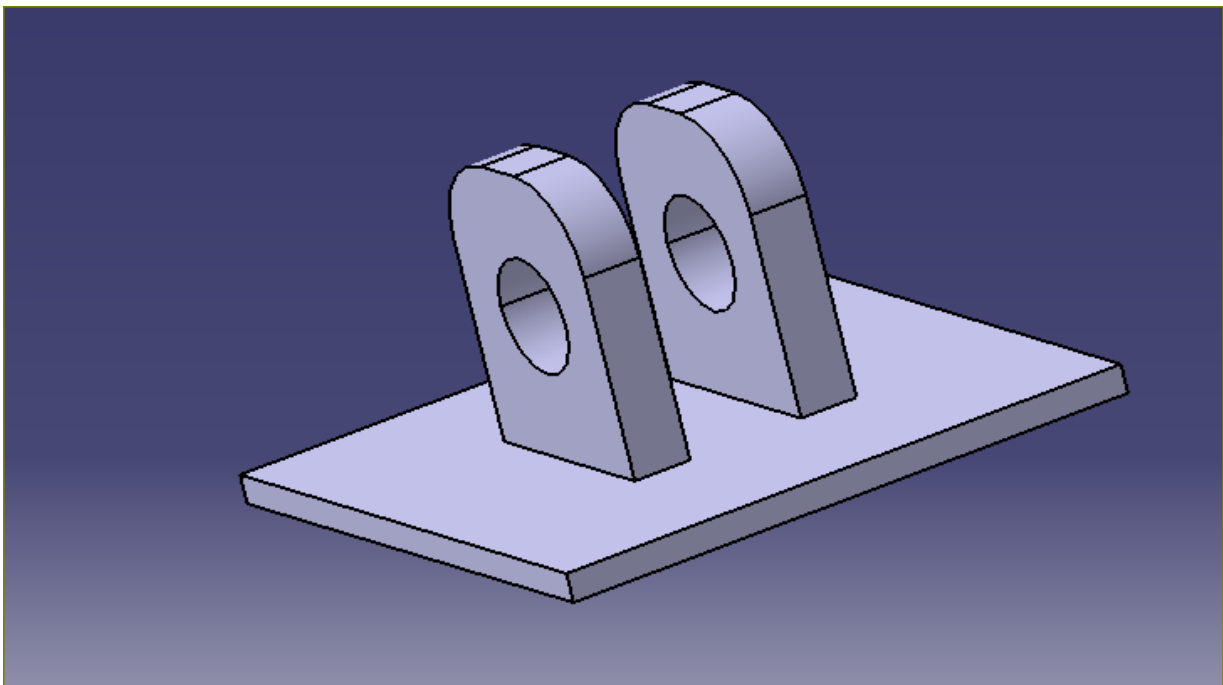


Figure III. 4. Rectangular flanges



Figure III. 5. Analysis result of the piston rod model with a mass 10kN

