

ADAMA SCIENCE AND TECHNOLOGY UNIVERSITY

School of Mechanical, Chemical and Materials Engineering



**Structural and Modal Analysis of Shock Absorber of Four wheel
small vehicle**

*A thesis submitted in partial fulfillment of the requirements for the
award of*

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In

Automotive Engineering

By

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vehicle

By:-Kibrom Menasbo

APPROVAL BY THE EXAMINING BOARD

Chairman, department graduate committee

Advisor

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CANDIDATE’S DECLARATION

I hereby declare that the thesis work which is being presented in the paper entitled “**Structural and Modal Analysis of Shock Absorber of Four wheel small vehicle**” in partial fulfillment of the requirements for the degree of masters of science in Automotive Engineering is an authentic record of my own work carried out from March 2017 to September 2017 under the supervision of Mr. Mekonnen Liben, Assistant Professor , Department of Mechanical and Vehicle Engineering ,Adama Science and Technology University ,Adama, Ethiopia

The matter embodies in this paper has not been submitted by me for the award of any other degree or diploma. All relevant resources of information used in this paper have been duly acknowledged.

Name: Kibrom Menasbo Signature: _____ Date: _____

This is to certify that the above declaration made by candidate is correct.

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Mr. Mekonnen Liben, (Assistant Professor)	Date
(Advisor)	

ACKNOLEDGMENT

God has everything to do. All things were made through him, and nobody was made without him. He is always the sponsor of my existence and gave a full health through this thesis work.

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DEDICATION

My dearest Mam, who leads me through the valley of darkness

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ACRONYMS AND SYMBOLS

2D	Two dimensional
3D	Three dimensional
FEM/A	Finite element method/analysis
CAD/E	Computer aided design/engineering
D_o	Outer diameter of cylinder
D_i	Inner diameter of cylinder
T	Thickness of cylinder
σ	Maximum allowable stress
$P_{\max}$	Maximum pressure in cylinder
E	Energy per stroke
m	Sprung mass
g	Acceleration due to gravity
h	Damping distance
s	Damping height
DCF	Damping correction factor
ζ	Damping ratio
F_{gas}	Force due to gas pressure
P_{gas}	Pressure of gas
A_{rod}	Area of piston rod
K	Stiffness of spring
P	load on spring
δ	Deflection of spring

G	Shear modules
N_t	Number of turns of spring
ζ	Permissible shear stress
s_{ut}	Ultimate stress of spring material
k_w	Wahl factor
C	Spring index
d	wire diameter
D_{mean}	Mean diameter of spring
P	Pitch of spring
μ	Viscosity of damping fluid
D	Equivalent diameter of flow
V_{avg}	Average velocity of piston
P_1	Pressure above piston
P_2	Pressure below piston
ADAMS	Automated Dynamic Analysis of Mechanical Systems)

ABSTRACT

Safety and driving comfort for a car's driver are both dependent on the vehicle's suspension system. Safety refers to the vehicle's handling and braking capabilities. The comfort of the occupants of a car correlates to tiredness and ability to travel long distance with minimal annoyance. Shock absorbers are a critical part of a suspension system, connecting the vehicle to its wheels. Essentially shock absorbers are devices that smooth out an impulse experienced by a vehicle, and appropriately dissipate or absorb the kinetic energy. Suspension systems consist of springs and dampers, which tend to limit the performance of a system due to their physical constraints. Suspension systems, comprising of springs and dampers, are usually designed for passengers' safety, and do little to improve passenger comfort. To meet the current demands of high speed and safety there is a need to design and developed such a shock absorber which can sustain more and more vibrations and also improves the safety. This research explores the analysis of a shock absorber to providing by controlling the vertical oscillation of the vehicle to achieve and enhance occupant as well as driver safety. The study is conducted starting from design of different parts of shock absorbers, where the performance potential of different concepts in energy absorbing elements is investigated. The model of the existing and modified shock absorber is prepared using CATIA and SOLID WORKS software. In the final phase, the proposed shock absorber concept is implemented in a solid work for motion analysis. In this case before proceeding through the solid work analysis of the modified shock absorber models, the static structural and modal analysis of the shock absorber assembly is performed at a design load. The analysis of the existing model in ANSYS, obtained with different material are compared with those of the modified model to illustrate the deformation, stress and energy absorbing properties of the existing one and the proposed concept. The existing shock absorber and modified shock absorber models analysis results shows at their applying load 5542N and 6020N respectively, the total deformation and stress of existing shock absorber model is much greater than the modified shock absorber model total deformation and stress. Thus the passenger's safety, comfort, and for better driving characteristic of the vehicle can greatly be enhanced by using the modified shock absorber model.

Keywords: Springs, dampers, Shock absorber, Foam cell, ANSYS 15.0.

CHAPTER ONE

1 INTRODUCTION

1.1 Background of the study

The current world-wide production of vehicle dampers, or so-called shock absorbers, is difficult to estimate with accuracy, but is probably around 50–100 million units per annum with a retail value well in excess of one billion dollars per annum. A typical European country has a demand for over 5 million units per year on new cars and over 1 million replacement units, The US market is several times that. If all is well, these suspension dampers do their work quietly and without fuss. Like punctuation or acting, dampers are at their best when they are not noticed - drivers and passengers simply want the dampers to be trouble free. In contrast, for the designer they are a constant interest and challenge. For the suspension engineer there is some satisfaction in creating a good new damper for a racing car or rally car and perhaps making some contribution to competition success. Less exciting, but economically more important, there is also satisfaction in seeing everyday vehicles travelling safely with comfortable occupants at speeds that would, even on good roads, be quite impractical without an effective suspension system[1].

Shock absorbers are a critical part of a suspension system, connecting the vehicle to its wheels. The need for dampers arises because of the roll and pitches associated with vehicle maneuvering, and from the roughness of roads. Shock absorbers are devices that smooth out an impulse experienced by a vehicle, and appropriately dissipate or absorb the kinetic energy. For many years vehicle dynamic engineers have struggled to achieve a compromise between vehicle handling, ride comfort and stability. Every automotive shock absorber has two goals: passenger comfort and vehicle control. Comfort is provided by isolating the vehicles passenger from road disturbances like bumps or potholes. Control is achieved by keeping the car body from rolling and pitching excessively, and maintains good contact between the tire and the road.

Today's vehicle use hydraulic shock absorber and springs that are charged with the tasks of absorbing bumps, minimizing the automobiles body motion while acceleration, breaking and turning and keeping the tires in contact with the road surface ,typically these goals are somewhat at odds with each other. A typical vehicle suspension is made up of two components: a spring and a damper. The spring is chosen based solely on the weight of the vehicle, while the dumper is the component that defines the suspension placement on the compromise curve .depending on the type of vehicle, a damper is chosen to make the vehicle perform best in its application ideally, the damper should isolate passengers from low –frequency disturbance when the damping is high. However, high damping provides poor high frequency absorption conversely, when the damping is low, the damper offers sufficient high-frequency absorption, at the expense of low –frequency isolation the need to reduce the effects of this compromise has given rise to several new advancement in automotive shock absorber .When a vehicle is travelling on a level road and the wheels strike a bump, the spring is compressed quickly. The compressed spring will attempt to return to its normal loaded length and, in so doing, will rebound past its normal height, causing the body to be lifted. The weight of the vehicle will then push the spring down below its normal loaded height. This, in turn, causes the spring to rebound again. This bouncing process is repeated over and over, a little less each time, until the up-and-down movement finally stops. If bouncing is allowed to go uncontrolled, it will not only cause an uncomfortable ride but will make handling of the vehicle very difficult. A safe vehicle must be able to stop and maneuver over a wide range of road conditions. Good contact between the tires and the road will able to stop and maneuver quickly. The design of spring in suspension system is very important. In this thesis a shock absorber is designed and a 3D model which is created by using CATIA. The Structural, Dynamic and modal analysis are done on the shock absorber. The analysis considered the loads, vehicle weight, two person and 4 persons. Structural analysis is done to validate the strength and modal analysis to determine the displacements for different frequencies for number of modes [2].

1.1.1 Description

Pneumatic and hydraulic shock absorbers commonly take the form of a cylinder with a sliding Piston inside. The cylinder is filled with a fluid (such as hydraulic fluid) or air. This fluid-filled Piston/cylinder combination is a dashpot.

1.1.2 Explanation

The shock absorbers duty is to absorb or dissipate energy. One design consideration, when designing or choosing a shock absorber, is where that energy will go. In most dashpots, energy is converted to heat inside the viscous fluid. In hydraulic cylinders, the hydraulic fluid will heat up, while in air cylinders, the hot air is usually exhausted to the atmosphere. In other types of dashpots, such as electromagnetic ones, the dissipated energy can be stored and used later. In general terms, shock absorbers help cushion cars on uneven roads.

1.1.3 Applications

Shock absorbers are an important part of automobile and motorcycle suspensions, aircraft landing gear, and the supports for many industrial machines. Large shock absorbers have also been used in structural engineering to reduce the susceptibility of structures to earthquake damage and resonance. A transverse mounted shock absorber, called a yaw damper, helps keep railcars from swaying excessively from side to side and are important in passenger railroads, commuter rail and rapid transit systems because they prevent rail cars from damaging station platforms. The success of passive damping technologies in suppressing vibration amplitudes could be ascertained with the fact that it has a market size of around \$ 4.5 billion [47].



Figure 1-1: Mounting of shock absorber in TOYOTA HILUX

1.1.4 Vehicle suspension

In a vehicle, it reduces the effect of traveling over rough ground, leading to improved ride quality, and increase in comfort due to substantially reduced amplitude of disturbances. Without shock absorbers, the vehicle would have a bouncing ride, as energy is stored in the spring and then released to the vehicle, possibly exceeding the allowed range of suspension movement. Control of excessive suspension movement without shock absorption requires stiffer (higher rate) springs, which would in turn give a harsh ride. Shock absorbers allow the use of soft (lower rate) springs while controlling the rate of suspension movement in response to bumps. They also, along with hysteresis in the tire itself, damp the motion of the unsprung weight up and down on the springiness of the tire. Since the tire is not as soft as the springs, effective wheel bounce damping may require stiffer shocks than would be ideal for the vehicle motion alone. Spring-based shock absorbers commonly use coil springs or leaf springs, though torsion bars can be used in tensional shocks as well. Ideal springs alone, however, are not shock absorbers as springs only store and do not dissipate or absorb energy. Vehicles typically employ springs and torsion bars as well as hydraulic shock absorbers. In this combination, "shock absorber" is reserved specifically for the hydraulic piston that absorbs and dissipates vibration.

1.1.5 Structures

Applied to a structure such as a building or bridge it may be part of a seismic retrofit or as part of new, earthquake resistant construction. In this application it allows yet restrains motion and absorbs resonant energy, which can cause excessive motion and eventual structural failure.

1.2 Types of shock absorbers

There are several commonly-used approaches to shock absorption: Hysteresis of structural material, for example the compression of rubber disks, stretching of rubber bands and cords, bending of steel springs, or twisting of torsion bars. Hysteresis is the tendency for otherwise elastic materials to rebound with less force than was required to deform them. Le vehicles with no separate shock absorbers are damped, to some extent, by the hysteresis of their springs and frames. Dry friction as used in wheel brakes, by using disks (classically made of leather) at the pivot of a lever,

with friction forced by springs. Used in early automobiles such as the Ford Model T, up through some British cars of the 1940s. Although now considered obsolete, an advantage of this system is its mechanical simplicity; the degree of damping can be easily adjusted by tightening or loosening the screw clamping the disks, and it can be easily rebuilt with simple hand tools. A disadvantage is that the damping force tends not to increase with the speed of the vertical motion. Solid state, tapered chain shock absorbers, using one or more tapered, axial align meant(s) of granular spheres, typically made of metals such as national, in a casing. Fluid friction, for example the flow of fluid through a narrow orifice (hydraulics), constitutes the vast majority of automotive shock absorbers [48].

1.2.1 Shock Absorber types

There are a number of different methods of converting an impact /collision into relatively smooth cushioned contact [48].

- Metal Spring
- Rubber Buffer
- Hydraulic Dashpot
- Collapsing safety Shock Absorbers
- Pneumatic Cylinders
- Self-compensating Hydraulic

1.2.1.1 Metal springs

Simply locating metal springs to absorb the impact loads are a low cost method of reducing the collision speed and reducing the shock loading. They are able to operate in very arduous conditions under a wide range of temperatures. These devices have high stopping forces at end of stroke. Metal springs store energy rather than dissipating it. If metal sprint type shock absorbers are used then measures hold is provided to limit Oscillations. Metal springs are often used with viscous dampers. There are a number of different types of metal springs including helical springs, bevel washers (cone-springs), leaf springs, ring springs, mesh springs etc. Each spring type has its own operating characteristics.

1.2.1.2 Elastomeric shock observers

These are low cost options for reducing the collision speed and reducing the shock loading and providing system damping. They are conveniently molded to suitable shapes. These devices have high stopping forces at end of stroke with significant internal damping. Elastomeric dampers are very widely used because of the associated advantages of low cost and together with performance benefits. The inherent damping of elastomers is useful in preventing excessive vibration amplitude at resonance – much reduced compared to metal springs.

1.2.1.3 Hydraulic Dashpot

This type of shock absorber is based on a simple hydraulic cylinder. As the piston rod is moved hydraulic fluid is forced through an orifice which restricts flow and consequently provides a controlled resistance to movement of the piston rod. With only one metering orifice the moving load is abruptly slowed down at the start of the stroke. The braking force rises to a very high peak at the start of the stroke and then falls away rapidly. On completion of the stroke the system is stable – the energy being dissipated in the hydraulic fluid as heat. This type of shock absorbers is provided with springs sufficient to return the actuator to its initial position after the impacting load is removed.

1.2.1.4 Collapsing Safety Shock Absorbers

These are single use units which are generally specially designed for specific duties. They are designed such that at impact they collapse and the impact energy is absorbed as the materials distort in their inelastic/yield range. They therefore are more compact compared to devices based on deflections within their elastic range.

1.2.1.5 Air (Pneumatic) spring

These devices use air as the resilient medium. Air has a high energy storage capacity compared to metal or elastomer materials. For duties with high loads and deflections the air spring is generally far more compact than the equivalent metal or elastomer device. Due to the compressibility of air these have a sharply rising force characteristic towards the end of the stroke. The majority of the

energy is absorbed near the end of the stroke. Air springs require more maintenance than metal or elastomer based springs and the temperature range is restricted compared to metal springs.

1.3 Statement of the problem

Comfort is the most common thing associated with shock absorbers. Since the invention of the first car, the challenge of enhancing passengers' comfort was a major part of manufacturing process. That's why absorbers were invented. Their job, apart from providing vehicle stability, is to ensure the comfort of all passengers. They reduce the magnitude of vibratory motions due to various road imperfections to provide a smooth and comfortable ride. When a vehicle is travelling on a level road and the wheels strike a bump, the spring is compressed quickly. The compressed spring will attempt to return to its normal loaded length. It will rebound past its normal height, causing the body to be lift. The weight of the vehicle will then push the spring down below its normal load height. This causes the spring to rebound again. This bouncing process is repeated over and over, a little less each time, until the up and down movement finally stops. If bouncing is allowed to go uncontrolled, it will not only cause an uncomfortable ride but will make handling of the vehicle very difficult, get more body roll, bob up and down for a while it will just be annoying . According to Mullica R. [5], so to have better handling of the vehicle and maintain comfortable ride improving the design of shock absorber to dampen the vibration is necessary. The materials that used to design the shock absorber have proven to be the best material as per cost reduction, strength, durability as well as the stress and deformation is concerned then the existing one. This work is concerned with the investigation and design of a hydraulic mono tube shock absorber that will reduce the amount of jerk that is transmitted by the shock absorber to the upper mass in an automobile type suspension system [6]. According to Doug Couch man [49] the problems are pre-condition leakage in shock absorber for almost always due to one of two things: age and accidents.

- **Shock age:** Modern shock absorbers and struts are designed to last several years and over 50,000 miles, but eventually the seals do wear out and they begin to leak. Your owner's manual may specify a time or mileage for shock replacement, but this is a guideline, not an absolute: driving style, road conditions, and even how much dirt the shock encounters can have an effect.

- **Accidents:** Any accident that involves the suspension has the potential to damage the shock absorbers; a bent or dented shock almost always needs to be replaced. After major accident the repair shop will examine your shocks to see whether they need replacement, but it's important to understand that for this purpose, "accident" includes not just serious crashes but also anything that jars the suspension particularly hard, including hitting curbs, large rocks, and deep potholes, or even a rock being kicked up against the shock as you drive on a dirt road.

When any of these things goes wrong it's almost always necessary to replace the shocks, as they usually can't be repaired or simply refilled. Also, it's important to replace a failing shock as soon as possible, because a car with a bad shock absorber can become difficult to control in an emergency due to excessive wheel bounce.

Due to these existing problems that mentioned above, different type of shock absorber will be happened. so there is a need to improve good shock absorber to reduce body roll, bob up and down caused due to wheels of the vehicle hit a bump.

1.4 Objectives

1.4.1 General Objective

The general objective of this thesis work is to design and analyze the exiting shock absorber and modified shock absorber of four wheel small vehicles.

1.4.2 Specific Objectives

- ✓ To design and shock absorber components such as
 - Cylinder design
 - Piston design
 - Piston rod design
 - Spring deign
- ✓ To study the design details of the existing shock absorber

- ✓ To estimate the loads on shock absorber
- ✓ To analyze the model of shock absorber
- ✓ To compare the performance of existing one with improved one.
- ✓ Motion analysis using SOLIDWORK software
- ✓ To construct prototype of improved one.

1.5 Significance of the study

This work helps in improving the crashworthiness of the vehicle during vehicles wheel striking a bump. In a vehicle, it reduces the effect of traveling over rough ground, leading to improved ride quality, and increase in comfort due to substantially reduced amplitude of disturbances. The drivers and passengers feel safe and enjoy the tension free driving or journey. It minimizes the injury of passenger and driver by reducing the forward velocity of the passenger as well as the driver. It helps the future researchers in their work.

Beneficiaries: Automotive industries, researchers in the area, society and the country as a whole

1.6 Delimitation (scope) study

The scope of study for this project includes:-

- ✓ Applying structural and modal analysis on the shock absorber.
- ✓ Application of Finite Element Analysis (FEA).
- ✓ Dynamic motion analysis.

1.7 Limitation

Shortage of time limited the work towards the modified shock absorber only analysis in solid work for motion analysis and model small vehicle for analysis only. None availability of high capacity computer is another factor which limited the work. Analysis using ANSYS and SOLID WORK software with small size element and the fine adjustment needs high capacity computer .due to non-availability of computer with high processing capacity, the analysis is limited to big element size 0.001mm with fine adjustment.

1.8 Organization of the study

This thesis consists of five chapters. The present chapter provides a general introductory part together with statement of the problem, objective of the study, delimitation or scope and significance of the study. The second chapter provides literature review of the most recent research works that are related to the field of study. Chapter three, materials and methods, shock absorber components are designed and modeled using CATIA. In this chapter the general idea on the ANSYS software package is outlined. Chapter 4 contains the result and discussion portion which shows the ANSYS software results on the deformation and stress in static structural and modal analysis and Motion Analysis in Solid works of shock absorber. Finally conclusion, the most significant suggestion and the future works are briefly described in chapter 5.

CHAPTER TWO

2 LITERATURE REVIEW

The earliest paper appears to be that by Hadley (1928), which deals with mechanical friction dampers, seeking to obtain a suitable characteristic by position dependence [7].

Weaver (1929) investigated damper characteristics by applying them to a vibrating system, actually a 272 kg mass on a 31 k N/m spring giving 1.71 Hz undamped natural frequency, with initial deflections of typically 76 mm, and producing free vibration $F(X)$ plots, which are roughly spiral for less than critical damping [8].

James and Ullery (1932) discuss the problems of Coulomb friction against the advantages of hydraulic damping with a suitable force–speed relationship [9].

Kindle (1933) proposed to regulate the damper by a valve controlled by body inertia, so that wheel only movements are free, but the body motion is damped [10].

Schilling and Fuchs (1941) show force–speed diagrams for year 1940 passenger car dampers, some of which are similar to modern dampers, showing a predominance of rebound force over bump [11].

Connor (1946) distinguishes various types of damping effect (Coulomb, real dry friction, viscous, degenerate viscous, ‘hydraulic’ damping proportional to V^2 , and solid material hysteresis). Typical damping ratios are stated to be 0.15–0.20, and the temperature rise in a telescopic damper is stated to be about $50\text{ }^{\circ}\text{C}$ on extremely rough roads, although only about $20\text{ }^{\circ}\text{C}$ for rotary and cam types because of conductivity into the vehicle body [12].

Brown (1948), Tatarinov (1948, 1949) and Brown (1950), the related area of buffer design is considered for buffering, the resisting force is commonly made a function of position. This is not normally done for car dampers, although it has been used on motorcycle front forks, where it gives a bump stop effect, but is regarded as superior to an elastic bump stop because of reduced spring back [13].

Kinchin and Stock (1951/1952), in a comprehensive paper, discuss various details, such as the physical layout of various hydraulic types, fluids, testing, operating problems such as aeration, Fade, noise problems, flow resistances, and force–displacement $F(X)$ diagram (work diagram) Shapes [14].

Peterson (1953) describes the introduction of a damper designed to have a linear characteristic (in contrast to many of the previous ones which were almost Coulomb in character, as may also be seen in some of the characteristics shown by Schilling and Fuchs (1941). Prototypes had tubes intended to give a simple viscous pressure drop, but actually giving force proportional to speed to the power 1.25. More conventional drilled and coined holes were used for production. The higher forces obtained at high speeds were considered a substantial improvement for rough roads [15].

Cline (1958) discusses various aspects of dampers in general terms, giving some example $F(X)$ curves, with brief comments on the effect of valve changes. He also shows a photograph of the then new Gabriel dampers with electrical remote adjustment [16].

Hoffman (1958) was amongst the first to do analytical work on the damper. He shows various experimental $F(V)$ and $F(X)$ curves for different types of damper, and used an analogue computer to study vehicle ride for various dampers [17].

Jackson (1959) discusses the basic principles of telescopic dampers and their manufacture, and the effect of valving, stating that control of unsprung mass vibrations had become the greater problem once independent suspension was introduced. [18].

Eberan-Eberhorst and Willich (1962) show experimental $F(X)$ curves for various strokes, and $F(X)$ and $F(V)$ curves with lag and hysteresis. The experimentally observed temperature effect on force was about $-0.1\%/K$ for compression and $-0.3\%/K$ for extension. A Plexiglas damper was built, allowing cavitation to be observed directly [19].

Polak and Burton (1971) discuss damper construction in general terms, giving attention to possible designs of amplitude sensitive dampers, to a multi-frequency ‘seismically’ actuated damper (mounted on the wheel only, not to the body), and propose a configuration with a conventional damper coupled between suspension and engine [20].

Woßner and Dantele (1971) compare the pressurized single-tube type (with and without a free separator piston) with the unpressurised dual-tube type, and give experimental results on the cooling effect of air streams [21].

Mitschke and Riesenberger (1972) discuss damper temperature rise on various types of road, and the consequences for fluid viscosity and damper force [22].

Mayne (1973) gives an analysis of the effect of liquid and mechanical compliance on buffer performance [23].

Jennings (1974) gives 21 F(X) loops for commercially available motorcycle front forks and rear dampers, which prove to be characterized by extreme compression/extension asymmetry [24].

Cline (1974) gives a simple review of the application of hydraulic dampers to recreational vehicles, considering briefly how to achieve some of the particular characteristics required. Schubert and Racca (1974) describe an unconventional 'elastomeric-pneumatic isolator with orifice type relaxation damping', proposing application to motor vehicles [25].

Dalibert (1977) considers the effect of some oil properties on damper performance. The sensitivity of peak force to temperature was found to be about $-0.3\% / ^\circ\text{C}$, giving a reduction of 35% at 130°C , which is stated to be the approximate maximum temperature encountered with hard driving on bad roads. Maximum safe viscosity is stated to be 4000cs at the lowest temperature, and a minimum of 4 to 7cs at 100°C , depending on design. Noise problems are considered in some detail [26].

Segel and Lang (1981) report a detailed investigation (additional information being given in Lang's PhD thesis (1977), using an 82 parameter analogue computer model that gave quite good agreement with the experimental data obtained. Compressibility of the liquid, plus a slight additional effect from cylinder compliance, was found to give hysteresis in the F (V) curve at higher frequencies (above 1 Hz) [27].

Arndt et al. (1981) consider seal design with tests of friction especially in the context of smaller, lighter vehicles where the problem seems more critical, advocating a lip design with a lubrication groove, allowing improved ride comfort and reduced pressurization [28].

Ohtake et al. (1981) also consider seal design, with tests of friction and durability, and analysis of relevant parameters, considering variation of optimum design with details of the application [29].

Fukushima et al. (1983) advocate stroke-sensitive damping, and present the arguments in its favor, including analysis of ride and handling motions. Fukushima et al. (1984) reiterate the points on stroke-sensitive damping, and consider a vortex valve type which has a resistance that depends on stroke as well as speed, being greater for larger stroke [30].

Sugasawa et al. (1985) studied theoretically the optimum damping for ride and handling independently for a two-degrees-of-freedom system, and tested an automatically adjusting damper system using sensors for control inputs (accelerator, breaks, steering) and an ultrasonic ride height measurement to appraise the road quality [31].

Browne and Hamburg (1986) measured damper temperatures, and also the simultaneous forces and velocities of dampers on the vehicle, to obtain the energy dissipation rates. For passenger cars on normal roadways this was found to be 3–60 W, and about 12W on average [32].

Karadayi and Masada (1986) consider factors such as directional asymmetry, dry friction, hysteresis, compressibility and backlash in a nonlinear model, aiming to create a simple damper model (i.e. fast computing) suitable for use in vehicle simulations. The total nonlinear compressibility is treated a piece wise linear, giving in effect a backlash plus a series stiffness, with Coulomb friction and asymmetric linear damping. [33].

Hall and Gill (1986) describe a CSMP digital simulation of a dual-tube damper, including the effect of valve mass and valve damping. The foot compression valve (compression control) is a side exit spool valve, giving a fluid momentum force that gives Coulomb friction between the spool and its guide—another possible source of $F(V)$ curve hysteresis. Gas compressibility is dealt with by adopting a high effective vapor pressure of 41–85kPa. In the simulation, a time step not exceeding 10ms was necessary to avoid instability, corresponding to 0.02° of phase angle [34].

Hennecke et al. (1990a, b) describe an in-production adaptive damping system, with three states independently for each axle according to road conditions and driver. This system uses body sensors for vertical acceleration above the axles, steering angle, and longitudinal speed and longitudinal

acceleration (rather than brake line pressure and engine output), with electronic control of solenoid valves [35].

Fan and Anderson (1990) tested a bus damper complete with its rubber mounting bushes, naturally obtaining a large hysteresis in the $F(V)$ curve. Modelling equations are presented, with the bushes represented as effective compressibility, and good correlation with experiments is shown, with frequency-dependent hysteresis [36].

Rakheja et al. (1990) studied a 'sequential' damper similar to a conventional positive damper with multi-stage asymmetrical valving proposed to be mounted externally to facilitate adjustment, concluding that it offered significant possible improvements in ride over a conventional damper [37].

Lemme (1990) considers the advantages of hydraulic control for variable dampers over electrical control, considering in detail the design of such a damper, with the idea that action of any one damper could produce a pressure to control all four damper settings, without other sensors [38].

Kumagai et al. (1991) studied the internal Coulomb friction of strut bearings relating to the transmission of NVH (noise, vibration and harshness) [39].

Shiozaki et al. (1991) propose a variable damper with piezoelectric sensor and actuator built into the damper rod, with rapid response (3ms) and high actuation force for an axial valve, favoring a 'normally hard' setting with switching to soft when appropriate, e.g. hitting a bump. The short piezoelectric actuator motion is amplified by stacking the elements, and then amplifying the total motion with an inverted hydraulic jack method to give 2 mm of motion at the valve [40].

Tamura et al. (1992) consider the development of materials for the rod-guide bush of strut type suspensions, which carries a large side load with unreliable lubrication, but needs to have low friction and long life, and is subject to other problems such as cavitation erosion. Various PTFE/Sn/Pb/Cu/PbF materials have been used [41].

Petek (1992) describes the design principles of and tests on an experimental design of variable damper using ER (electro rheological) liquid, with durability tests and a retrofit vehicle road test. The ER effect is claimed to be fast in action (time constant about 5ms), with good control power

input efficiency (about 4 W), and giving lower cost and better reliability than mechanical valves, although the desired operating temperature range of -40 to +120 deg C may be a problem[42].

Besinger et al. (1995) describe a seven parameter damper model, particularly developed for use with studies of heavy vehicles. The model is compared with test damper data, and the effect of variation of the parameters on vehicle ride is investigated by simulation [43].

Yamauchi et al. (2003) investigated a noisy strut suspension vibration arising from rod bending stimulated by damper piston friction against the tube, successfully obtaining a simple design criterion and testing a double-piston solution giving improved alignment [44].

M.Lavanya, “Comparison between two materials to verify the best material for spring in two wheelers mono Shock absorber”. Structural analysis and modal analysis are done on the shock absorber by varying material for spring, Spring Steel, Phosphorus bronze and Beryllium Copper. [54].

Siddhant. R. Patil, 2P. S. Talmale^{1, 2}, “Analysis of Shock Absorber Coil Spring” Using Fem Approach. A 3D model of coil spring is created using Cero software. Structural analysis and modal analysis are done on the shock absorber by varying material for spring, structural Steel, Brass and Berylliums copper [55].

R.Kannan^{*1}, C.Rajaganapathy^{2*1}, “Finite Element Analysis of Shock Absorber for Spring Steel and Carbon fiber”. A shock absorber is designed and a 3D model is created using Solid works. Structural analyses are done on the shock absorber for various material, Spring Steel and Carbon fiber. The analysis is done by various factors loads, weight, for single person and double person. Modeling is done in SOLID WORKS and analysis is done in ANSYS [56].

Devdutt¹and M.L. Aggarwal, “Comparative analysis of passenger ride comfort” using various semi active suspension alternatives in this paper, different semi-active control strategies for non-linear quarter car model equipped with controllable magneto-rheological (MR) shock absorbers and fuzzy logic controller are compared. Polynomial model is selected to characterize the experimental results of MR shock absorber. A combination of Forward Fuzzy Logic Controller (FFLC) and Inverse Fuzzy Logic Controller (IFLC) is designed for proper working of MR shock

absorber i.e. generation and supply of control current to MR shock absorber, which provides damping force to suspension system for vibration control purpose. Simulink responses of four different cases are evaluated for passenger ride comfort analysis taking uncontrolled, primary suspension controlled, secondary suspension controlled and fully controlled quarter car models. The obtained results in graphical and mathematical form demonstrate that the fully controlled quarter car system provides excellent performance in suppression of passenger seat vibrations compared to control strategies while the vehicle travels over the sinusoidal type of input road profile [57].

Sri L. Satya Bhaskar, Kum. G.N.Tirumaleswari “Design and analysis of a shock absorber a shock absorber is designed and a 3D model is created using Pro/Engineer”. The model is also changed by changing the thickness of the spring. Structural analysis and modal analysis are done on the shock absorber by varying material for spring, Spring Steel and Beryllium Copper [58].

Mohdsufian bin jamaludin “There are some problem happen at the vehicle when ride at the bumping road condition”. One of the problems is that the vehicle bounce continuously more than one times and it is called as bouncing problems. Study the causes of the bouncing problem at the vehicle and to compare the failure parts performance in terms of its yield strength between steel and beryllium copper. Experimental of suspension systems and bouncing problem in vehicle, apply structural analysis and model analysis on the shock absorber and application of Finite Element Analysis (FEA [59].

Achyut P. Banginwar ,Nitin D. Bhusale, Kautuk V. Totawar, “Structural analysis and modal analysis are done on the shock absorber by varying material for spring”, Spring Steel and Phosphor Bronze . Structural analysis is done to validate the strength and modal analysis is done to determine the displacements for different frequencies for number of modes [60].

Davis Jose, Anthony Tony, “Structural analysis in ANSYS”. ANSYS is general-purpose finite element analysis (FEA) software package. Finite Element Analysis is a numerical method of deconstructing a complex system into very small pieces (of user designated size) called elements. Analysis is done on the shock absorber by varying material for spring: Spring Steel and Beryllium [61].

Rahul Tekade¹, Chinmay Patil²M.E, The different stress and deflection values in shock absorber components have been obtained using FEA tools and compared with analytical solutions. Percentage error is calculated and it is found that percentage error is less than 15%. Various stress results are below allowable limits of material. Error in FE and analytical results occurs because of various reasons such as assumptions in analytical formulation, approximations in FE formulations, choice of element in FE analysis, etc. In this project a shock absorber will be designed and a 3D model which will be created is by using Pro/Engineer. The model is also changed by changing the thickness of the spring. Structural analysis and modal analysis are done on the shock absorber by varying material for spring. The analysis will consider the loads, bike weight, single person and 2 persons. Structural analysis will be done to validate the strength and modal analysis to determine the displacements for different frequencies for number of modes. Comparison will be done for two materials to verify best material for spring in Shock absorber [62].

Karthik A. S. Manojkumar M.Hanumanalli “shock absorber is designed to automobile of capacity 150cc and a 3D model is created using NX UNIGRAPHICS”. Structural analysis is carried out using ANSYS 14.5 on the shock absorber by varying material for spring, Structural Steel, Titanium alloy, Copper alloy, Aluminum alloy [63].

Chinna Mahammad bhasha, Vijay rami reddy, Rajnaveen, “a suspension system is designed and a 3D model is created using CATIA V5 R21”. The model is also changed by changing the thickness of the spring. Structural analysis and modal analysis are done on the shock absorber by varying different spring materials. Spring materials are Spring Steel, Phosphor bronze, Beryllium Copper and Titanium alloy [64].

Abhishek Dixit¹, Sachin Bhatia², Nitin Srivastava “Varying different spring materials like steel, titanium alloy, copper alloy and Phosphor” [65].

2.1 Types of shock absorber

 Twin tube shock absorber

 Mono tube shock absorber

1. Twin tube shock absorber

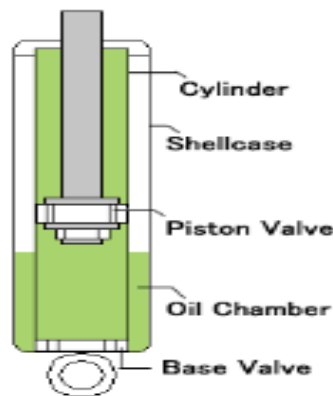


Figure 2-1: Twin tube shock absorber

Twin tube shock absorber use an inner and outer tube which separates the oil and gas inside the damper. As you see from Figure 2-1, the smaller inner tube houses the piston shaft assembly, base valve, and oil. The outer tube contains both nitrogen gas and the damper oil. Twin tube shock absorbers are the most commonly used type of shock absorber by OEM and aftermarket manufacturers as they are the cheapest shock absorber to make. However, twin tube dampers do not perform as well as mono tube shock absorbers as the oil heats up and destabilizes under extreme usage [45].

2. Mono tube shock absorber

Mono tube shock absorber uses a single outer tube. As you see from Figure 2-1; the oil and nitrogen gas inside are separated by a free piston. Mono tube shock absorbers use much higher gas pressure than twin tube shock absorbers to better stabilize the oil inside under extreme usage.

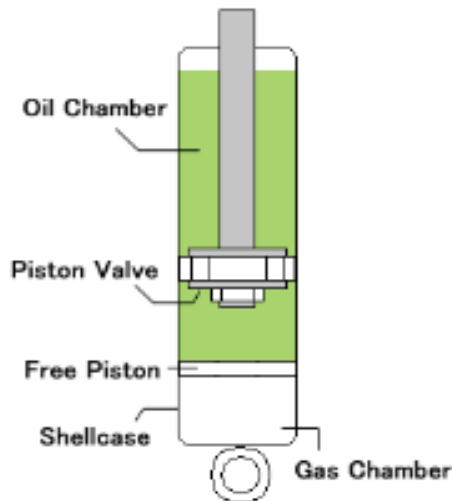


Figure 2-2: Mono tube shock absorber

The advantages of the mono tube design are larger internal parts, which mean greater damping force, increase oil capacity, improved heat dissipation, and the ability to function when inverted. Mono tube shock absorber are found in some OEM vehicle application, mainly higher end performance vehicle such as EVO MR, WRXSTI, Porsche, etc.

2.1.1 Study of shock absorber:

Selecting a right shock absorber is crucial part of designing a suspension system .there are two types of dampers used in mono suspension system ,i.e. twin tube and mono tube shock absorber. In FZ mono suspension system, twin tube damper is used .but in our project we designed mono tube shock absorber. Both the dampers were studied in detail which included its cost action and working .also its advantages and dis advantages were studied [46].

2.1.2 Advantage of mono tube shock absorber over twin tube shock absorber;

- Very wide damping rates can be achieved through bigger diameter pistons and shims design.
- Plush feel and full control over all piston speeds through higher gas and damping rates.

- Cools faster and more reliable than the single precision tube shock body.
- Bigger diameter piston rods can be used because of bigger internal chambers with sufficient oil.
- Can withstand higher pressure and temperature without aeration or foaming.

2.2 Working inside a damper:

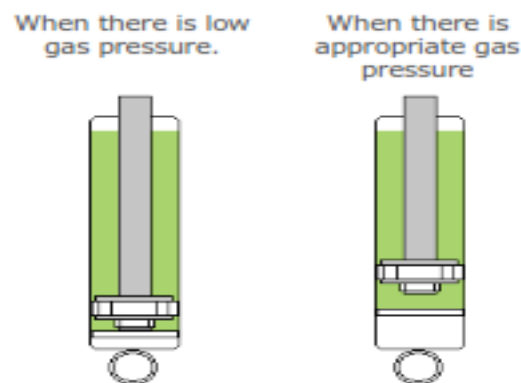


Figure 2-3: working inside a damper

Dampers also known shock absorbers works on the principle of fluid displacement on both the compression and extension cycle. As you see from Figure 2-3 ;The compression cycle controls the motion of a vehicles unsprung weight, while extension controls the heavier sprung weight .there are two cycles in which shock absorber works :

1. Compression

In the compression cycle the piston moves downward and compresses the hydraulic fluid in the chamber which is situated below the piston. In this cycle or downward movement, the fluid flows to upper chamber from down chamber through piston .some of the fluid also flows into reserve tube through the compression valve. Flow is controlled by valves in the piston and in the compression valve.

2. Extension

In the extension cycle the piston moves upwards toward the top of the pressure tube. The upward movement results in the compressing of the fluid in the chamber lying above the piston. The extension cycle generally provide more resistance than compression cycle [47].

2.3 Design Considerations

To form accurate conclusions as to the feasibility of the production of a customized set of dampers, certain design issues must first be taken into account. The factors which require consideration include manufacturability, cost, durability, heat dissipation, assembly and disassembly procedures and sealing [48].

2.3.1 Manufacturability

For the design of the new damper to be feasible, the design must be such that it can be manufactured, preferably in house at the university. As expressed previously, dampers require exacting tolerances to be adhered to if quality items are to be produced. The mechanical engineering workshop has the ability to CNC machine parts to excellent accuracy, such that I believe it would be possible to manufacture a set of dampers with the current tooling.

2.3.2 Cost

The overall cost of the dampers can be reduced if careful consideration is given to the component designs. One area where potential savings exist over purchased dampers is in assembly, with students being able to assemble to units when the components have been manufactured. An actual costing analysis of the damper production will be performed after the design has been presented.

2.3.3 Durability

Dampers need to be designed with durability in mind as they form the compliant link between the suspension and the chassis. As dampers are usually one of the most expensive items on the vehicle, it is beneficial to be able to re-use them. To be able to reuse the dampers, they should be designed such that major components do not wear to the point where replacement is necessary. This may

mean increasing the weight of some components to extend their fatigue life and exerting higher tolerances on machined parts, both of which increase the cost of the damper.

2.3.4 Heat Dissipation

Dampers produce a resistive force by passing oil through narrow passages. As time passes, frictional forces within the fluid and damper mechanisms generate heat which raises the temperature of the oil. Short term temperature variations will affect the viscosity of the damper oil, in some cases drastically altering the performance of the damper. Long term thermal cycling of oil eventually degrades its performance as its chemical properties change, thus good heat dissipation prolongs the life of the damper, requiring less frequent maintenance. Heat dissipation away from dampers is usually left to the vehicle designer, who must provide adequate airflow around the unit.

2.3.5 Assembly / Disassembly Considerations

As the damper consists of many smaller components, due consideration must be given as to how the damper is going to be assembled or disassembled. Most components are circular by nature and hence threads are prolific. Accessing these threads, by virtue of being able to apply enough torque to tighten or loosen them, must be considered.

2.3.6 Sealing

Dampers generate resistive forces by generate large internal pressures. To contain the contents of the damper under these pressures, adequate sealing must be provided. Static seals usually consist of rubber O-rings fitting into machined grooves with specific dimensions as to provide sufficient ‘squish’ to form a seal. Another type of seal often found in dampers is the sliding seal. Sliding seals are used around the piston, the main shaft and possibly in the external reservoir. These sliding seals usually preform dual functions, providing both a sealing surface and axial support for the particular component.

2.3.7 Suspension troubleshooting chart

In this table the major defects that occur in the suspension system of the automobile will be discussed [66].

Table 2-1: Suspension troubleshooting chart

Problem	Causes	Remedies
• Rough ride	1. The leaves of the spring may be rusted resulting in excessive friction 2. In the case of torsion bars, they may have not been adjusted properly.	1. Immediate lubrication of the springs to reduce friction. 2. They should be adjusted properly.
• Vehicle drag	1. The spring on the sagging side may have broken or become weak due to constant use.	1. Replace 2. They should be adjusted properly
• Vehicle bouncing after crossing bump	1. Worn out shock absorber 2. Damaged or slipped leaf spring	1. Replace 2. Spring should be checked, repaired or replaced.
• Knocking during crossing of bump	1. The shock absorber or strut may have worn out 2. Ball joint may be loose or worn out	1. Shock absorber or strut bearing have to be replaced 2. Ball joints have to be checked and serviced or replaced as required.
• Noises	1. U- bolts may be loose	1. Tighten wherever necessary

	2. There may be side play in shackles 3. The shackle pins and bushes may be loose 4. There may be some defect in the shock absorber	2. Lubricate
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CHAPTER THREE

3 METHODS AND MATERIALS

3.1 Methods

3.1.1 Hydraulic cylinder

Hydraulic cylinder is most important part of hydraulic press. It develops the necessary force required to carry out a pressing operation.as cylinder is one of the most important part of a press. It is designed by CATIA software the steps to draw this part is shown in the flow chart of Figure 3.2the detail drawing of the part is attached in the appendix- I.

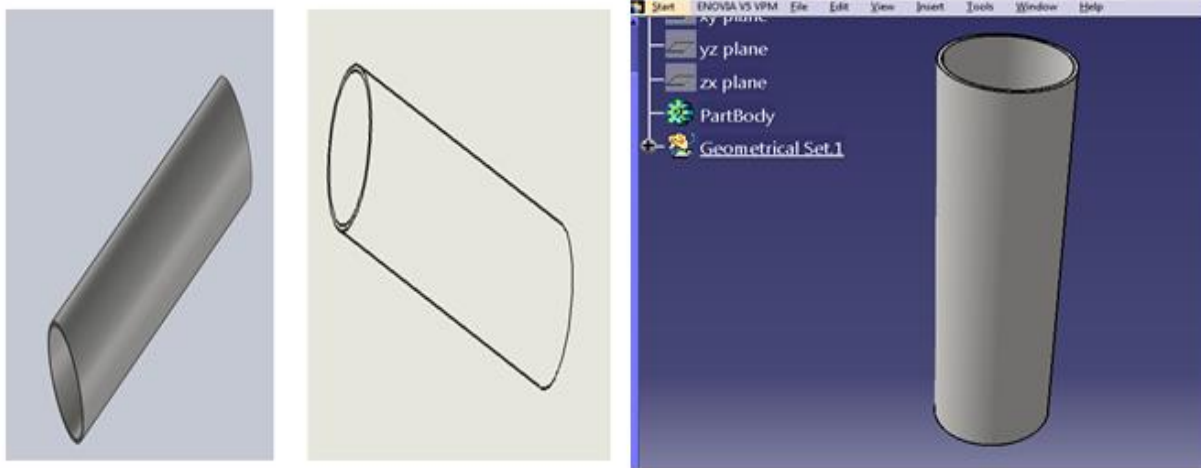


Figure 3-1: Cylinder 2D and 3D model

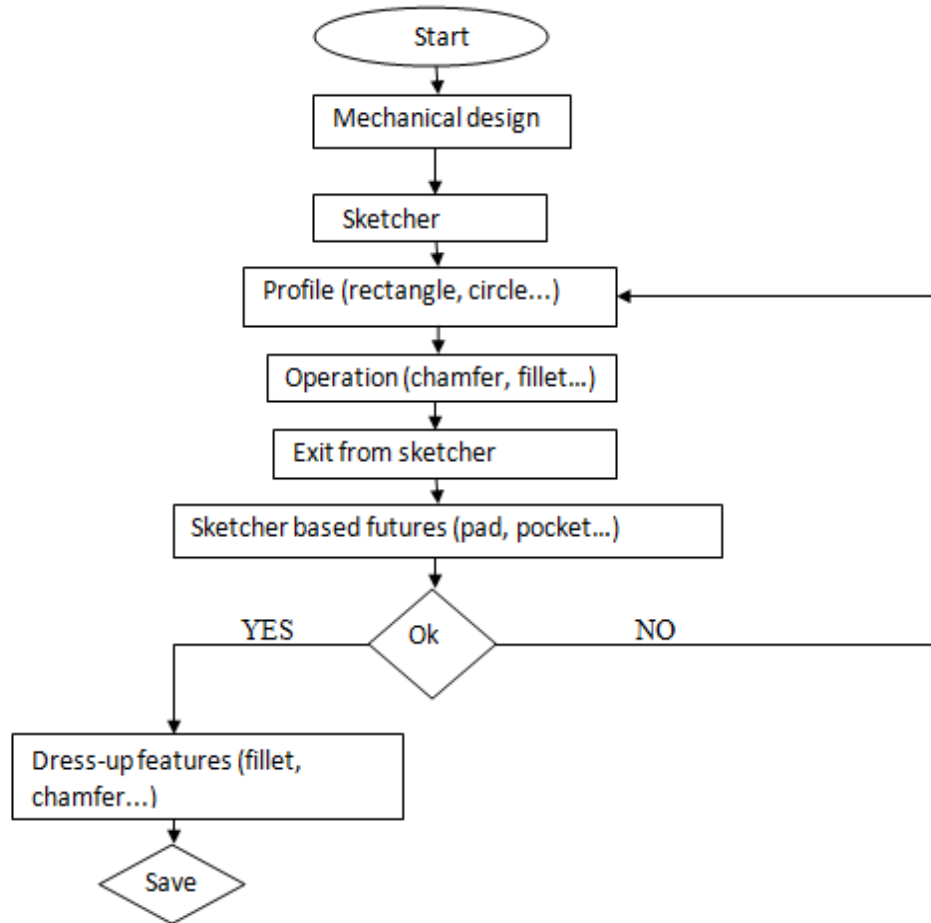


Figure 3-2: Flow chart that shows the general steps to draw cylinder, piston

3.2 Design of cylinder

For finding out inner diameter of cylinder

- Material structural steel [ductile]
- Tensile stress of structural steel: 215000 kPa
- Poisson ratio: 0.29
- Maximum pressure inside cylinder: 2100 kPa [52].