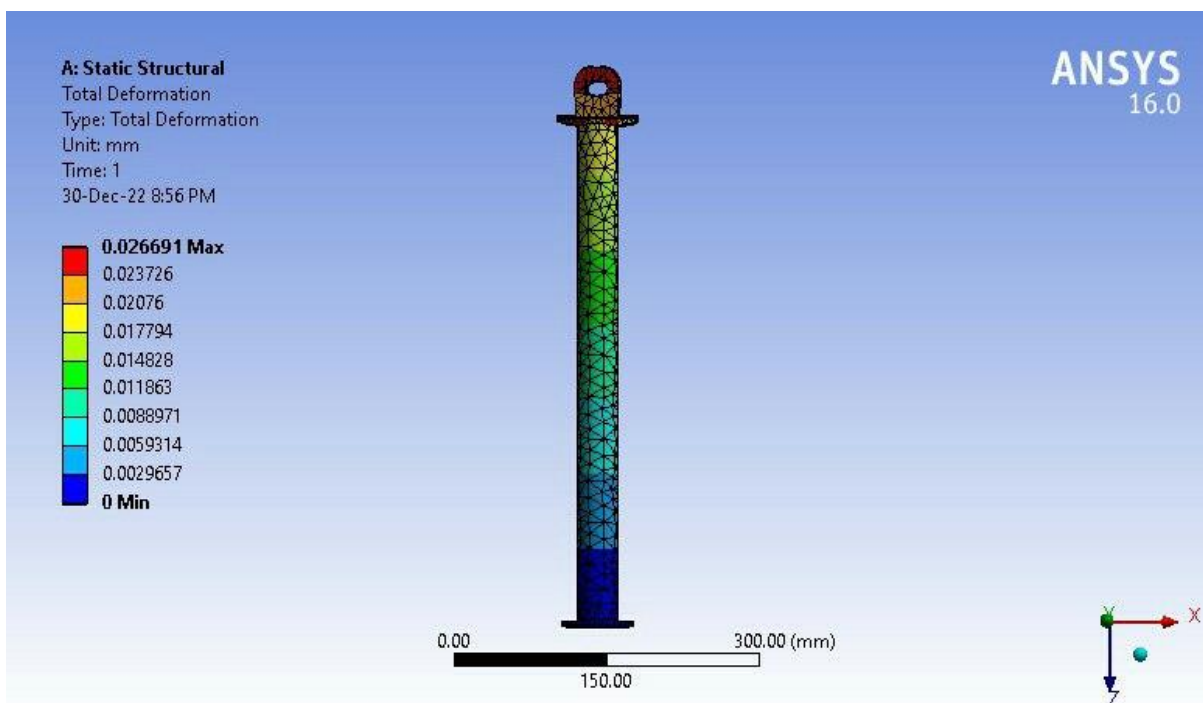


Figure III. 6. Analysis result of the piston road model with a mass of 10 kN

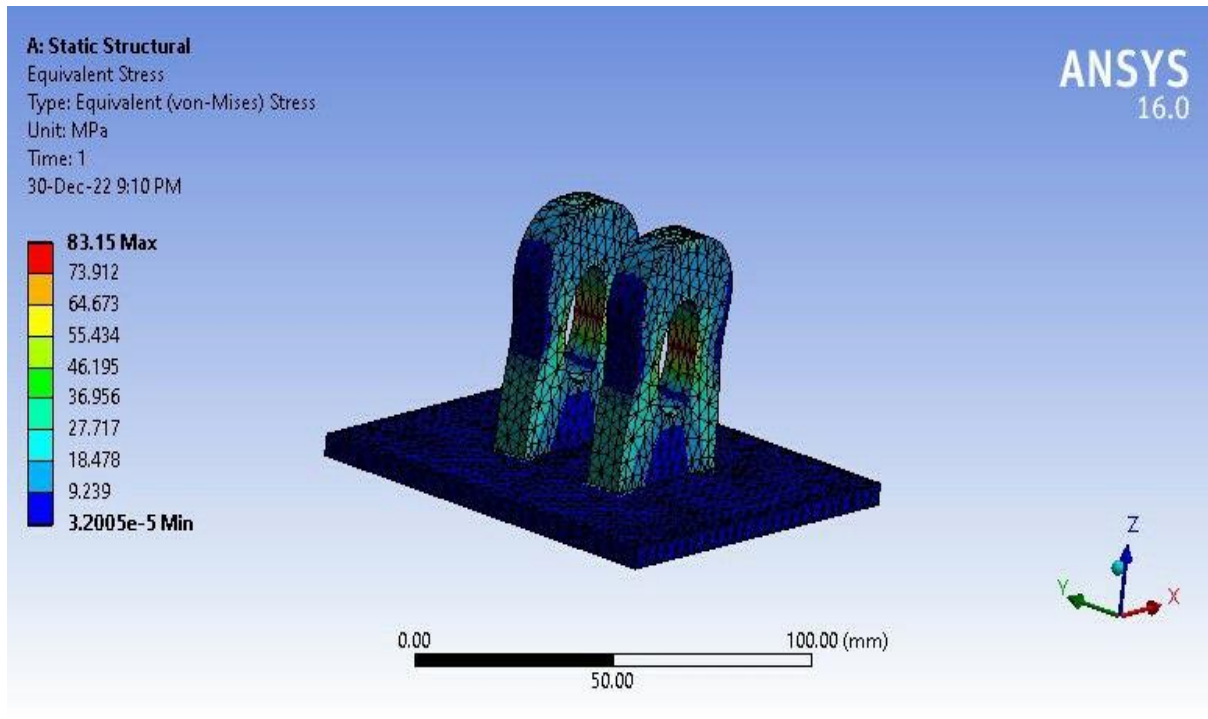


Figure III. 7. Analysis result of the flange model with a mass 10 kN

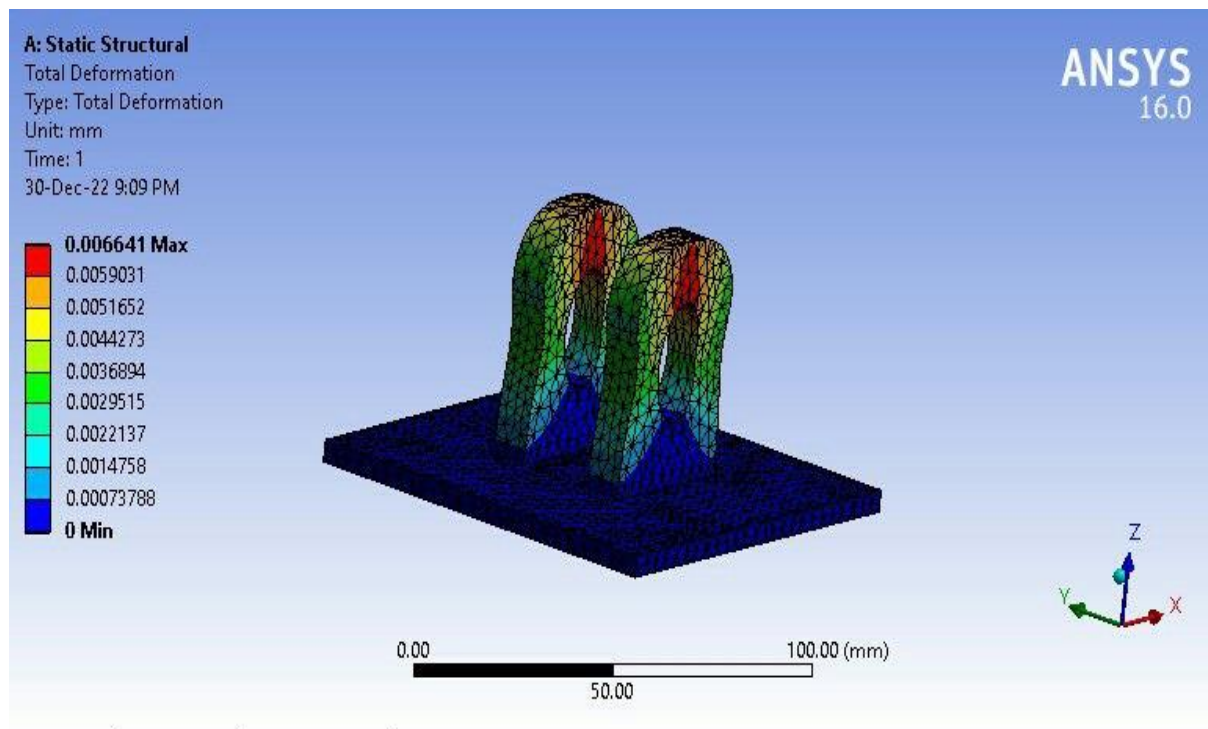


Figure III. 8. Analysis result of the