$$\frac{D_o}{D_i} = \sqrt{\frac{6 + p_i(1 - 2\mu)}{6 - p_i(1 - 2\mu)}} \quad (3-1)$$

Putting $D_o=61.5\text{mm}$

$\sigma= 214\text{Mpa}$

$P_i =2100 \text{ kPa}$

$\mu=0.29$

We get $D_i=57.5\text{mm}$ [dimension taken from direct measuring of the existing Shock absorber model in workshop and modifying with modified shock absorber]

Now, thickness of the cylinder,

$$T = \frac{D_o - D_i}{2} \quad (3-2)$$

Putting $D_o= 61.5\text{mm}$

$D_i= 57.5\text{mm}$

We get, $T= 2 \text{ mm}$

Where, $D_o=$ Outer diameter of cylinder

$D_i=$ Inner diameter of cylinder

$T=$ thickness of cylinder

$\sigma =$ Maximum allowable stress

$P_i=$ Maximum pressure inside cylinder

Table 3-1: Inner cylinder dimension

NO	Parameter	Measured value (mm)
1	Inner diameter	57.5

2	Outer diameter	61.5
3	Length	230
4	Tensile stress of structural steel	21500kpa
5	Poisson ratio	0.29
6	Body material	Structural steel

Table 3-2: Outer cylinder parameters

NO	Parameter	Measured value (mm)
1	Inner diameter	65
2	Outer diameter	80
3	Length	273
4	Tensile stress of structural steel	21500kpa
5	Poisson ratio	0.29
6	Body material	Structural steel

3.3 Design of piston

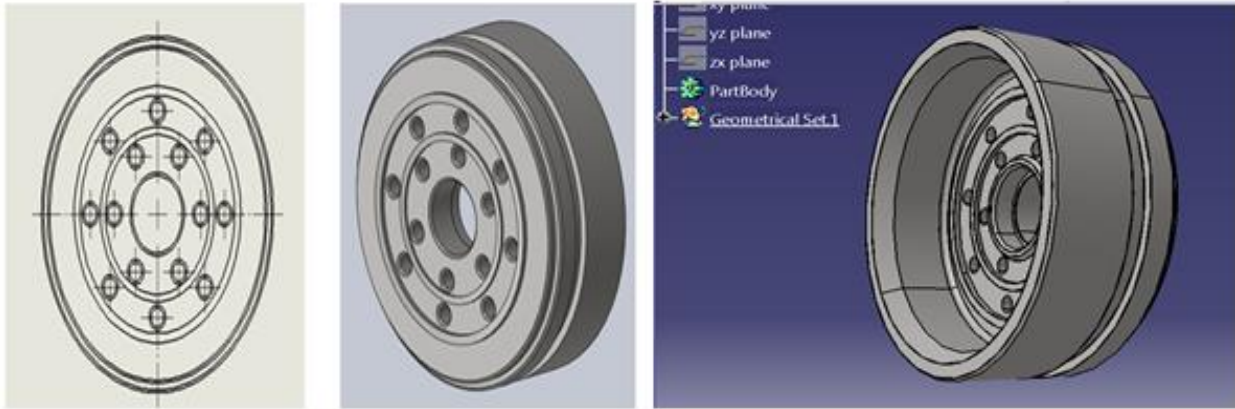


Figure 3-3: 2D and 3D model of the proposed design of the piston for the damper

❖ **Calculation of piston dimensions:** [52].

$$\text{Piston diameter} = \text{Inner cylinder diameter} - \text{Two} \quad (3-3)$$

Inner cylinder diameter = 57.5 [dimension taken from direct measuring of the Existing Shock absorber model in workshop and modifying with modified Shock Absorber]

$$\begin{aligned} \text{Therefore, Piston diameter} &= 57.5 - 2 \\ &= 55.5 \text{ mm} \end{aligned}$$

Therefore, diameter of piston = 55.5 mm

A perfectly straight absorber piston act as bar of uniform cross section with two axis of symmetry C_x and C_y in the cross section as shown in figure 3-4, end thrust P are applied along the centroidal axis C_z of stiffness for bending about C_x .

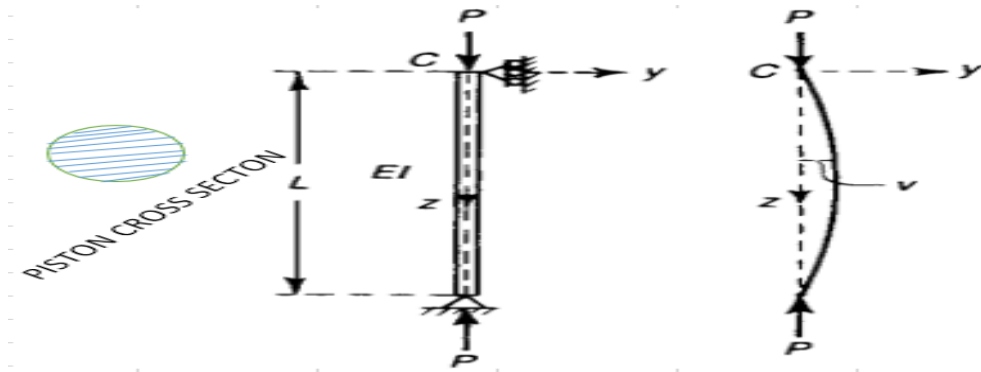


Figure 3-4: Cross section of piston

Now C_x is the weakest axis of bending of the absorber piston, and if bowing of the compressed absorber piston occurs we should expect bending to take place in the $y-z$ plane. Consider the possibility that at some value of P , the end thrust, the strut can buckle laterally in the $y-z$ plane, there can be no lateral deflection at the end of the strut; suppose ' v ' is the displacement of the center – line of the piston parallel to c_y at any point. There can be no forces at the hinges parallel to c_y , as these would imply bending moments at the ends of the bar. The only two external forces are the end thrusts P , which are assumed at any section of the bar is then,

$$M = P v \quad (3-4)$$

This is the general equation of columns. From this general equation, the following formula with some rearrangements obtained

$$P_{cr} = \frac{\pi EI}{l^2} \quad (3-5)$$

P_{cr} = the critical load that leads to column buckle can express by P .

E = elastic modulus of the material,

I = moment of inertia

l = overall length of column or in this case the length of absorber piston that obtained

From the overall structural mounting assemblies of vibration load absorber in CATIA Software.

$$I = \frac{\pi d^4}{64} \quad (3-6)$$

Where d= diameter of the piston.

Now, to make a safe absorber piston design, the critical stress must be less than the allowable or yield strength of the material.

$$\text{i.e. } \sigma_c < \sigma_y$$

$$\text{Where, } \sigma_c = \frac{P_{cr}}{A} \quad (3-7)$$

σ_y = allowable yield strength of the steel material

σ_c = critical stress

The circular cross sectional area of the piston can be calculated as follows.

$$A = \frac{\pi d^2}{4} \quad (3-8)$$

The value of various parameters related to piston buckling is given in table 3.3 [50].

Table 3-3: Parameters under piston buckling analysis

No.	Parameter	Values
1	P_{cr}	705590.42N
2	E	210GPa
3	L	22mm
4	D	57mm
5	σ_y	280MPa
6	σ_c	260MPa

As the critical stress and yielding stress values are shown in table 3.3, the critical stress is less than the yielding stress of the material. Thus the material of the piston can withstand the vibration load up to critical load of 705590.42N.

Table 3-4: Piston parameters

NO	Parameter	Measured value (mm)
1	piston diameter	55.5
2	Opening diameter	3
3	Length	22
4	Body material	Structural steel

3.4 Design of Piston –rod

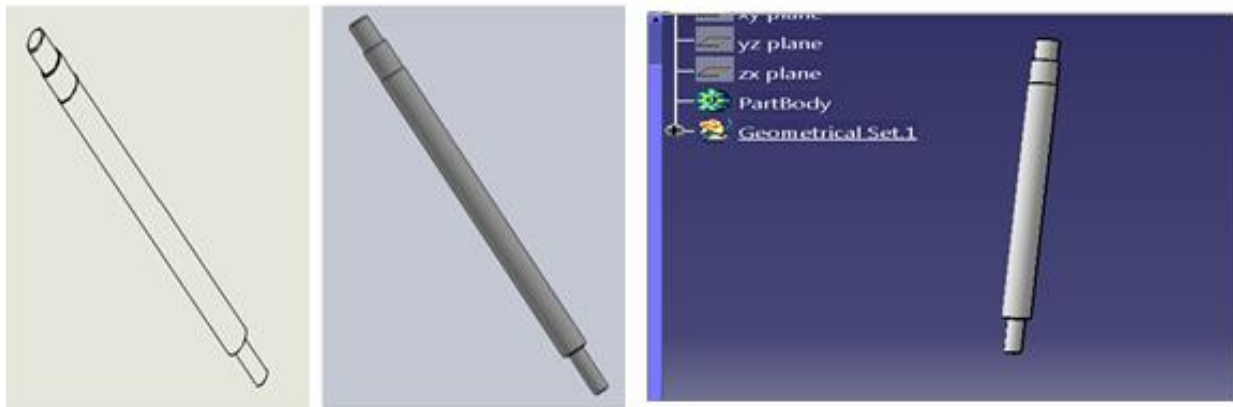


Figure 3-5: 2D and 3D model of piston rod

Hence when length of piston rod is shorter, then the cross section required to transfer force within safe stress limit can be calculated using basic simple formula such as [52].

Force to be transfer = cross –section area x permissible compressive or tensile Stress.

- Estimation of damping force according to Loading. The equation used for finding the damping force was,

$$\text{Damping force} = \frac{\text{Energy per strok[N.m]} * \text{correction Factor} * 1000}{\text{stroke[mm]}} \quad (3-9)$$

- For finding out damping force, first you need to find (E) energy per stroke in N.m.

$$\text{Energy per stroke, } E = m * g * h + m * s \quad (3-10)$$

Where,

m = impact mass [kg]

g = acceleration due to gravity [m/s²] = 9.81

s = acceleration height [m]

h = damping distance[m]

In our case, impact mass, m= 1890 kg [designing shock absorber for 1890 kg]

Acceleration height = 0 m

Damping distance = 0.04 sin 26° = 0.017 m

Substituting the above values in equation, we get,

Energy per stroke = 314.874 N. m

❖ Damping correction factor:

$$\text{DCF} = \left[\frac{10}{5 + \zeta} \right]^{0.25} \quad (3-11)$$

For under damped system, $\zeta = 0.4$

Putting it in above equation, we get DCF = 1.17

Where, ζ = Damping ratio (zeta)

❖ Spring Stroke

From the design we found out the free length, i.e. 279.099 mm and solid length 80 mm.

From this we calculated stroke= 199.099 mm

Substituting all the values of 1, 2 and 3 in equation of damping force (3-9), we get

$$\text{Damping force} = \frac{314.874 * 1.17 * 1000}{199.099} \quad (3-12)$$
$$= 1851.269 \text{ N}$$

Considering 1851N damping forces

Now, finding out the force due to gas pressure, which is given by the equation (3-13)

$$F_{\text{gas}} = p_{\text{gas}} * A_{\text{rod}} \quad (3-13)$$

In this shock absorber, the gas is replaced by gas foam or foam cell. The gas foam is pocket of air inside let us assume the pressure ranges between 260-360 psi.

Considering 300 psi, i.e. 2100 kPa/ 21 bar

$$A_{\text{rod}} = \frac{\pi}{4} (\text{diameter of piston rod})^2 \quad (3-14)$$

$$A_{\text{rod}} = \frac{\pi}{4} (20)^2$$

$$A_{\text{rod}} = 314 \text{ mm}^2$$

Putting this in equation of Gas pressure force, (3-13)

$$F_{\text{gas}} = 21 \text{ bar} \times 314 \text{ mm}^2 = 6594 \text{ N}$$

Therefore, total damping force considered for analysis in ANSYS is,

$$= 1851 \text{ N} + 6594 \text{ N}$$

=2165 N

Table 3-5: Piston rod parameters

NO	Parameter	Measured value (mm)
1	Piston rod diameter	20
2	Length	323
3	Body material	Structural steel

3.5 Design of spring

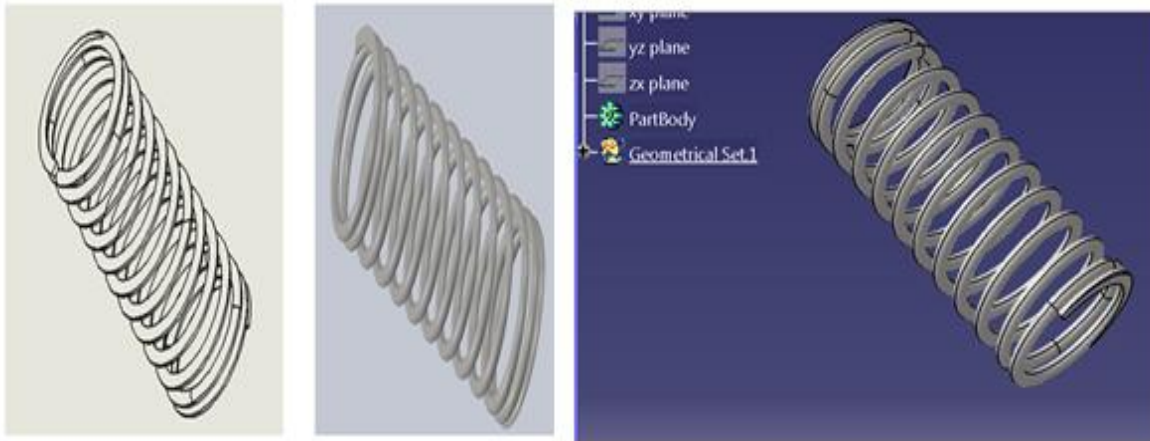


Figure 3-6: 2D and 3D model of spring

Material : (ASTM A228 [HIGH CARBON SPRING WIRE]) [67].

- Properties: Young's Modulus (EX): 20850000 kpa
- Poisson's Ratio (PRXY): 0.31
- Density: 7860 Kg/m³

The material of the spring should have high fatigue strength, high ductility, high resilience and it should be creep resistant.

For the springs in this discussion, Hooke's Law is typically assumed to hold [51].

$$F=k\Delta \quad (3-15)$$

We can expand the spring constant k as a function of the material properties of the spring. Doing so and solving for the spring displacement gives,

$$\Delta = \frac{F}{K} = \frac{8FD^3n_a}{Gd^4} \quad (3-16)$$

Where G = is the material shear modulus

n_a = is the number of active coils,

D = Mean diameter of spring

d = Mean coil diameter

F = spring force

K = spring stiffness

The number of active coils is equal to the total number of coils n_t minus the number of end coils n^* that do not help carry the load,

$$n_a = n_t - n^* \quad (3-17)$$

Where, n_t = total number of coil

The value for n^* depends on the ends of the spring. See the following illustration for different n^* values:

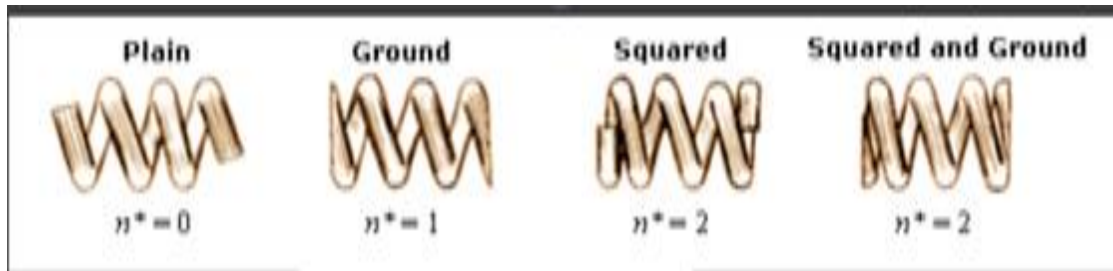


Figure 3-7: Helical coil spring parameters

- ❖ Estimating of force acting on spring and finding out stiffness of spring:

Assuming 5542N and 6020N load acting on the suspension system and damping

Force= 1851N and force of gas= 6594 N

- ✓ For (W= 5542 N)

$$\text{Load acting on spring} = \text{load acting on suspension} - \left[\frac{\text{damping force}}{2} + \text{force of gas} \right] \quad (3-18)$$

$$= 5542 - \left[\frac{1851\text{N}}{2} + 6594 \text{ N} \right]$$

$$= 1977.5\text{N}$$

- ✓ For (W= 6020 N)

$$\text{Load acting on spring} = \text{load acting on suspension} - \left[\frac{\text{damping force}}{2} + \text{force of gas} \right] \quad (3-19)$$

$$= 6020 - \left[\frac{1851\text{N}}{2} + 6594 \text{ N} \right]$$

$$= 1499\text{N}$$

- ❖ Stiffness of spring

$$K = \frac{W}{\delta} \quad (3-20)$$

We Know that, compression of spring (δ), for $w=1977.5\text{N}$

$$(\delta) = \frac{(wD^3n)}{(Gd^4)} \quad (3-21)$$

$$(\delta) = \frac{1977.5 * 11 * 120^3}{(20850 * 10^4)} = 180 \text{ mm}$$

$$K = \frac{W}{\delta} = 1977.5 / 180 = 11 \text{ N/mm}$$

Where, δ = compression of spring

W = applied load

✓ Compression of spring (δ), for $w=1499\text{N}$

$$(\delta) = \frac{1499 * 11 * 120^3}{(20850 * 10^4)} = 137 \text{ mm}$$

$$K = \frac{W}{\delta} = 1499 / 137 = 11 \text{ N/mm}$$

✓ Estimating the stiffness of spring for load acting externally

$$K = \frac{W}{\delta} \quad (3-22)$$

We Know that, compression of spring (δ), for $w=5542\text{N}$

$$(\delta) = \frac{(wD^3n)}{(Gd^4)} \quad (3-23)$$

$$(\delta) = \frac{5542 * 11 * 120^3}{(20850 * 10^4)}$$

$$(\delta) = 505 \text{ mm}$$

$$K = \frac{W}{\delta} = 5542 / 505 = 11 \text{ N/m}$$

✓ Compression of spring (δ), for $w=6020$

$$(\delta) = \frac{(wD^3n)}{(Gd^4)}$$

$$(\delta) = \frac{6020 \cdot 11 \cdot 120^3}{(20850 \cdot 10^4)} = 548 \text{ mm}$$

$$K = \frac{W}{\delta} = 6020/548 = 11 \text{ N/mm}$$

- ❖ Spring stroke or absorber stroke = The difference between fully extended and fully compressed position (3-24)

$$\text{Stroke} = 279.099 - 80 = 199.099$$

- ❖ Shear stress factor or stress correction factor or Wahl factor is given by

$$\mathbf{k_s} = \frac{4c-1}{4c-4} + \frac{0.615}{c} \quad (3-25)$$

$$= 1.11943181$$

- ❖ The angle of the spring coil relative to the mounting base is found:

$$\theta = \tan^{-1} [C_p / \pi D] \quad (3-26)$$

Where, C_p = is pitch pf the springe

D = is diameter of the coil.

θ = angle of the spring coil relative to the mounting base

$$\theta = \tan^{-1} [27.9/3.14 \times 61.5] = 9^\circ$$

- ❖ Permissible shear stress induced inside spring,

$$\zeta = 0.5 s_{ut} \quad (3-27)$$

Where, ζ = Permissible shear stress

s_{ut} = Ultimate stress of spring material

For (ASTM A228 [HIGH CARBON SPRING WIRE]), $s_{ut} = 1050 \text{ N/mm}^2$

Therefore,

$$\zeta = 0.5 s_{ut}$$

$$\zeta = 526 \text{ N/mm}^2$$

❖ Spring Index

$$C = \left[\frac{D}{d} \right] \quad (3-28)$$

$$\left[\frac{120}{10} \right] = 12$$

❖ Mean coil diameter,

Permissible shear stress induced is also given by,

$$\zeta = \frac{K_s \cdot 8 \cdot P \cdot c}{\pi d^2} \quad (3-29)$$

Putting values of

$$K_s = 1.13$$

$$P = 16281.5 \text{ N}$$

$$c = 12$$

$$\zeta = 526$$

We will get $d = 10 \text{ mm}$

❖ Mean diameter of spring

Spring index is given by,

$$C = \frac{D}{d} \quad (3-30)$$

Putting the values of c and d , we get

$$D= 120\text{mm}$$

❖ Number of active coils

Putting the values of δ , p. D. d and $G=20850000 \text{ kpa} / 20850.045 \text{ N/mm}^2$ for (ASTM A228 [HIGH CARBON SPRING WIRE]), we get number of active coils.

Assuming square and ground spring,

$$\begin{aligned} \text{No. of active coils} &= \text{total number of coils} - \text{Two} \\ &= 11 \end{aligned} \quad (3-31)$$

❖ Solid length

$$\begin{aligned} \text{Solid length} &= N_t \times d \\ &= 110 \text{ mm} \end{aligned} \quad (3-32)$$

❖ Gap

$$\begin{aligned} \text{Total gap} &= \text{compressed length} - \text{solid length} \\ &= 209.099 - 110 = 99.099\text{mm} \end{aligned} \quad (3-33)$$

$$\text{Gap between two coils} = \frac{\text{Total gap}}{N_t - 1} \quad (3-34)$$

$$\begin{aligned} &= \frac{99.099}{11 - 1} \\ &= 9.9099\text{mm} \end{aligned}$$

❖ Pitch

$$\text{Pitch} = \frac{\text{free length}}{N_t - 1} \quad (3-35)$$

$$\begin{aligned} &= \frac{279.099}{10} \\ &= 27.9099\text{mm} \end{aligned}$$

❖ Stresses in helical springs, maximum shear stress induced in the wire

✓ At load $W = 4655\text{N}$ is

$$\tau_{sp} = k_s \times \frac{8 \times W \times c}{\pi d^2} \quad (3-36)$$

Where, $k_s = 1.11$

$$1.119 \times \frac{8 \times 4655 \times 12}{3.14 \times 10^2} = 1592.54 \text{ pa}$$

✓ At load $W = 18522\text{N}$

$$\tau_{sp} = k_s \times \frac{8 \times W \times c}{\pi d^2} = \text{where, } k_s = 1.119$$

$$1.119 \times \frac{8 \times 18522 \times 12}{3.14 \times 10^2} = 5662.77 \text{ pa}$$

❖ Buckling of compression springs:

$$W_{cr} = k \times k_B \times L_F \quad (3-37)$$

Where, k = spring rate or stiffness of spring $= \frac{W}{\delta}$

L_F = free length of the springs

k_B = Buckling factor depending up on the ratio $= \frac{L_F}{D}$

Value of buckling factor k_B

$$\frac{L_F}{D} = \frac{279.0}{120} = 2.32$$

$K = 0.05$ (for hinged and spring)

The buckling factor for the hinged end and built-in end spring, therefore

$$W_{cr} = 0.05 \times 1.37138 \times 279 = 19.130751\text{N}$$

Table 3-6: Size specification of outer spring

No	Parameter	Value
1	Mass of the vehicle model	1590kg
2	Maximum deflection of outer spring	0.331384895
3	Force acting on spring	16281N
4	Spring index	12
5	Wire diameter	10mm
6	Spring mean diameter	120
7	Total number of turns	11
8	Free length	279
9	Pitch	27.9099mm
10	Spring stiffness	11N/mm

3.5.1 Selection of fluid for damping;

3.5.1.1 Dynamics of the Damper

Dampers provide a resistive force to counteract the release of stored energy in a spring by passing oil through small passages. The viscosity of the oil causes a pressure drop which in turn, produces force acting on the piston. The forces produced in each stroke can be tailored to similarity and dependence or uniqueness and independence depending, on the construction of the damper [53].