flange model with a mass 10 kN

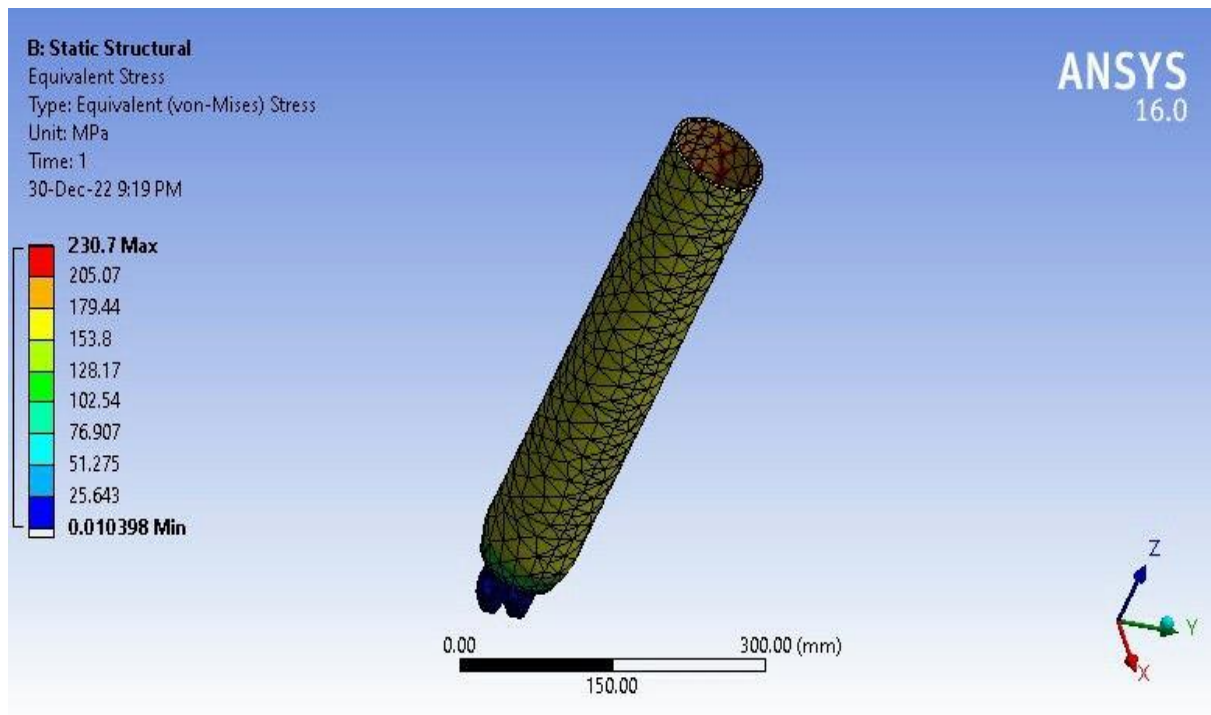


Figure III. 9. Analysis result of the cylinder model with a pressure of 250bar.