❖ Damping force inside damper is given by,

$$F = (P_1 - P_2) \times A \quad (3-38)$$

$$\text{Therefore, } (P_1 - P_2) = \frac{F}{A}$$

Where:

P_1 = Pressure above piston

P_2 = Pressure below piston

$$(P_1 - P_2) = 31.9227388 \text{ bar}$$

❖ Equation of pressure difference for laminar flow is given by,

$$P_1 - P_2 = (32 \times \mu \times V_{\text{avg}} \times l) / D^2 \quad (3-39)$$

D = equivalent diameter of flow

μ = dynamic viscosity

l = length of the flow

V_{avg} = velocity of piston

Equivalent diameter,

$$4\left[\frac{\pi}{4} d^2\right] = \frac{\pi}{4} D^2 \quad (3-40)$$

$$D = 2d$$

Replacing value of D in equation 3.40 we get

$$P1-P2 = (32 \times \mu \times V_{avg} \times l) / 4d^2$$

Putting $F=8999 \text{ N}$, $A=0.002550465 \text{ m}^2$, $V_{avg}=1.5 \text{ m/s}$, $l = 0.23 \text{ m}$, $d= 0.0575 \text{ m}$

In above Equation (3-39) we get viscosity,

$$\mu = 0.0087 \text{ Ns/m}^2$$

❖ Weight of vehicle = 1590kg/15582N

Let weight of one person = 75 Kgs /735N

❖ For Weight of two person

Weight of two person=150 Kgs/ 1470N

Total Weight ($\mathbf{W_t}$) = Weight of vehicle (curb weight) + Weight of two persons
(3-41)

❖ Vehicle :Toyota Hilux japan SR5 4X4 ,1999

$$= 1590 + 150 = 1740 \text{ Kgs} / 17052 \text{ N}$$

Rear suspension = 65%

$$65\% \text{ of } 1740 = 1131 \text{ Kgs}$$

Considering dynamic loads it will be doubled

$$\mathbf{W_t} = 2262 \text{ Kgs} = 22167.6 \text{ N}$$

Single shock absorber weight ($\mathbf{W}$) = $\mathbf{W_t} / 4$ (3-42)

$$= 22167.6 / 4 = 5541.9$$

❖ For Weight of four person

Weight of four person=300 Kgs/2940N

$$(3-43) \quad \text{Total Weight (W}_t\text{)} = \text{Weight of vehicle (curb weight)} + \text{Weight of four persons}$$

$$= 1590 + 300 = 1890 \text{Kgs}/18522 \text{N}$$

Rear suspension = 65%

$$65\% \text{ of } 1890 = 1228.5 \text{Kgs}$$

Considering dynamic loads it will be doubled

$$\mathbf{W}_t = 2457 \text{Kgs} = 24078.6 \text{N}$$

For single shock absorber weight (W) = $\mathbf{W}_t/4$

(3-44)

$$= 24078.6/4 = 6019.65 \text{N}$$

3.5.2 Shock absorber assembly

This assembly is done using CATIA software in the assembly workbench .To get the assembly work Bench; click start, mechanical design, assembly design and after that import each component by clicking “existing component with positioning “.the assembly was done orderly starting from cylinder as base for others. Then bring all inner parts and join its axis and one end ground seat with the cylinder inner base surface. After this assembly as shown in figure 3.9,bring piston and outer spring turn by turn following the same procedure .the detail assembly drawing is attached in the Appendix –A.

3.5.2.1 Shock absorber assembly with sectional view



Figure 3-8: Existing shock absorber model

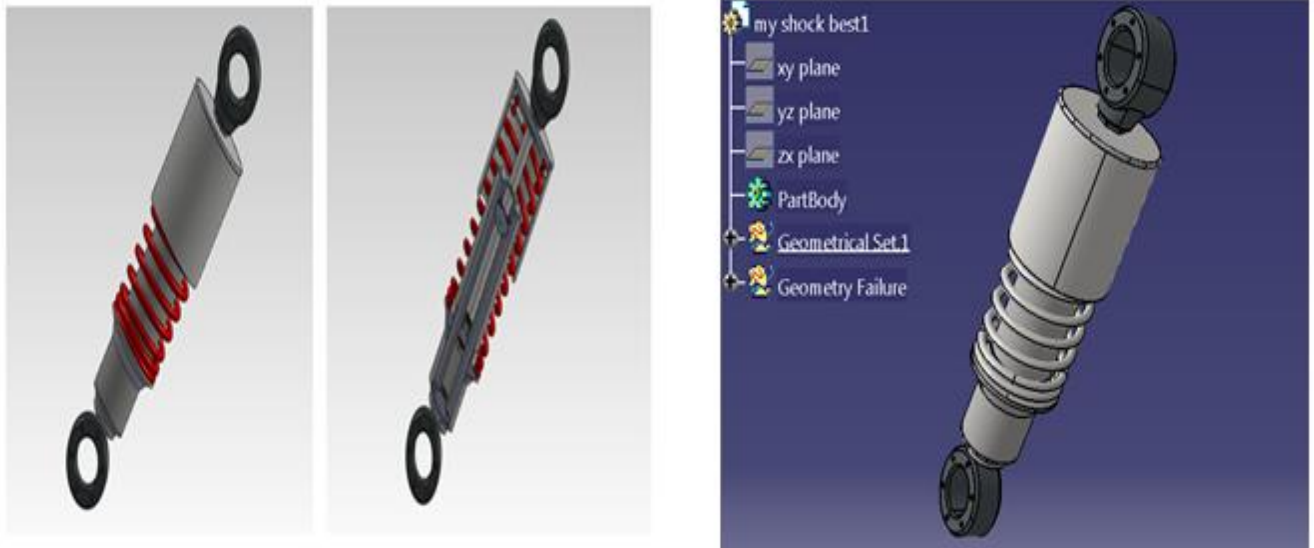


Figure 3-9: Modified shock absorber model



Figure 3-10: Exploded view of the existing shock absorber

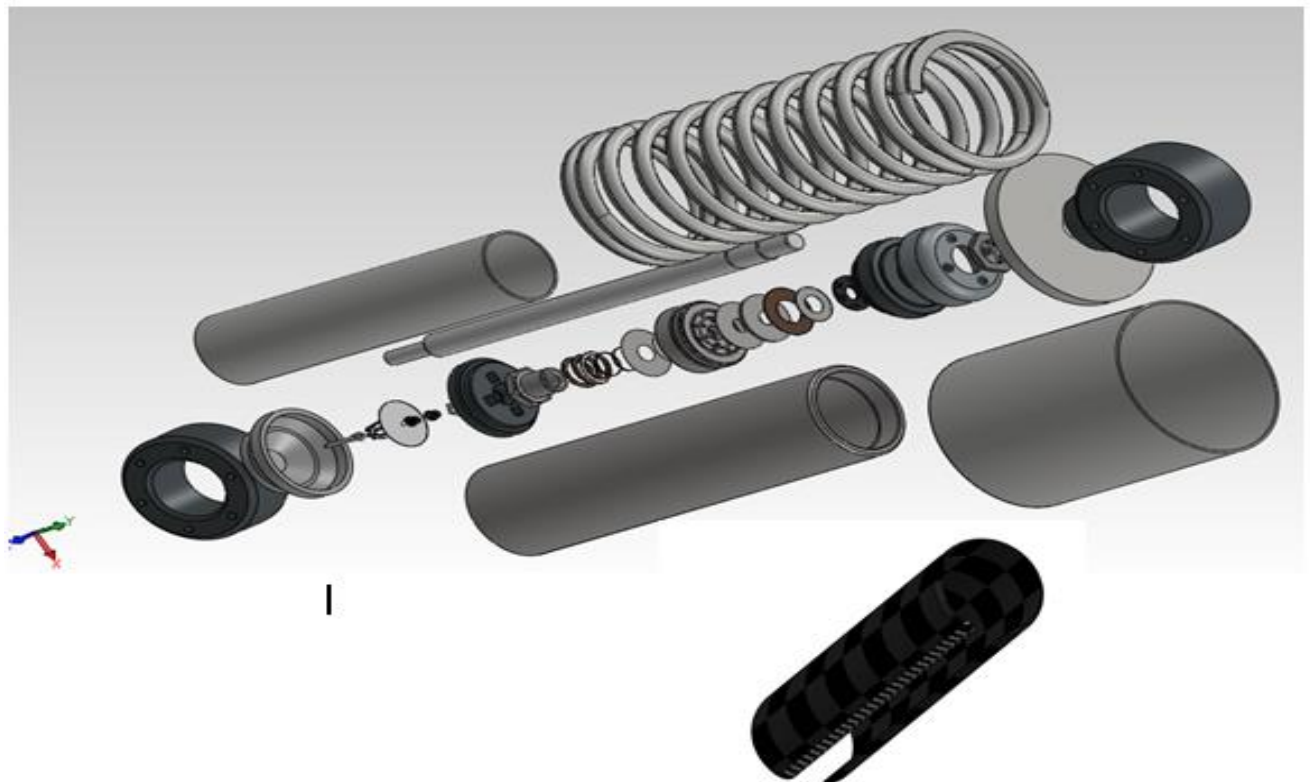


Figure 3-11: Exploded view of the modified shock absorber

3.5.3 Modified shock absorber

There are two big distinct differences to the N_2 gas charged or the existing shock absorber

1. Compressed spring : “To maximize tire control with the rod and reduce riding height”
2. Foam cell /Gas cell; a gas impregnated sheet of closed cell foam is positioned inside the shock absorber to replace the floating gas. This separates the oil and gas, allowing full oil contact with the outer cylinder to radiate heat away from the shock absorber more effectively



Figure 3-12: Foam cell

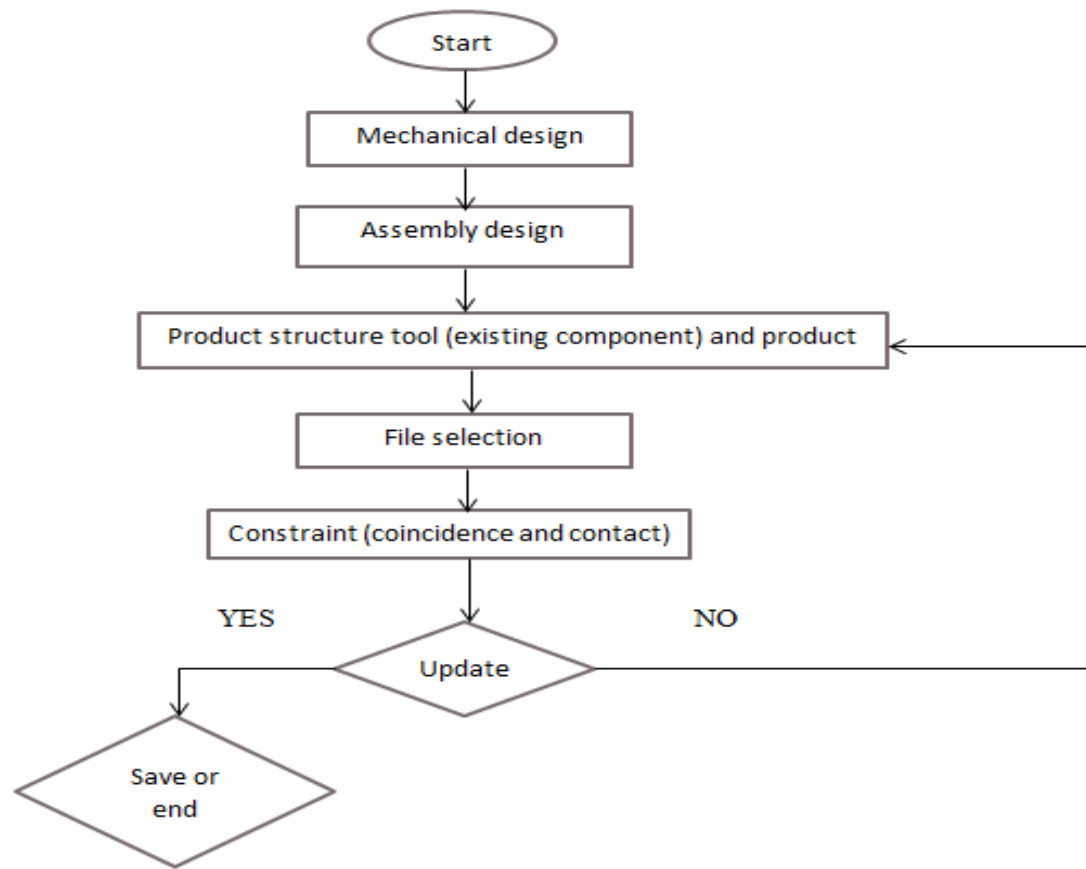


Figure 3-13: Flow chart that shows the general assembly of models

3.5.4 Finite element analysis

ANSYS R15 is powerful numerical solver that is used due to its ability in FEA given the mechanical nature of this thesis .The details of the simulation are mentioned later in this report, however ,the basic format of the program is parceled interface with sub windows and subroutines that define each given detail of the simulation separately and combine using the master window .This interface is ideal for the application at hand due to its ability in running multiple steps because each parameter is defined in its own subroutine and can be shared between simulation .In this case for motion analysis solid work is used and static and modal analysis is used for deformation analysis .

3.5.5 Meshing

Meshing is a process used commonly by numerical solver, breaking a complex geometry part in to small geometric shapes. These shapes are then mathematically coupled and solved as individual parts for stiffness and deflection being combined in the end for a global result of the effect on the part or assembly as a whole. These elements come in many shape and size limited only by the user's imaginations to fit specific need. A smaller mesh size means having more, or smaller element filling the part allowing for a more refined definition of how the stress is passing through the part. Having more elements also increase the computational time linearly with elements and to the power of two in relationship to the area being covered. Refinements to a standard mesh include techniques such as sizing, sweeping and shaping, amongst others not used here. sweeping is a technique that places a planar mesh surface in the thickness of the part .Shaping can be utilized much like swapping coordinate system, for instance a hexagonal mesh is beneficial for round parts and a triangular mesh is better for planar parts .This can also get more complex with the introduction of trapezoids and other simple polygons and can be found to be more beneficial for a given part by trial and error.

The steps to mesh the given geometry are as follows;

- ✓ On the model workbench, click 'mesh' on the project
- ✓ Then on the detail edit the relevance from 0 to 1
- ✓ Under sizing the element size is edited depending on the geometry size and the capacity of the computer
- ✓ The relevance center should edit to either coarse, medium or fine. From these alternative fine is the recommended one for suitable a smaller element size depend on the computer capacity
- ✓ The smoothing item should be edited to either high, medium or low, high is better.
- ✓ Such other necessary things should be edited.

- ✓ Finally after all necessary things are edited, click 'update' or 'generate mesh' and wait up to get the required result

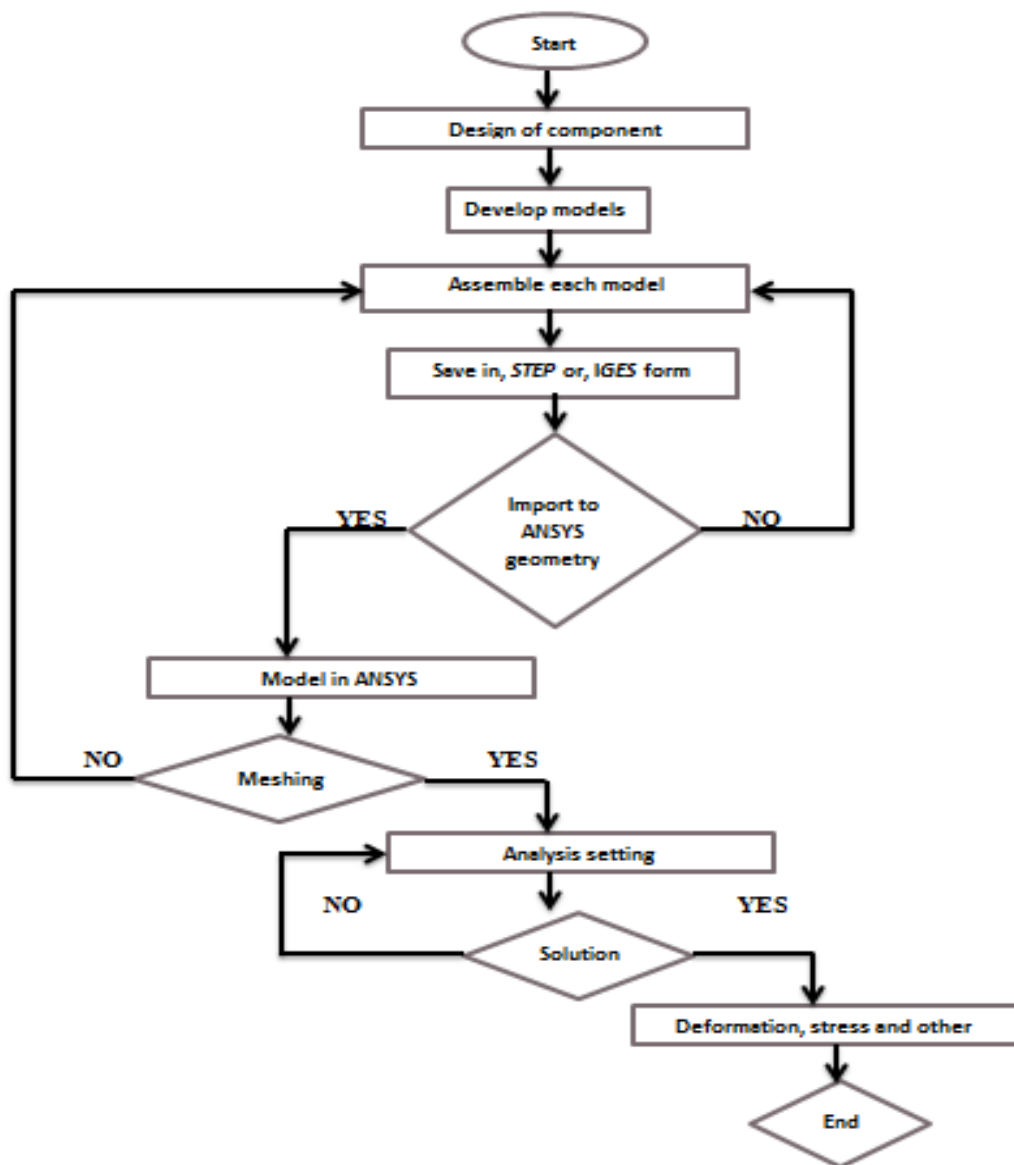


Figure 3-14: Flow chart procedure for modeling in ANSYS workbench

3.5.6 Material selection

Selection of the suitable material for mechanical engineering areas is a key aspect for every researcher. Even a small amount in weight reduction of the vehicle, have a wider economic impact. The material which is capable of absorbing more energy in the form of elastic deformation would be preferred for shock absorber and other vehicle body and chassis frames.

CHAPTER FOUR

4 RESULT AND DISCUSSION

4.1 Finite element analysis

Finite Element Analysis (FEA) is a computer-based numerical technique for calculating the strength and behavior of engineering structures. It can be used to calculate deflection, stress, vibration, buckling behavior and many other phenomena. It can be used to analyze either small or large-scale deflection under loading or applied displacement. It can analyze elastic deformation, or "permanently bent out of shape" plastic deformation. The computer is required because of the astronomical number of calculations needed to analyze a large structure. The power and low cost of modern computers has made Finite Element Analysis available to many disciplines and companies. In the finite element method, a structure is broken down into many small simple blocks or elements. The behavior of an individual element can be described with a relatively simple set of equations. Just as the set of elements would be joined together to build the whole structure, the equations describing the behaviors of the individual elements are joined into an extremely large set of equations that describe the behavior of the whole structure. The computer can solve this large set of simultaneous equations. From the solution, the computer extracts the behavior of the individual elements. From this, it can get the stress and deflection of all the parts of the structure. The stresses will be compared to allowed values of stress for the materials to be used, to see if the structure is strong enough. The term "finite element" distinguishes the technique from the use of infinitesimal "differential elements" used in calculus, differential equations, and partial differential equations. The method is also distinguished from finite difference equations, for which although the steps into which space is divided are finite in size, there is little freedom in the shapes that the discrete steps can take. Finite element analysis is a way to deal with structures that are more complex than can be dealt with analytically using partial differential equations. FEA deals with complex boundaries better than finite difference equations will, and gives answers to "real world" structural problems. It has been substantially extended in scope during the roughly 40 years of its use.

4.1.1 Analysis using ANSYS

ANSYS is an engineering simulation software provider founded by software engineer John Swanson. It develops general-purpose finite element analysis and computational fluid dynamics software. While ANSYS has developed a range of computer-aided engineering (CAE) products, it is perhaps best known for its ANSYS Mechanical and ANSYS Multi-physics products. ANSYS Mechanical and ANSYS Multi-physics software are non-exportable analysis tools incorporating pre-processing (geometry creation, meshing), solver and post-processing modules in a graphical user interface. These are general-purpose finite element modelling packages for numerically solving mechanical problems, including static/dynamic structural analysis (both linear and non-linear), heat transfer and fluid problems, as well as acoustic and electro-magnetic problems. ANSYS Mechanical technology incorporates both structural and material non-linearity. ANSYS Multi-physics software includes solvers for thermal, structural, CFD, electro-magnetic, and acoustics and can sometimes couple these separate physics together in order to address multidisciplinary applications. ANSYS software can also be used in civil engineering, electrical engineering, physics and chemistry. Every analysis involves four main steps

4.2 Structural Analysis

It is used to determine displacements stresses, etc. under static loading conditions. ANSYS can compute both linear and nonlinear static analyses. Nonlinearities can include plasticity, stress stiffening, large deflection, large strain, hyper elasticity, contact surfaces, and creep.

4.3 Static Structural Analysis

4.3.1 Boundary condition

The boundary conditions applied to the model were critical to create a realistic load pattern. Because the goal of a shock absorber is to minimize the motion of the car body, the top bracket was constrained to be immobile. The rest of the shock absorber was constrained to allow motion only in the vertical (y) direction. Finally, the load was applied; a traction force of 5542 N and 6020N was applied to the bottom bracket in the vertical direction. Structural analysis of existing shock absorber model

- ✓ Considering the weight of the vehicle and weight of 2 person (When $W = 1740\text{Kgs}/5542\text{N}$)
 - Material; Structural steel

Step 1: Importing Shock Absorber Model in ANSYS workbench and assigning the Material (Material Properties).

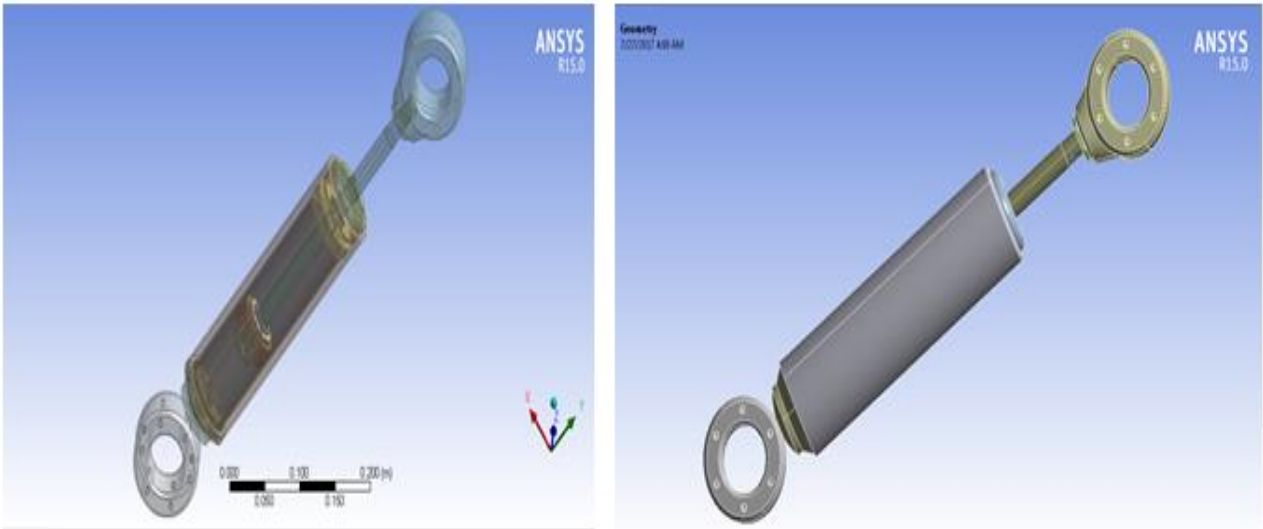


Figure 4-1: Existing shock absorber in Add frozen and in Add material

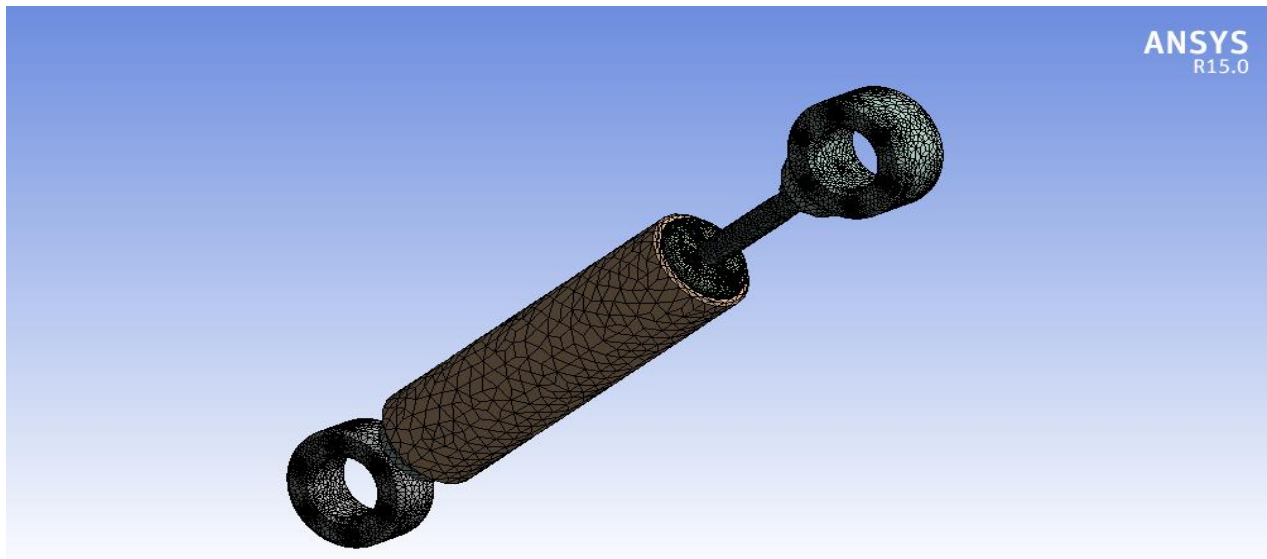


Figure 4-2: Applying the Mesh (Meshing)

Table 4-1: Meshing Statistics of the static structural analysis

Number of nodes	76344
Number of elements	38597
Element size	0.001
Analysis type	Static structural

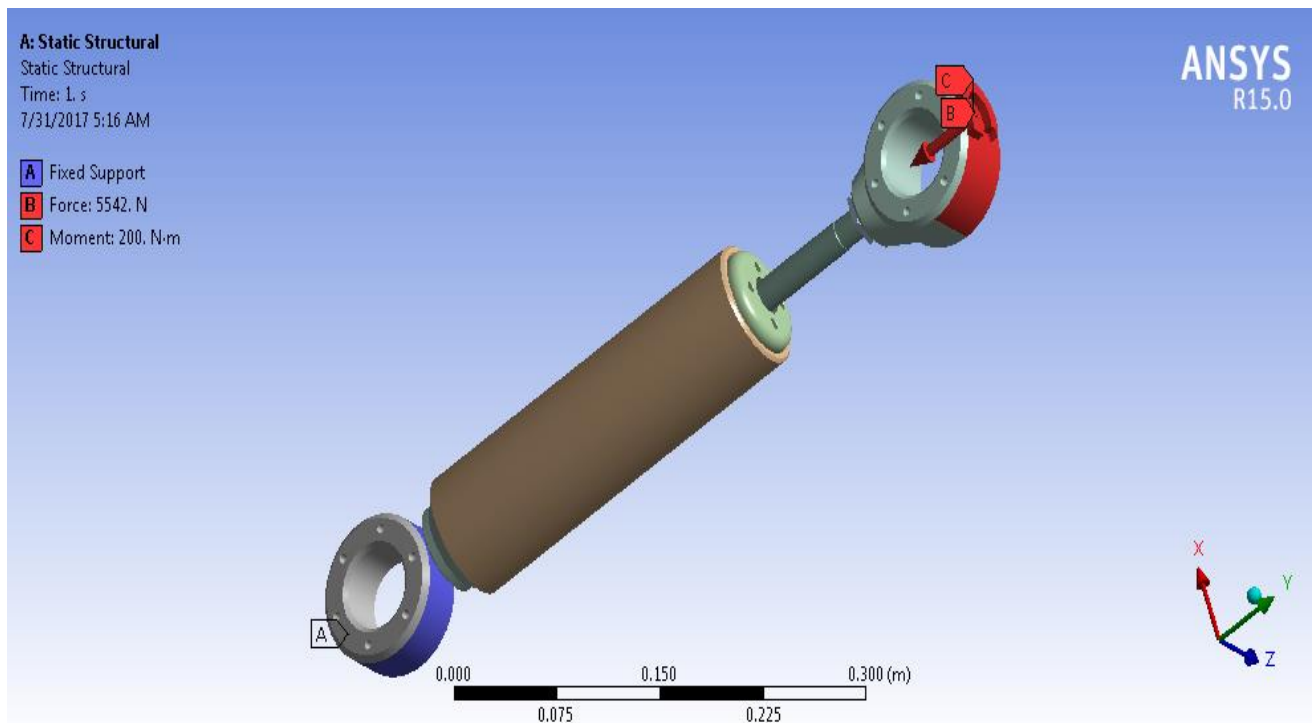


Figure 4-3: Applying the Loads, moment and Boundary Conditions

Figures 4-3 shows that the loading and boundary condition on a shock absorber, Also the boundary condition is applied at fixed position of a model which is consider as a supports, the load is applied considering weight of vehicle and two persons at top part of shock absorber model.

4.3.2 Obtaining results after applying loads

Figure 4-4 shows the total deflection in the shock absorber when the load of 5542 N is applied for the existing shock absorber model.

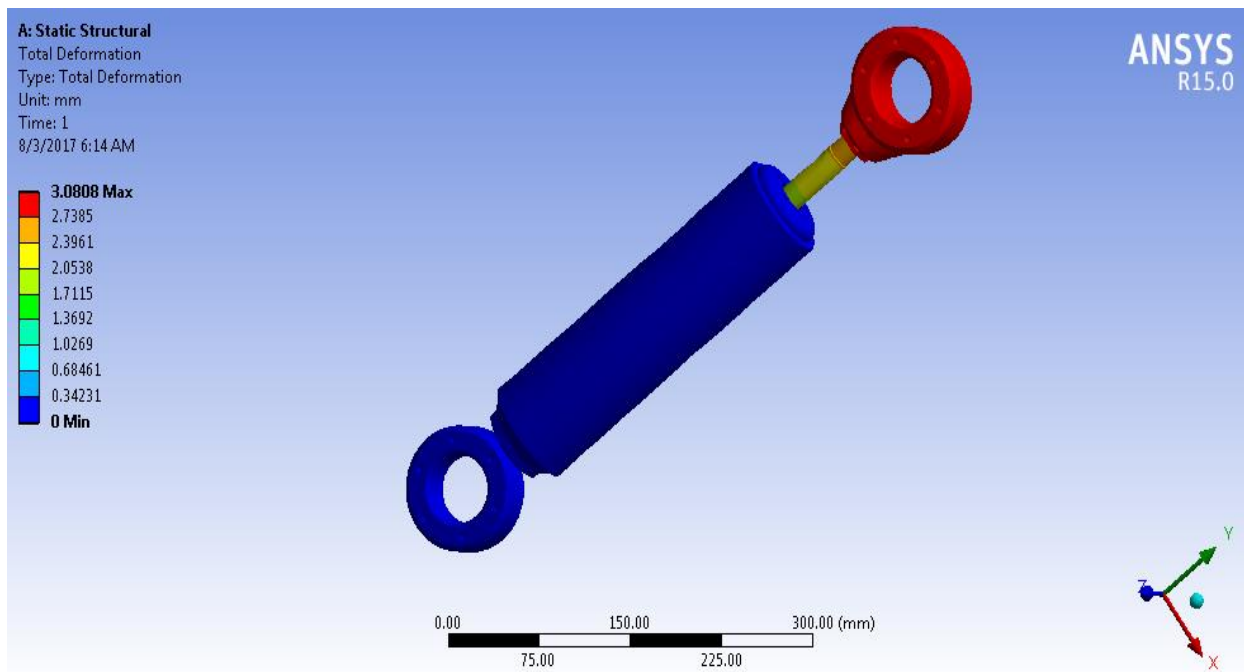


Figure 4-4: Total Deflection for load of ($W = 5542\text{N}$)

- ✚ The maximum value (maximum deformation) is indicated with red color which is 3.0000mm and blue color indicates the minimum value as shown in figure 4-4;

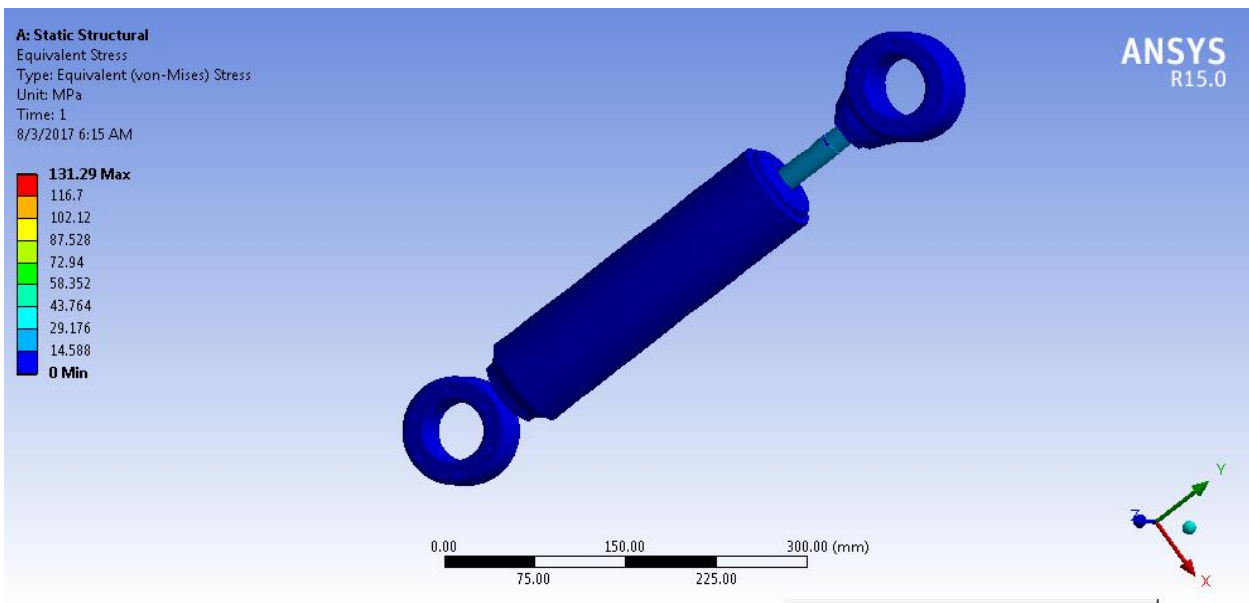


Figure 4-5: Equivalent (von-mises) stress for load of (W= 5542N)

- The maximum value (maximum equivalent stress(von-mises stress)) is indicated with red color which is 131.29 MPa and blue color indicates the minimum value as shown in figure 4-5;

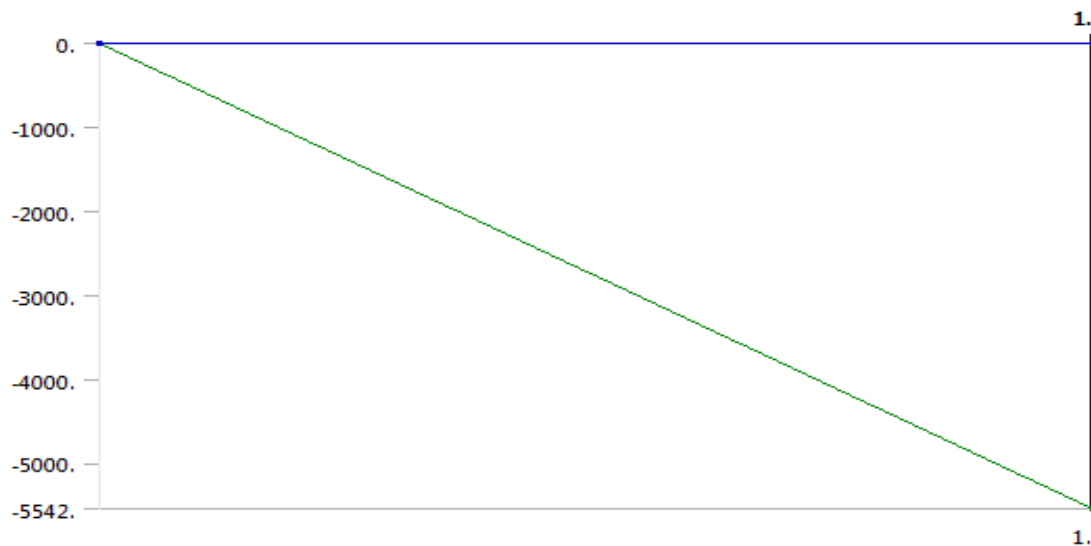


Figure 4-6: Force plot for load (W= 5542N)

- This figure 4-6; describes when applying the load (W= 5542 N) the direction of the force in the y direction is negative direction (W= -5542 N)

4.3.2.1 Obtaining results after applying loads

Figure shows the total deflection in the shock absorber when the load of 6020 N is applied for the existing shock absorber model.

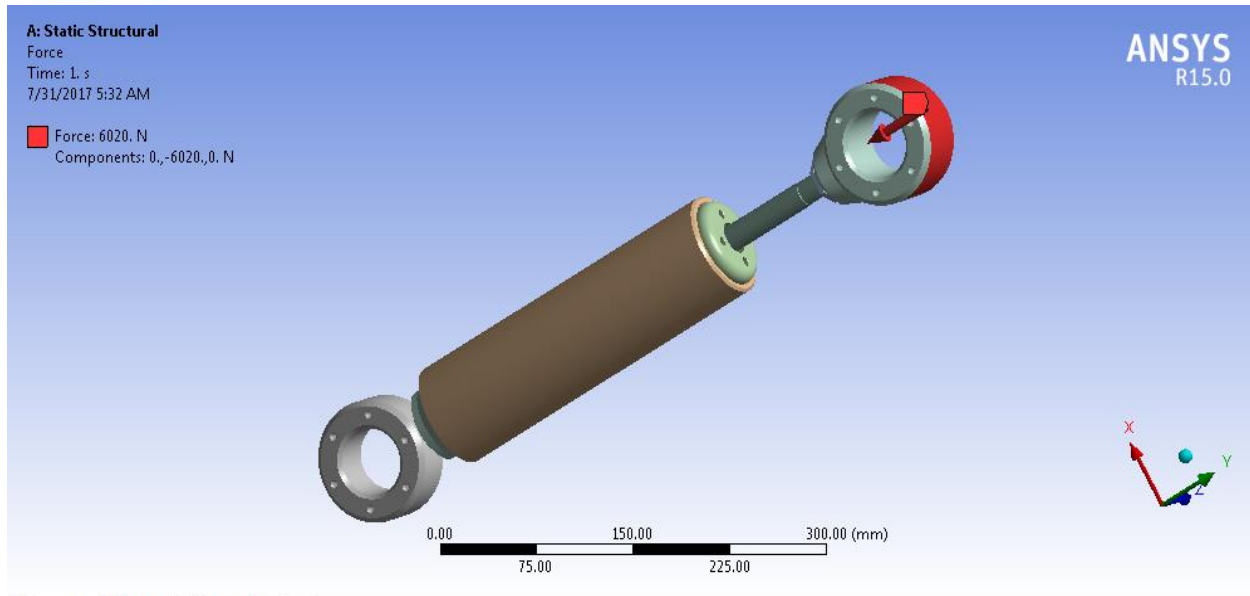


Figure 4-7: Applying load (W= 6020N)

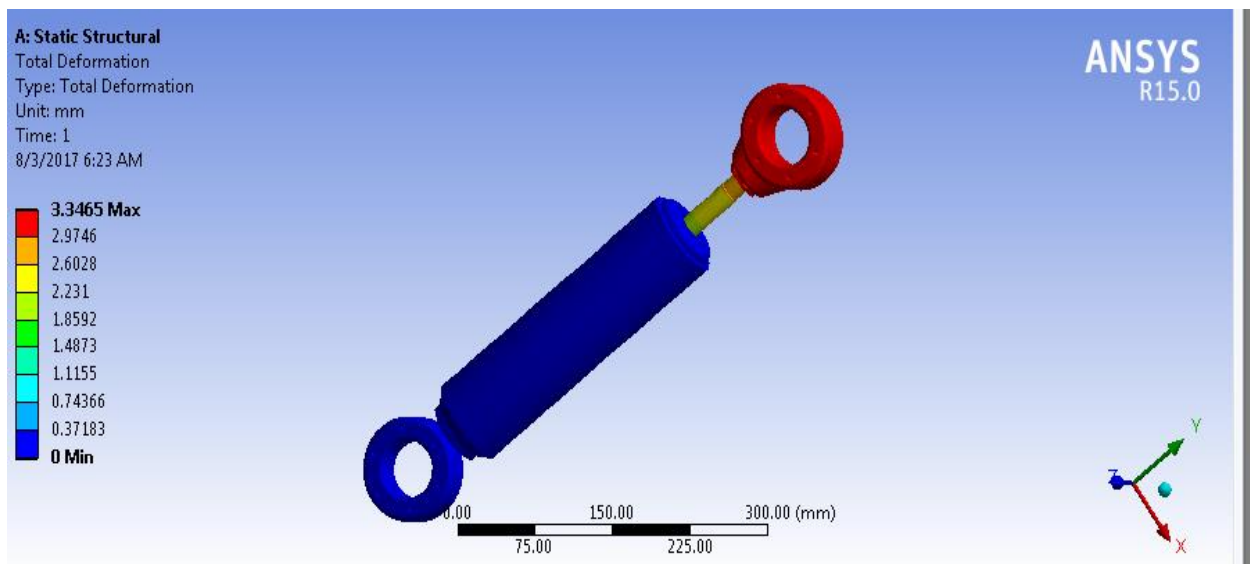


Figure 4-8: Total deformation

- ✚ The maximum value (maximum deformation) is indicated with red color which is 3.3465 mm and blue color indicates the minimum value as shown in figure 4-8;

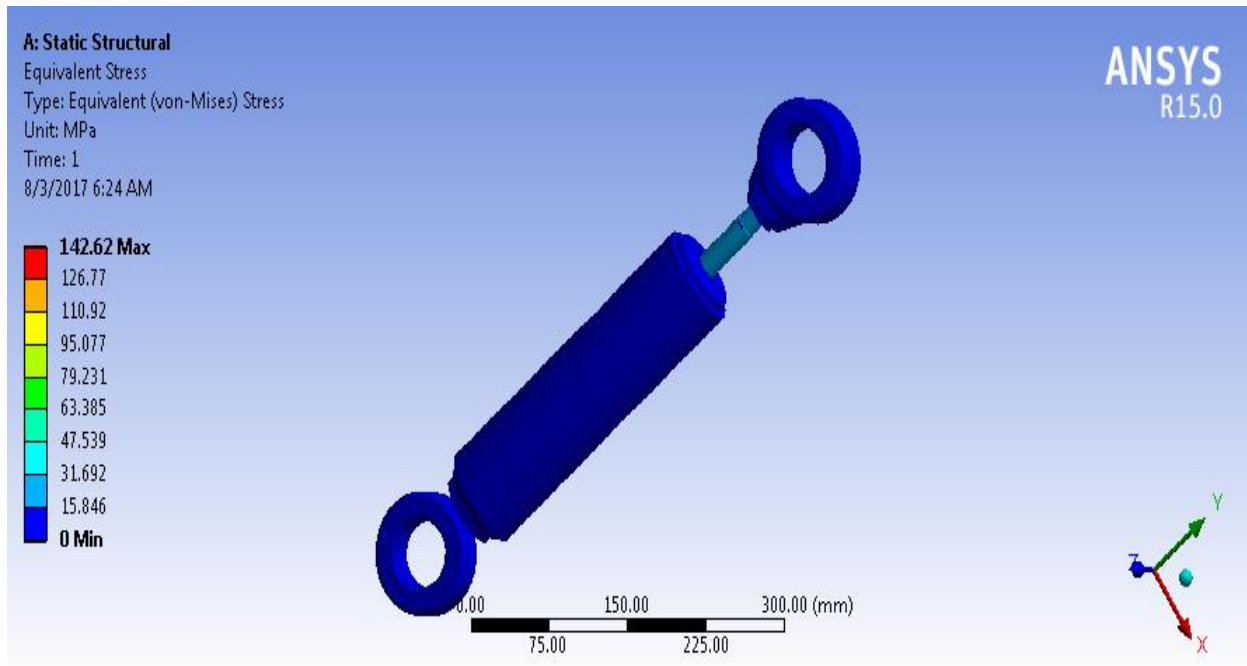


Figure 4-9: Equivalent (von-mises) stress

- ✚ The maximum value (maximum equivalent stress(von-mises stress)) is indicated with red color which is 142.62 Mpa and blue color indicates the minimum value as shown in figure 4-9;

4.4 Modal Analysis for existing shock absorber

A modal analysis is typically used to determine the vibration characteristics (natural frequencies and mode shapes) of a structure or a machine component while it is being designed. It can also serve as a starting point for another, more detailed, dynamic analysis, such as a harmonic response or full transient dynamic analysis. Modal analyses, while being one of the most basic dynamic analysis types available in ANSYS, can also be more computationally time consuming than a typical static analysis. A reduced solver, utilizing automatically or manually selected master degrees of freedom is used to drastically reduce the problem size and solution time.