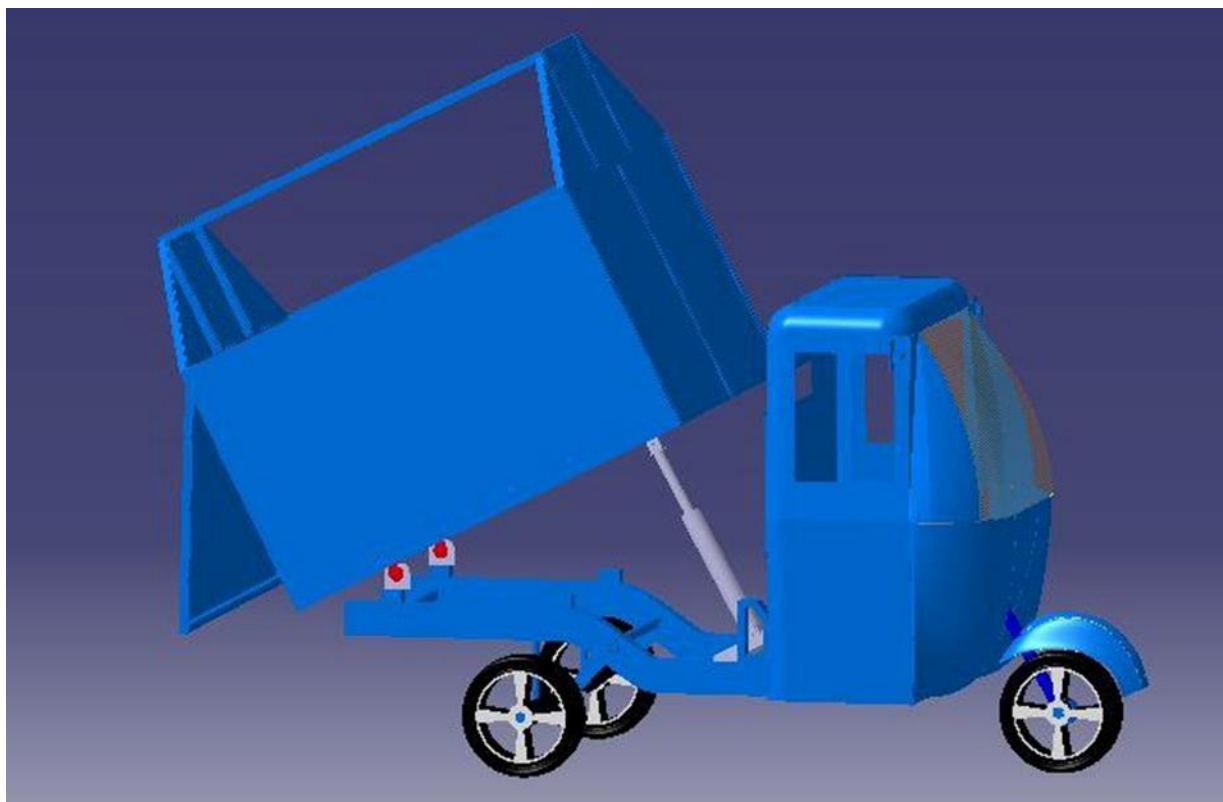


Figure III. 10. 3D maxima cargo Bajaj

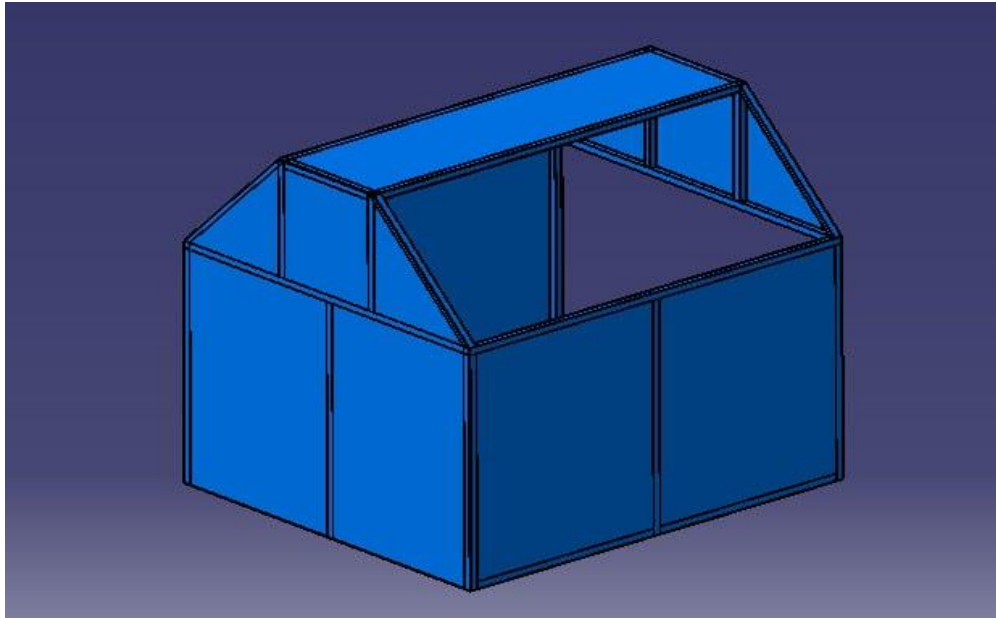


Figure III. 11. Body modification