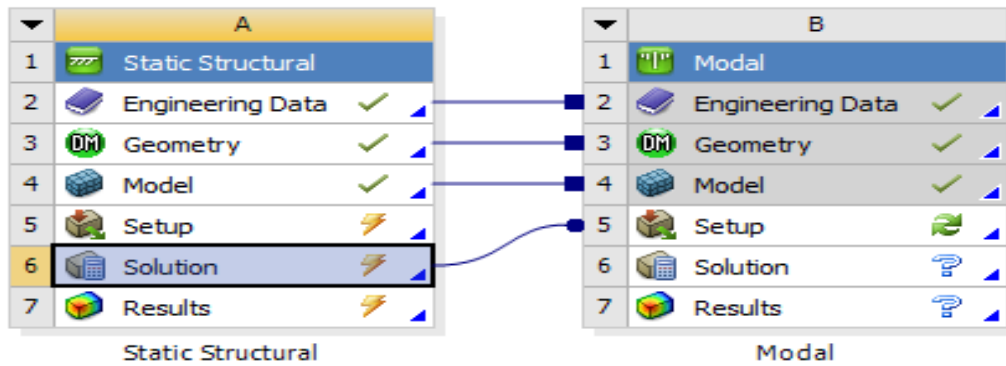


Figure 4-10: Transferring solution static structural to modal

4.4.1 The following bar chart indicates the frequency at each calculated mode:

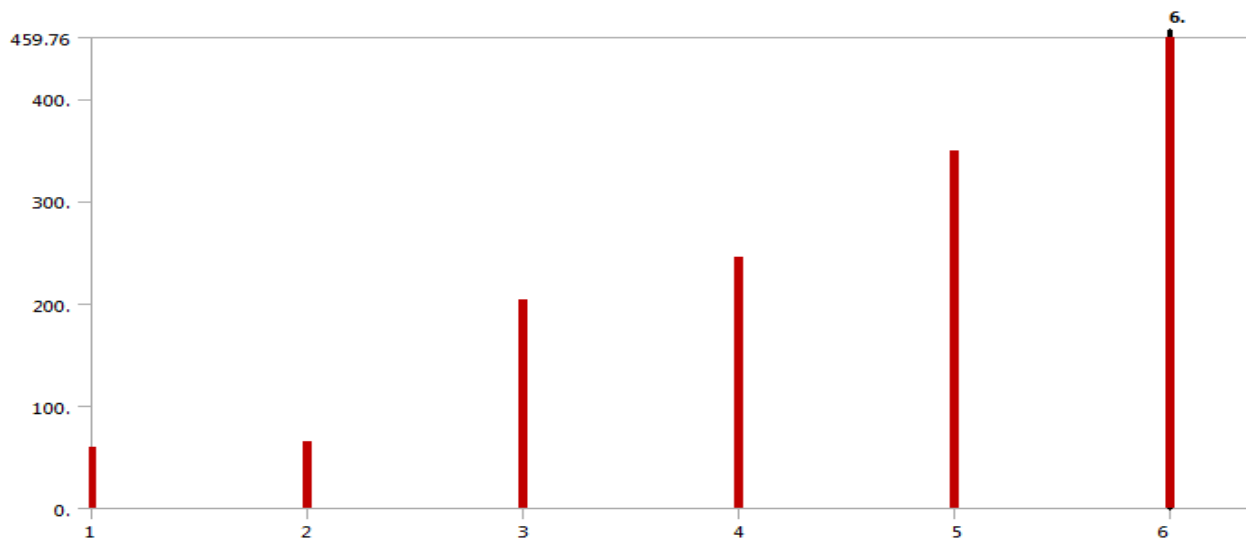


Figure 4-11: Frequency of shock absorber for existing material

The figure 4-11; Shows that the minimum and maximum frequency of the material at six nodes.

Table 4-2: Model > Modal > Solution

Mode	Frequency[Hz]
1	60.1
2	64.9
3	203.8
4	244.5
5	349.0
6	459.7

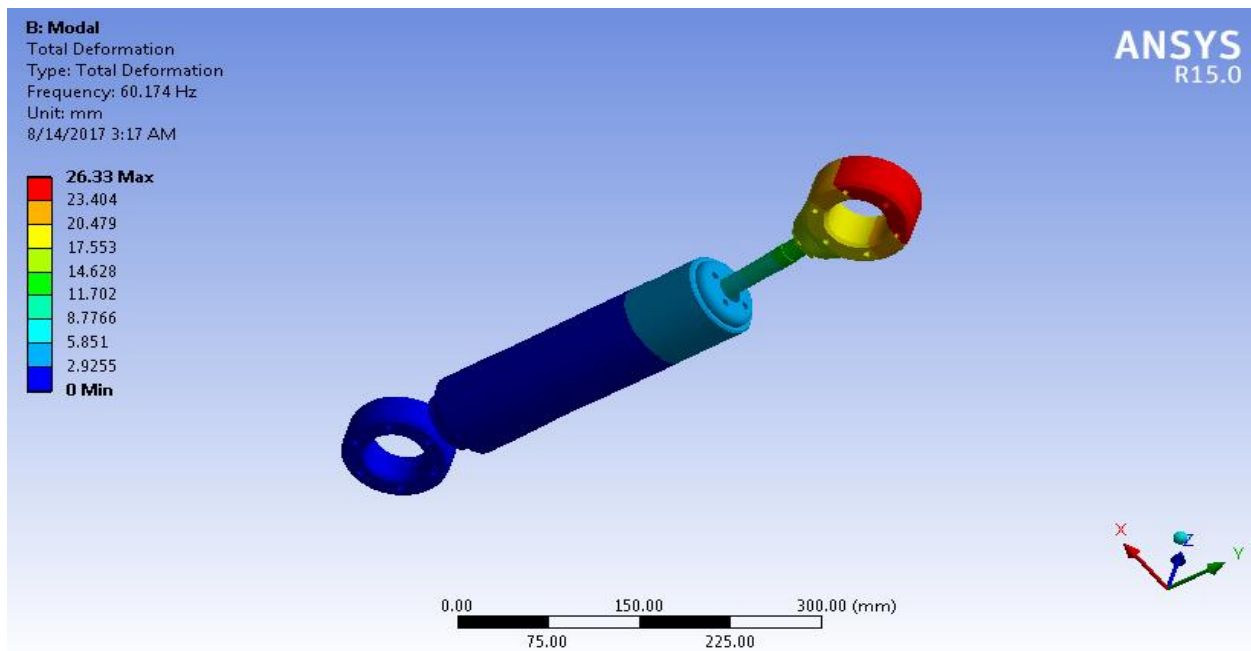


Figure 4-12: First Mode Shape

- ✚ The maximum value (maximum deformation) is indicated with red color which is 26.33mm and blue color indicates the minimum value as shown in figure 4-12.

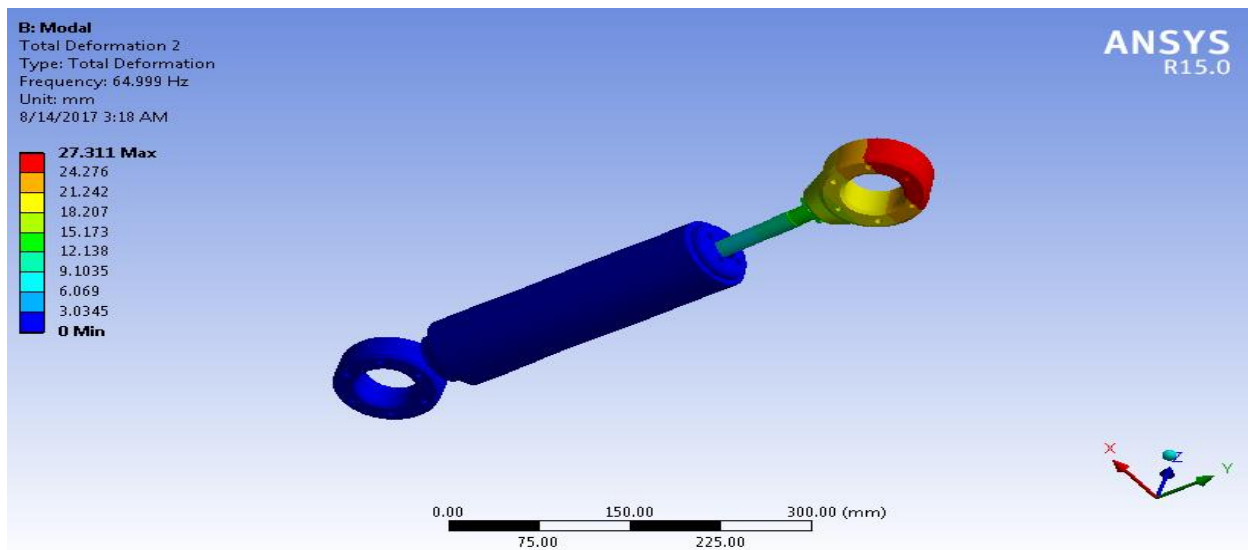


Figure 4-13: Second Mode Shape

- The maximum value (maximum deformation) is indicated with red color which is 27.311 mm and blue color indicates the minimum value as shown in figure 4-13.

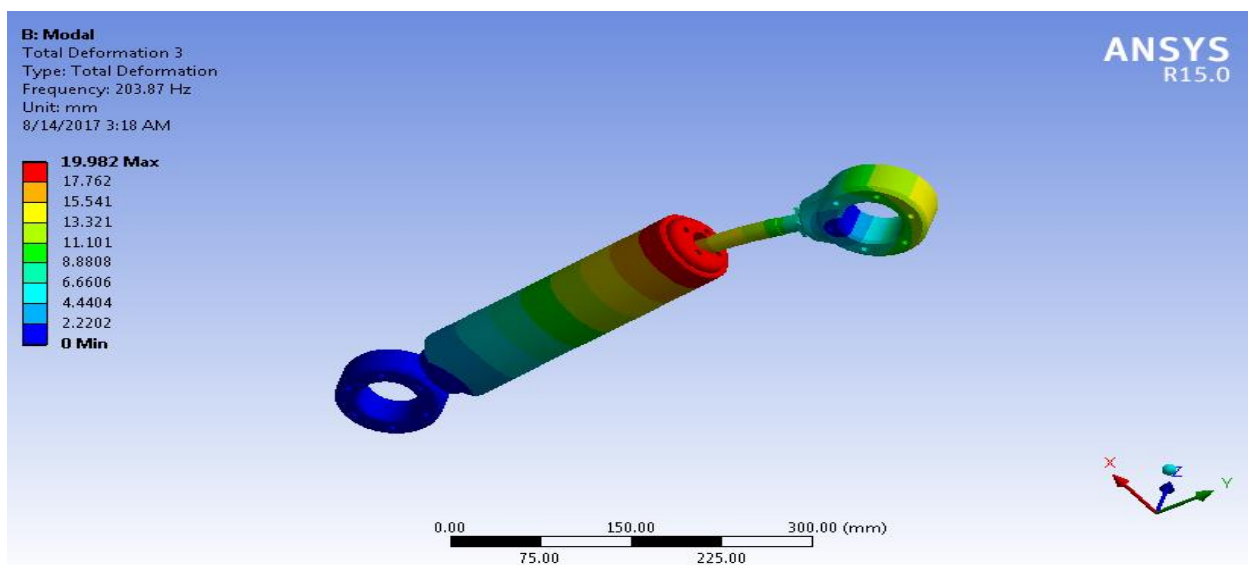


Figure 4-14: Third Mode Shape

- The maximum value (maximum deformation) is indicated with red color which is 19.9 mm and blue color indicates the minimum value as shown in figure 4-14.

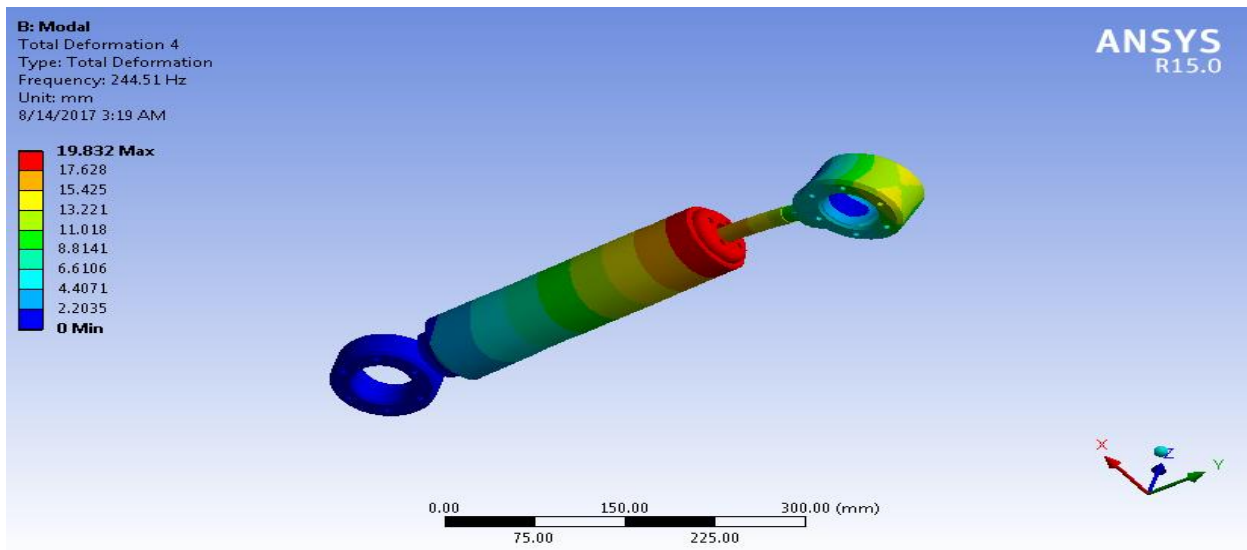


Figure 4-15: Fourth Mode Shape

- The maximum value (maximum deformation) is indicated with red color which is 19.8 mm and blue color indicates the minimum value as shown in figure 4-15.

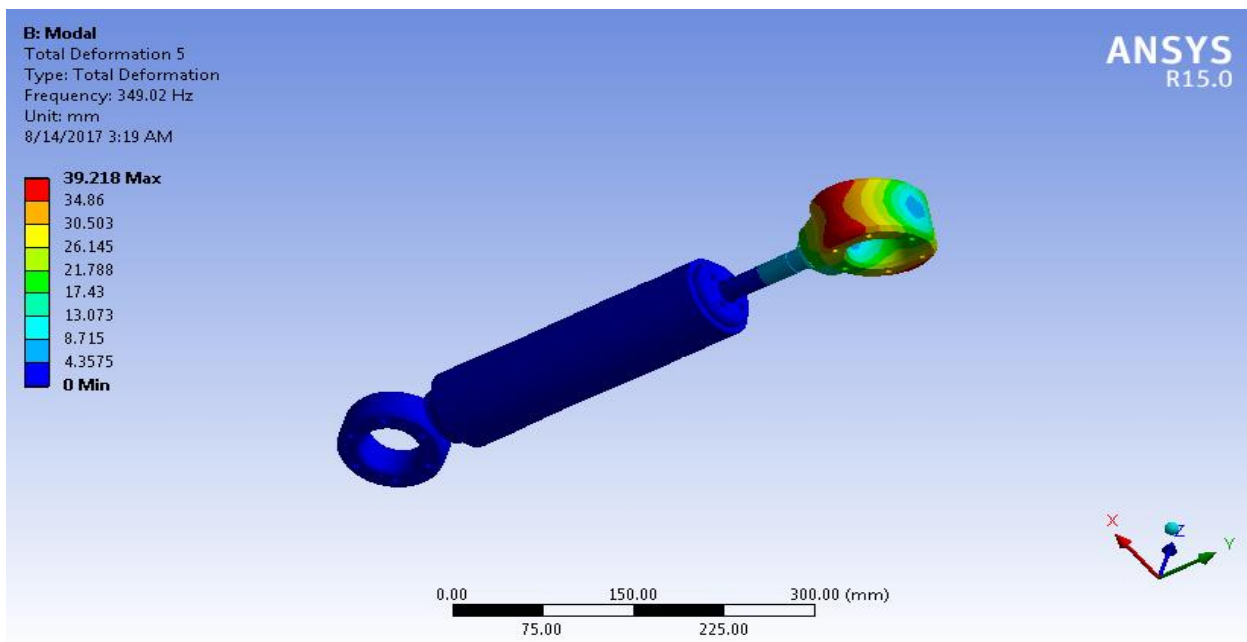


Figure 4-16: Fifth Mode Shape

- The maximum value (maximum deformation) is indicated with red color which is 39.21mm and blue color indicates the minimum value as shown in figure 4-16.

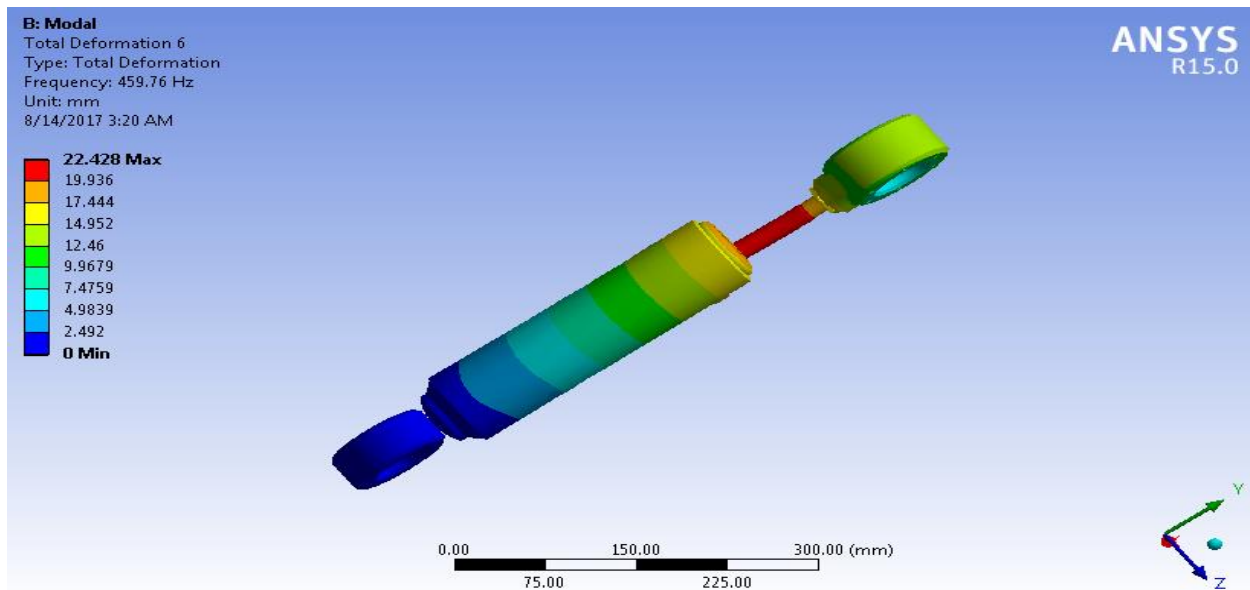


Figure 4-17: Sixth Mode Shape

- ✚ The maximum value (maximum deformation) is indicated with red color which is 22.428 mm and blue color indicates the minimum value as shown in figure 4-17.

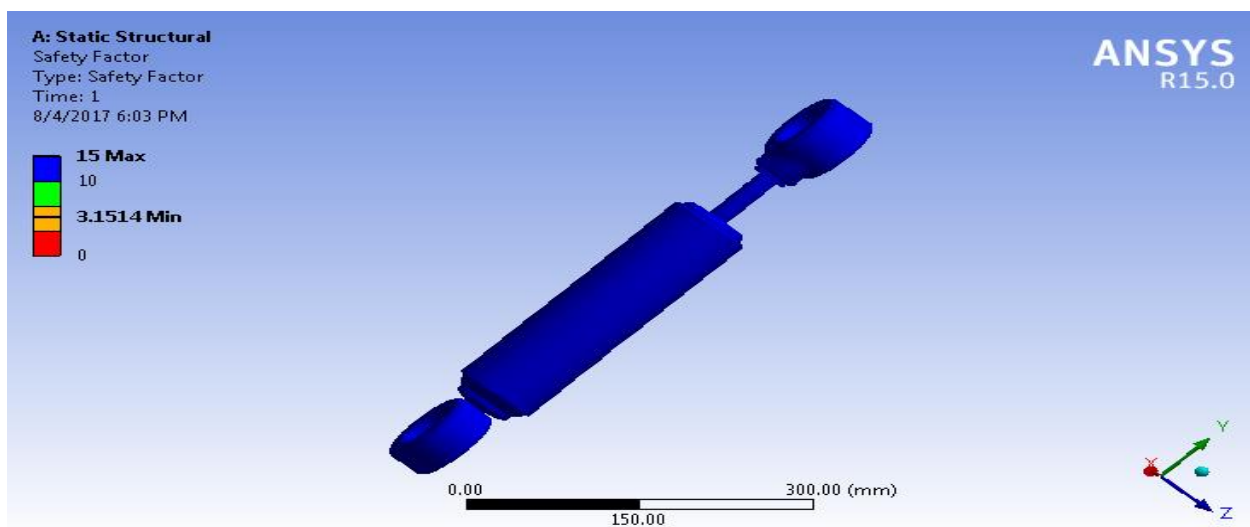


Figure 4-18: Factor of safety

- ✚ Figure 4-18; shows that the factor of safety that taken for analysis that is 3.1.

4.5 Structural analysis of modified shock absorber model

4.5.1 Static Structural Analysis

- ✓ Considering the weight of the vehicle and weight of (When $W=5542\text{ N}$)
- ✓ Material for spring :(ASTM A228 [HIGH CARBON SPRING WIRE])
- ✓ Properties: Young's Modulus (EX): 20850000 Kpa
- ✓ Poisson's Ratio (PRXY): 0.31
- ✓ Density: 7860 Kg/m³

Step 1: Importing Shock Absorber Model in ANSYS workbench and assigning the Material (Material Properties)

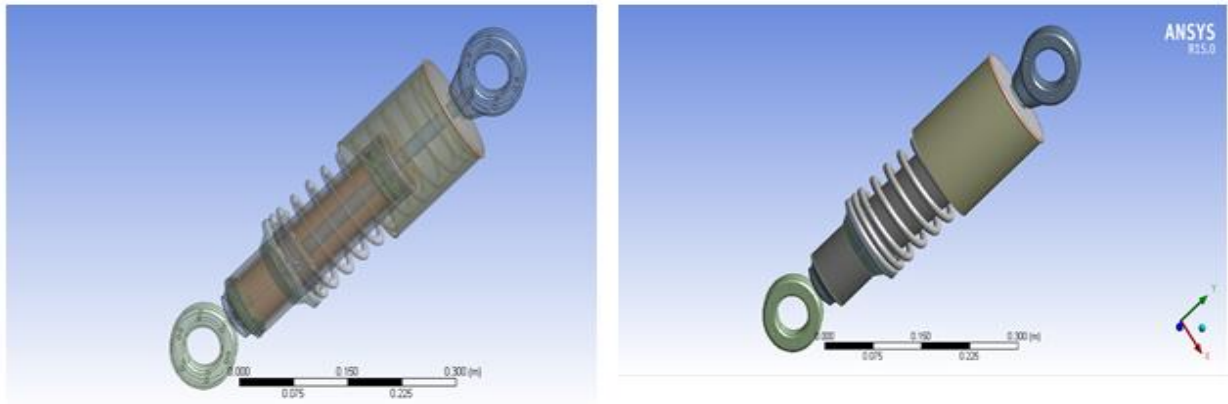


Figure 4-19: Shock absorber in Add frozen and in Add material

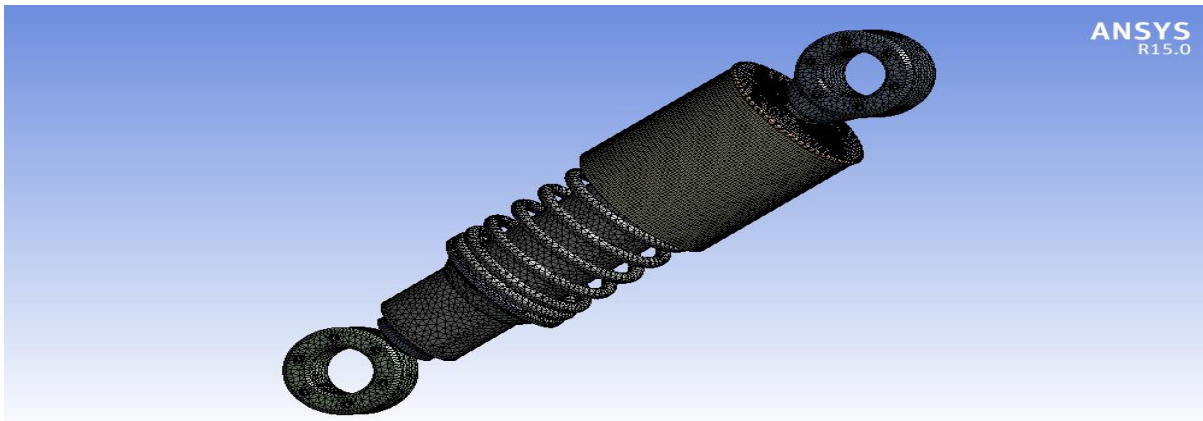


Figure 4-20: Applying the Mesh (Meshing)

Table 4-3: Meshing Statistics of the static structural analysis

Number of nodes	94769
Number of element	44872
Element size(mm)	0.001
Analysis type	Static structural

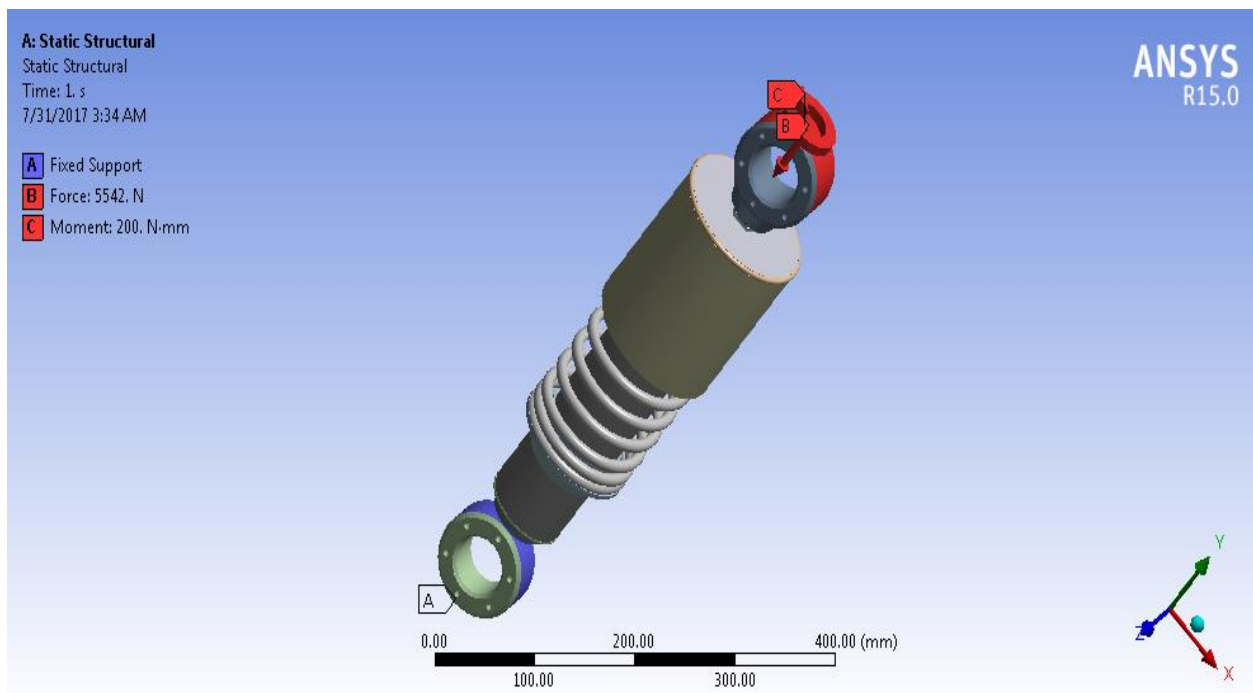


Figure 4-21: Applying the Loads, moment and Boundary Conditions

The figure 4-21, shows that the loading and boundary condition on a shock absorber, Also the boundary condition is applied at fixed position of a model which is consider as a support, the load is applied considering weight of vehicle and two and four person at top part of shock absorber model.

4.5.2 Obtaining results after applying loads

Figure 4-22, shows the total deflection in the shock absorber when the load of 5542N is applied for the existing shock absorber.

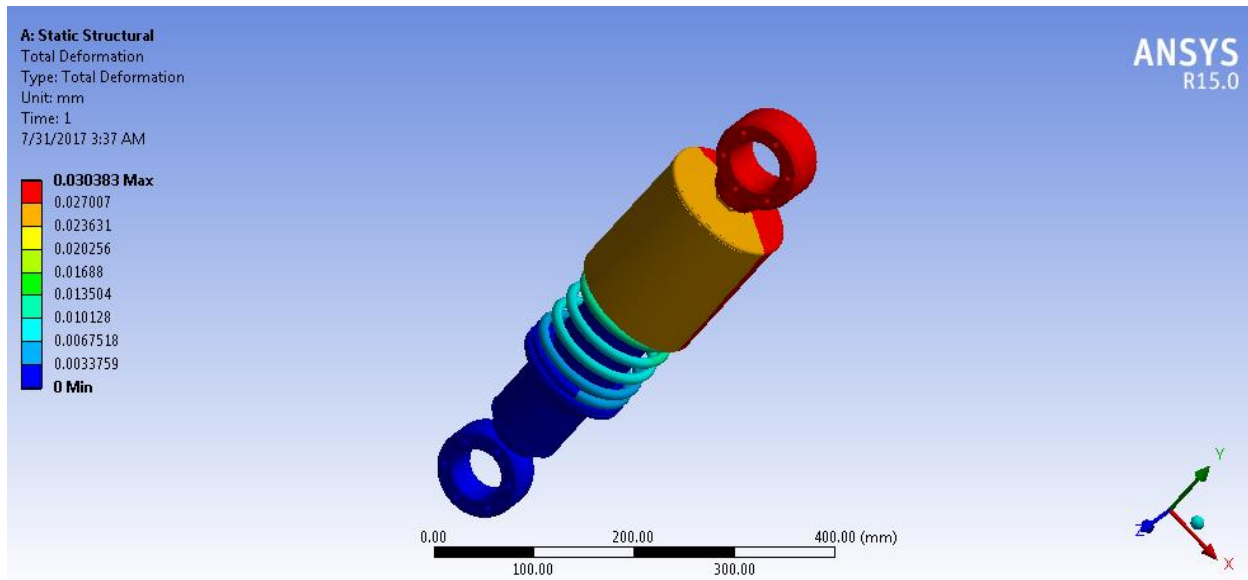


Figure 4-22: Total deformation for load of (W=5542N)

- The maximum value (maximum deformation) is indicated with red color which is 0.030383 mm and blue color indicates the minimum value as shown in figure 4-22;

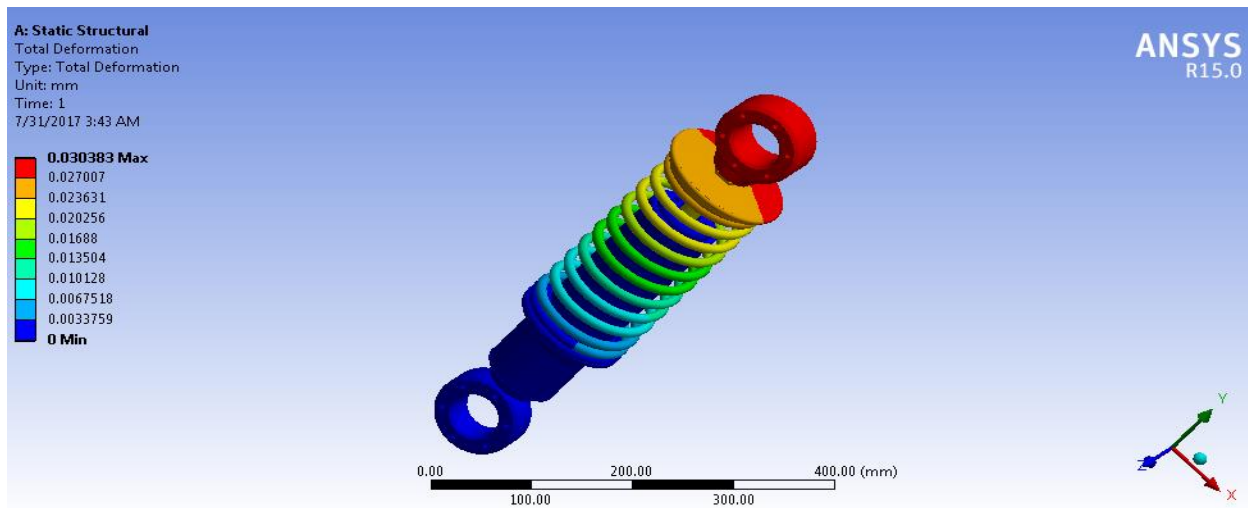


Figure 4-23: Total Deformation for load of (W= 5542N)

Figure 4-24, shows the total equivalent stress in the shock absorber when the load of 5542N is applied for the existing shock absorber.

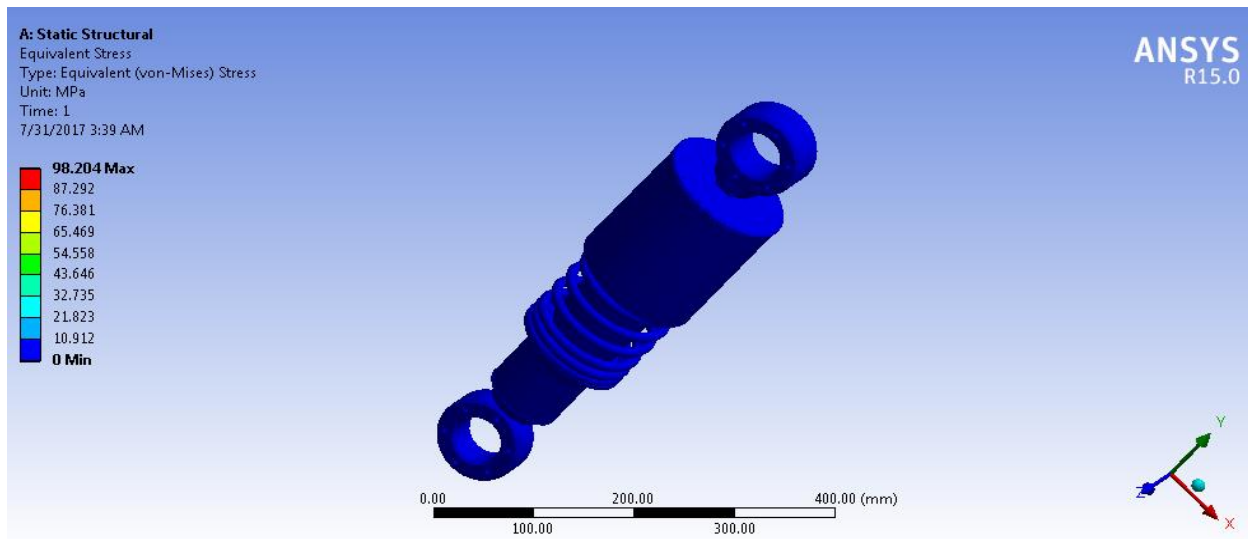


Figure 4-24: Equivalent (von- mises) stress for load of (W=5542N)

The maximum value (maximum equivalent stress(von-mises stress)) is indicated with red color which is 98.204 Mpa and blue color indicates the minimum value as shown in figure 4-24;

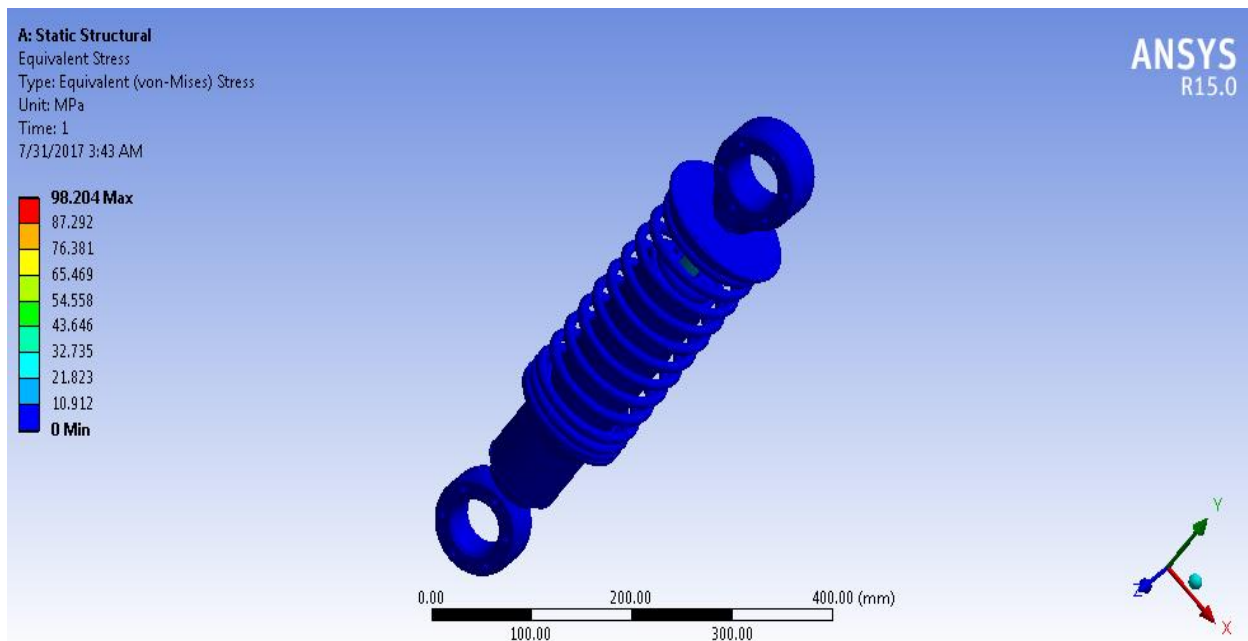


Figure 4-25: Equivalent (von- mises) stress for load of (W=5542N)

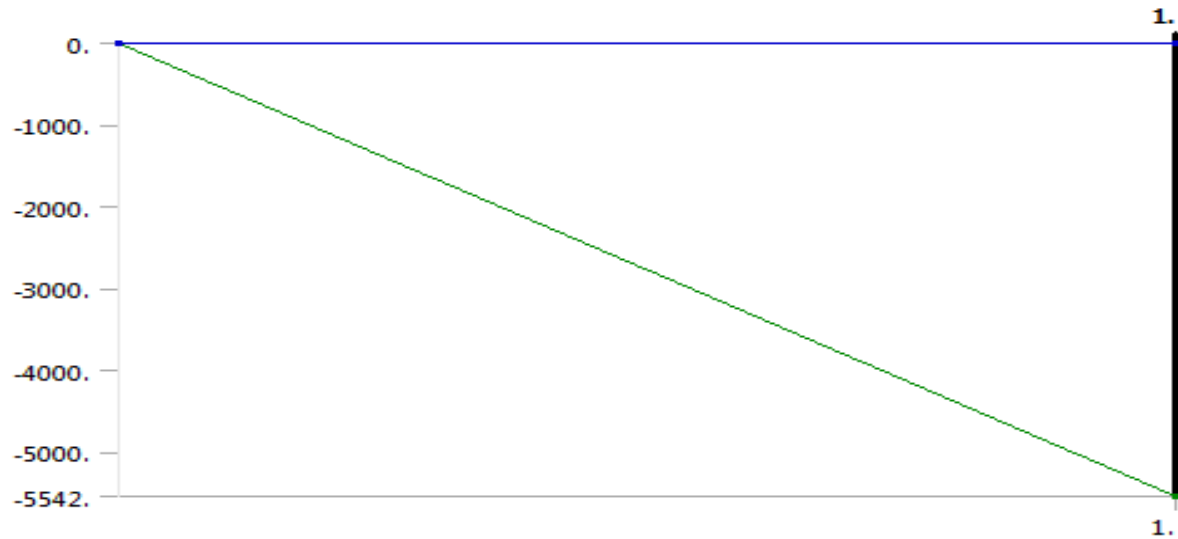


Figure 4-26: Force plot

- ✚ This figure 4-26; describes when applying the load ($W = 5542 \text{ N}$) the direction of the force in the y direction is negative direction ($W = -5542 \text{ N}$)
- ✓ Considering the weight of 4 persons (When $W = 6020 \text{ N}$)
- ✓ Material for spring :(ASTM A228 [HIGH CARBON SPRING WIRE])
- ✓ Properties: Young's Modulus (EX): 20850000 Kpa
- ✓ Poisson's Ratio (PRXY): 0.31
- ✓ Density: 7860 Kg/m^3

Step 1: Importing Shock Absorber Model in ANSYS Workbench and Assigning the Material

(Material Properties)

4.5.3 Applying the Loads and Boundary Conditions:

The following figure shows that the loading and boundary condition on a shock absorber, this load is applied according to the design of spring, also the boundary condition is applied at fixed position of a model which is considered as a support, and the load is applied considering weight of four persons at top part of shock absorber model.

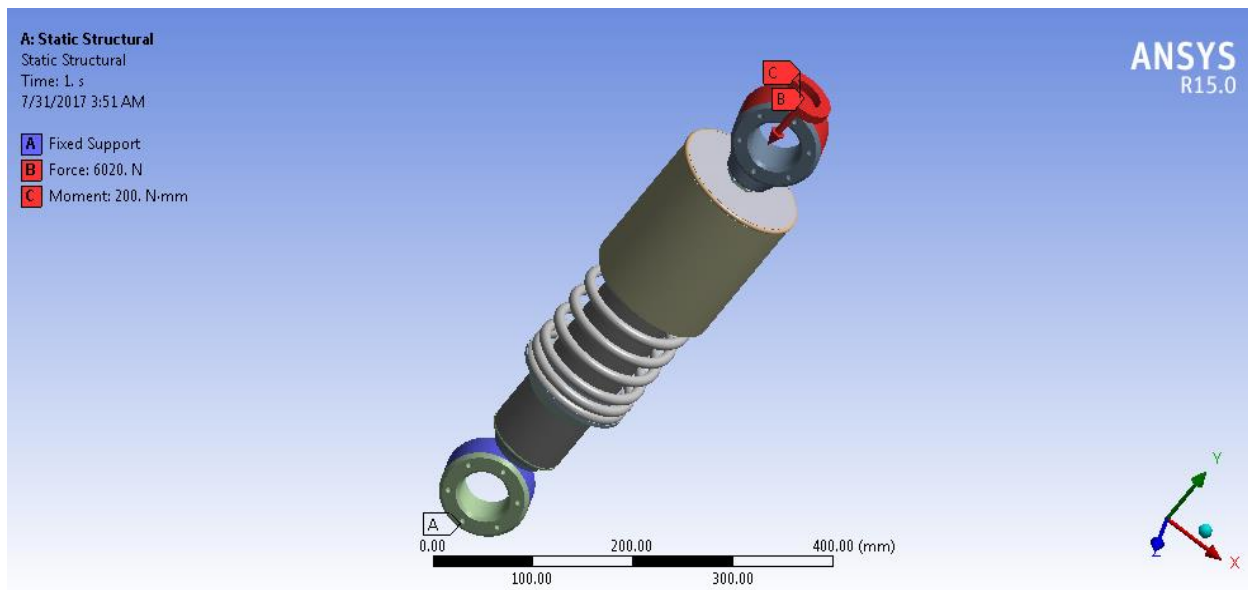


Figure 4-27: Applying loads

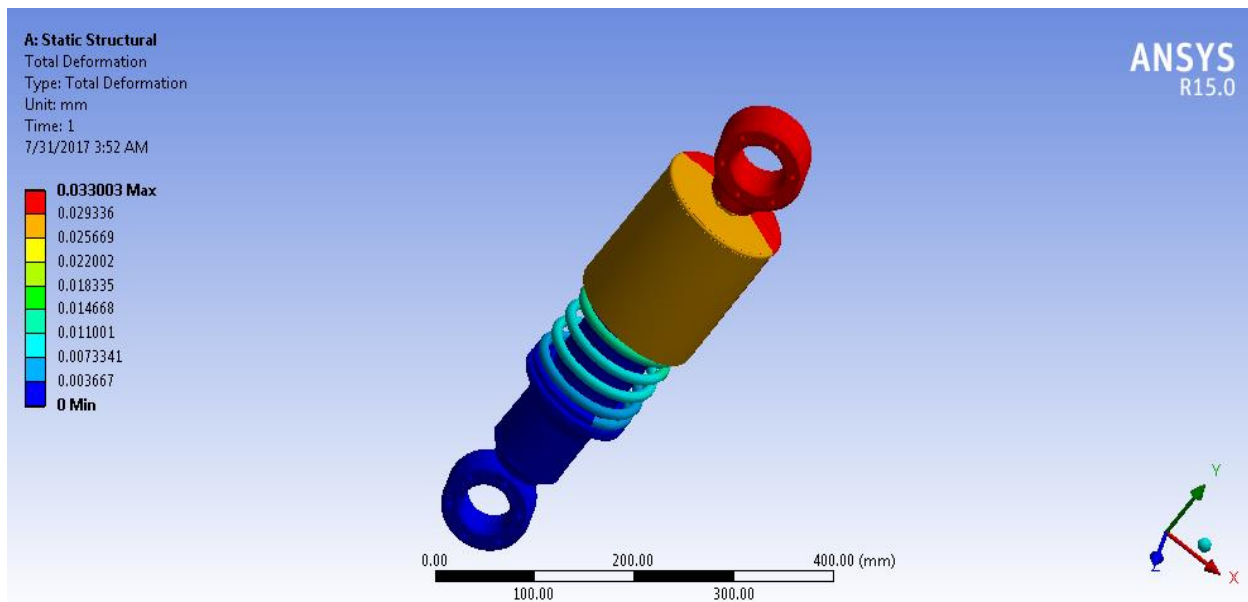


Figure 4-28: Total deformation for load of (W=6020N)

- The maximum value (maximum deformation) is indicated with red color which is 0.033003 mm and blue color indicates the minimum value as shown in figure 4-28;

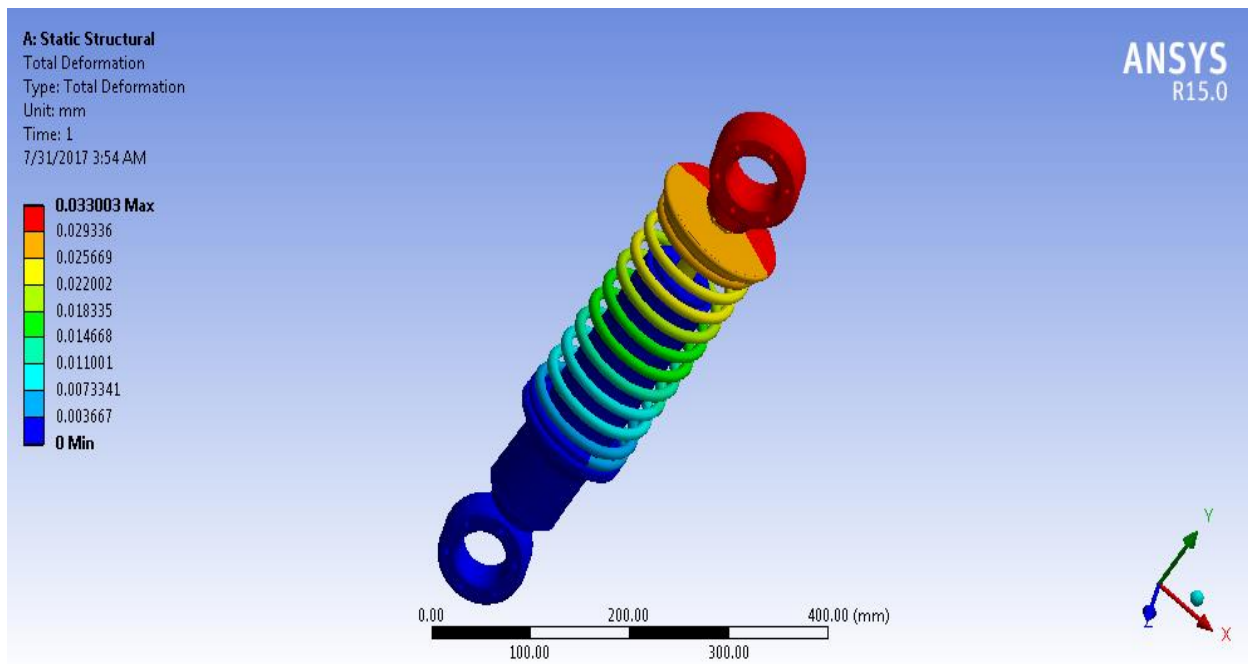


Figure 4-29: Total Deflection for load of (W= 6020N).