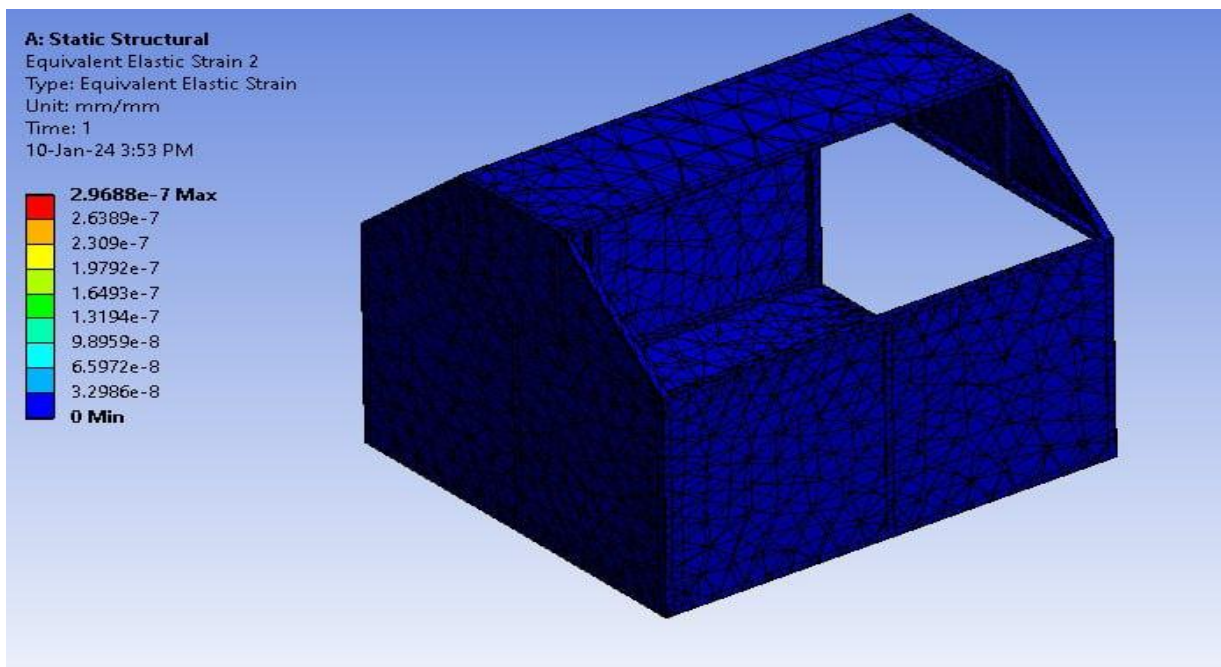


Figure III. 12. Equivalent elastic strain type of static structure

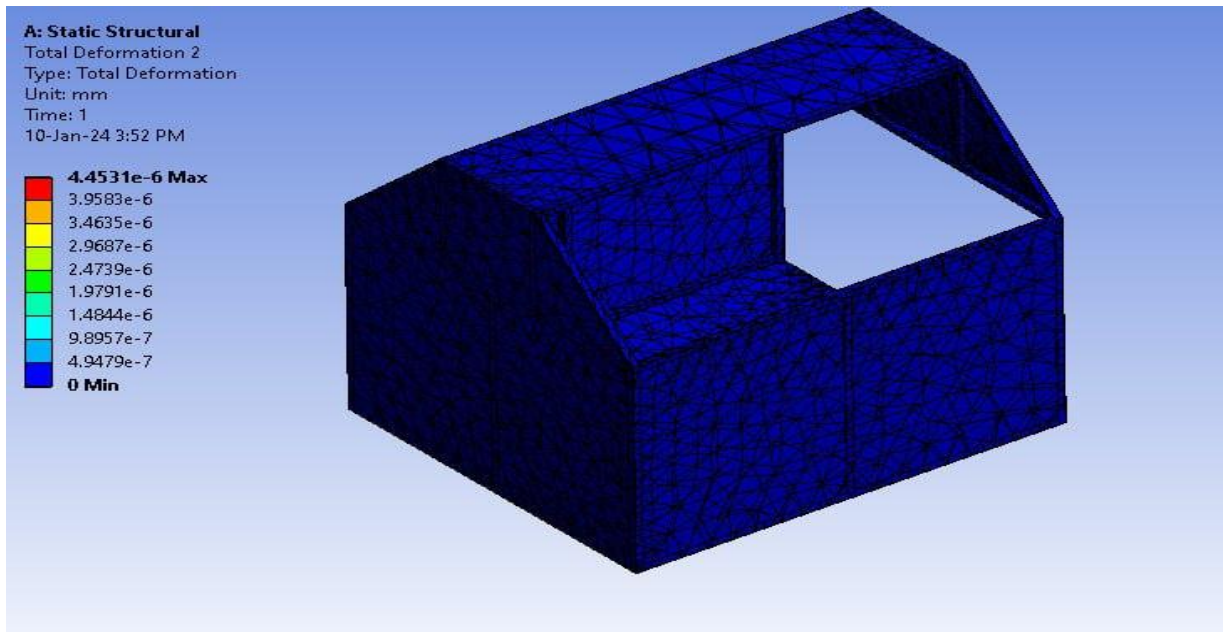


Figure III. 13. Total deformation type of static structure

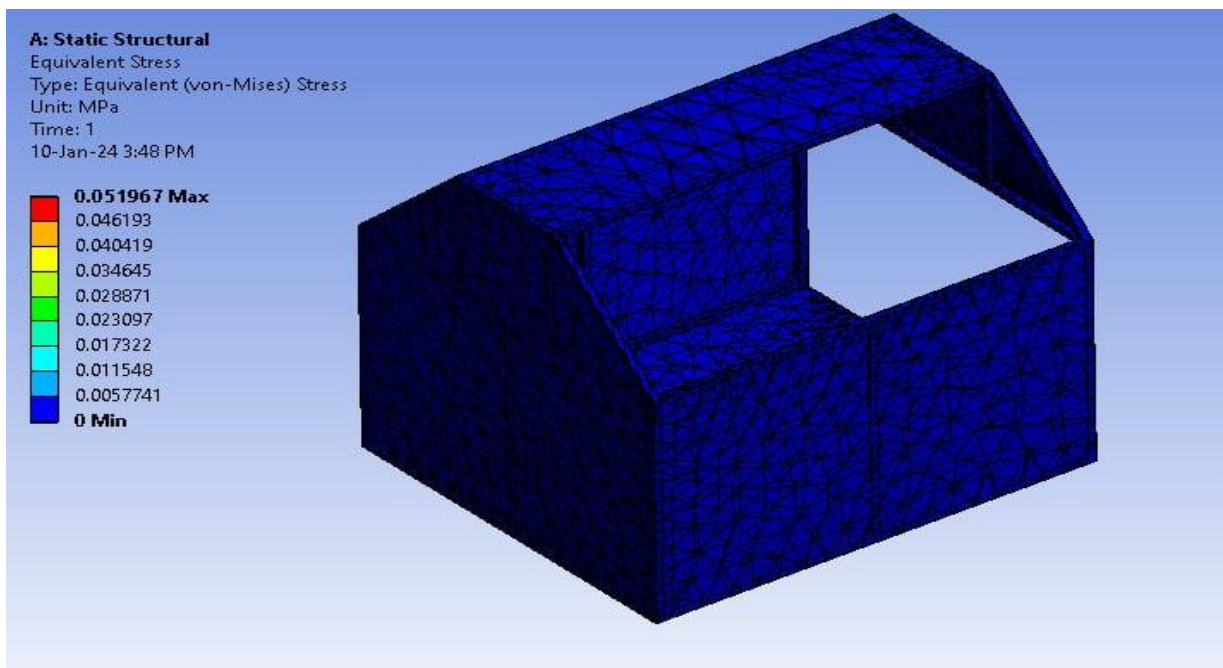


Figure III. 14. Equivalent (von-mises) stress type of static structure



Figure III. 15. Maxima cargo Bajaj (photo)