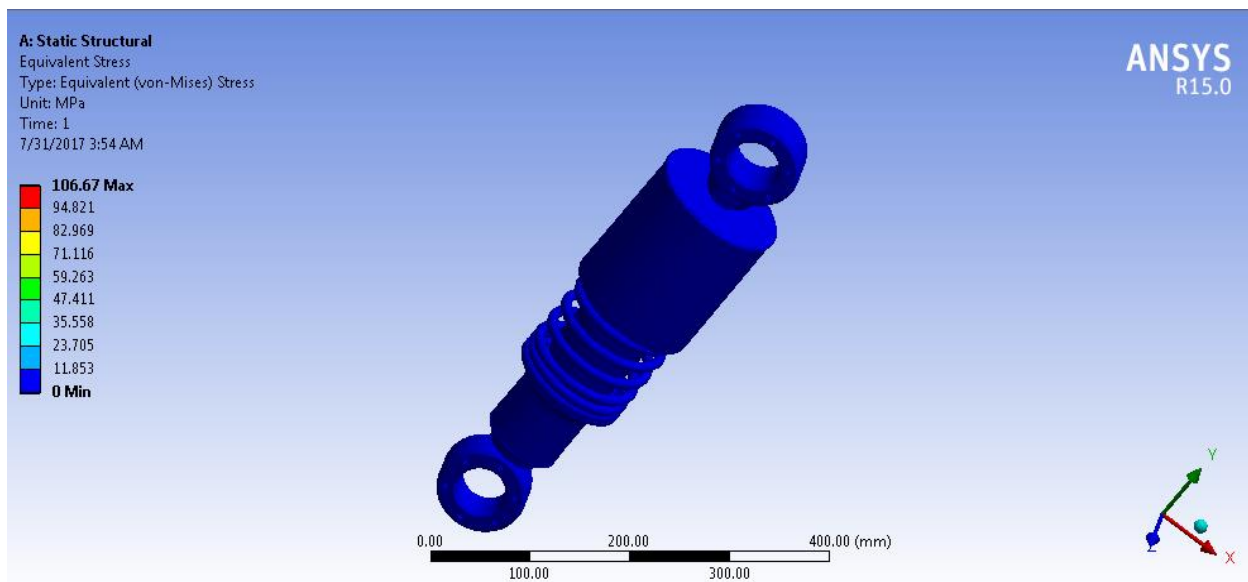


Figure 4-30: Equivalent (von- mises) stress when the load of (W=6020N) Applied

- ✚ The maximum value (maximum equivalent stress(von-mises stress)) is indicated with red color which is 106.67 MPa and blue color indicates the minimum value as shown in figure 4-30;

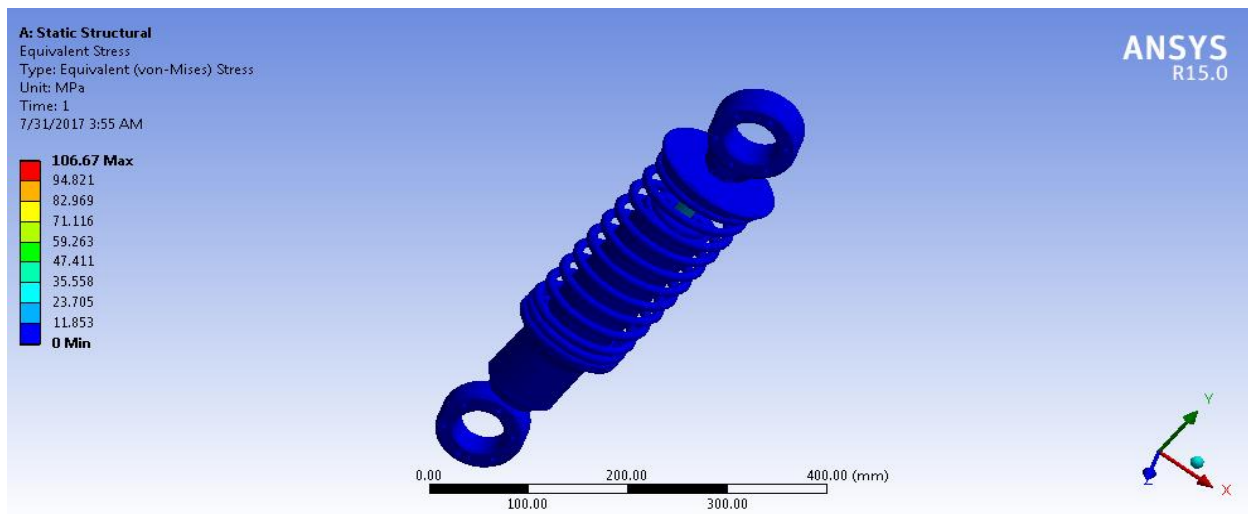


Figure 4-31: Equivalent (von- mises) stress when the load of ($W=6020\text{N}$) Applied

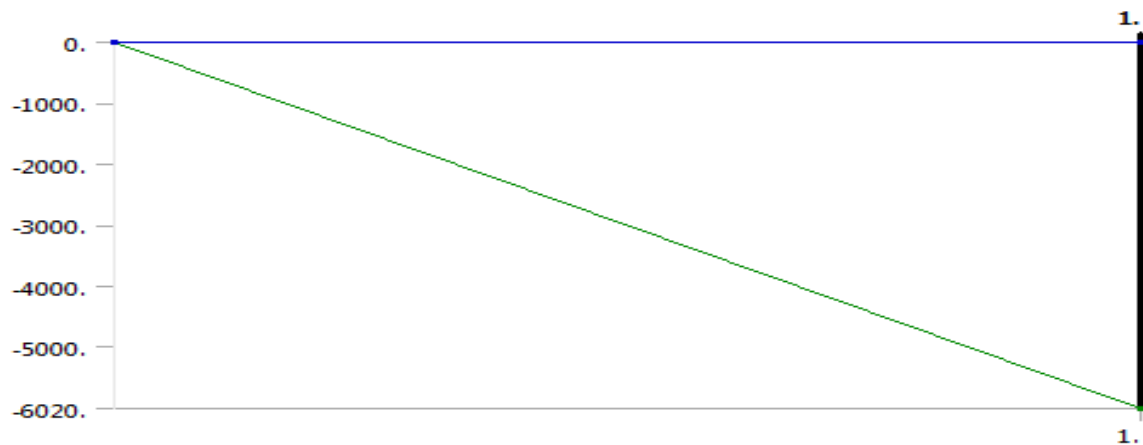


Figure 4-32: Force plot

✚ This figure 4-6; describes when applying the load ($W= 6020\text{ N}$) the direction of the force in the y direction is negative direction ($W= -6020\text{ N}$)

4.6 Modal Analysis for modified shock absorber

A modal analysis is typically used to determine the vibration characteristics (natural frequencies and mode shapes) of a structure or a machine component while it is being designed. It can also serve as a starting point for another, more detailed, dynamic analysis, such as a harmonic response or full transient dynamic analysis. Modal analyses, while being one of the most basic dynamic analysis types available in ANSYS, can also be more computationally time consuming than a typical static analysis. A reduced solver, utilizing automatically or manually selected master degrees of freedom is used to drastically reduce the problem size and solution time.

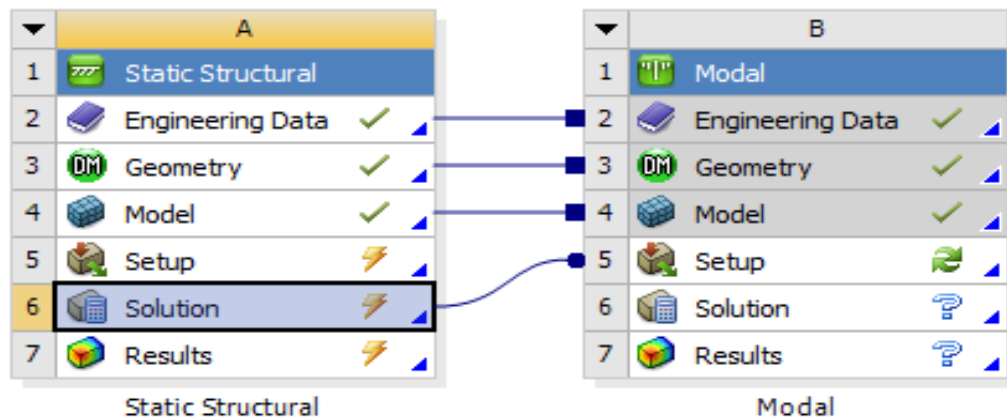


Figure 4-33: Transferring solution from structural analysis to modal

4.6.1 Modal Analysis Results for Existing shock absorber

- ✓ The following bar chart indicates the frequency at each calculated mode:

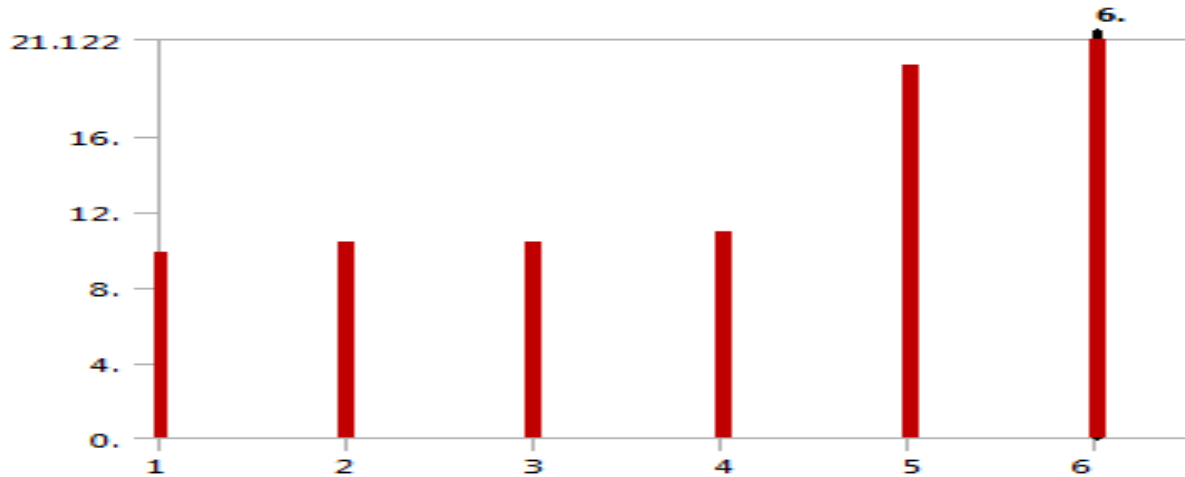


Figure 4-34: Frequency of shock absorber for existing material

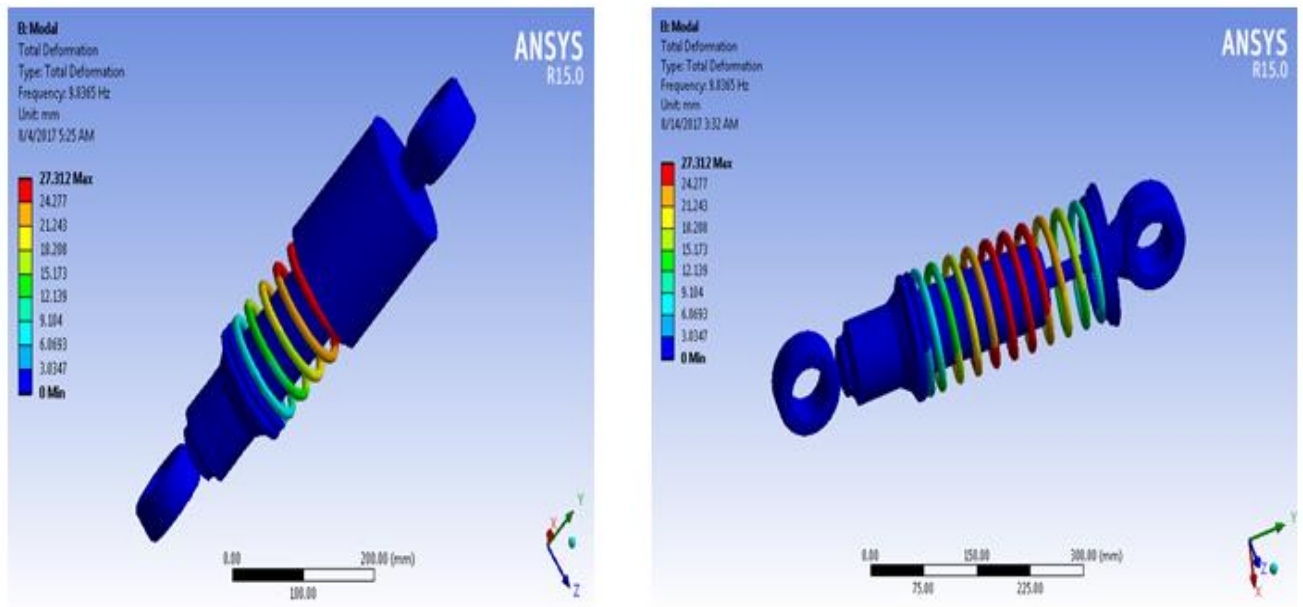


Figure 4-35: First Mode Shape

- ✚ The maximum value (maximum deformation) is indicated with red color which is 27.328 mm and blue color indicates the minimum value as shown in figure 4-35.

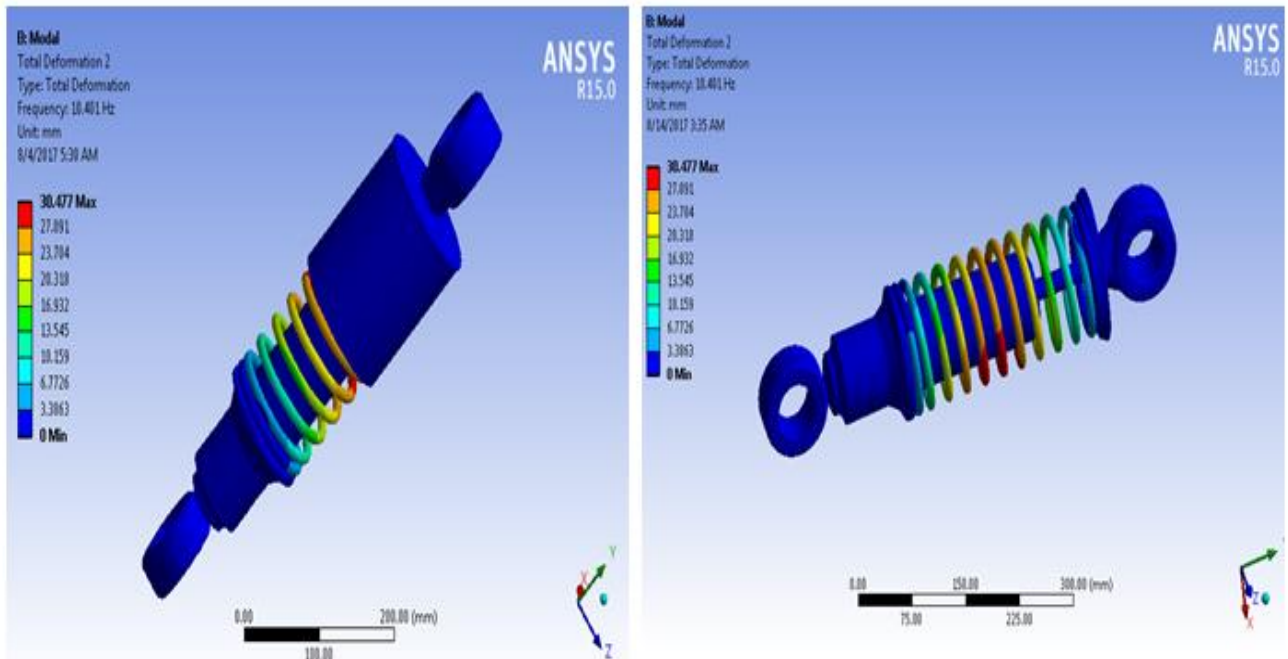


Figure 4-36: Second Mode Shape

- The maximum value (maximum deformation) is indicated with red color which is 30.47 mm and blue color indicates the minimum value as shown in figure 4-36;

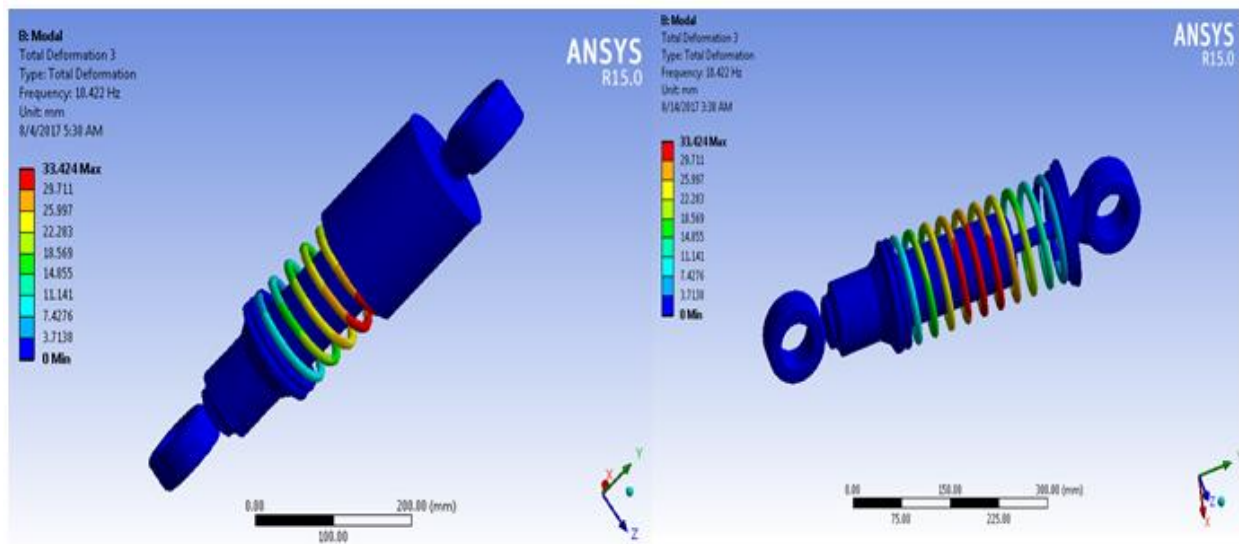


Figure 4-37: Third Mode Shape

- ✚ The maximum value (maximum deformation) is indicated with red color which is 33.428 mm and blue color indicates the minimum value as shown in figure 4-37;

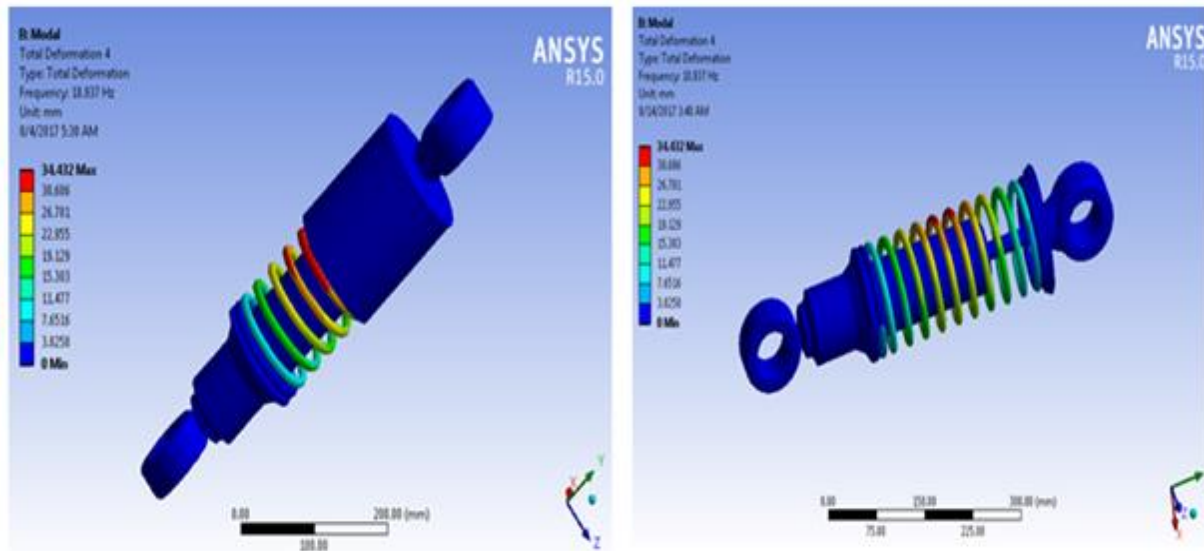


Figure 4-38: Fourth Mode Shape

- ✚ The maximum value (maximum deformation) is indicated with red color which is 34.4 mm and blue color indicates the minimum value as shown in figure 4-38;

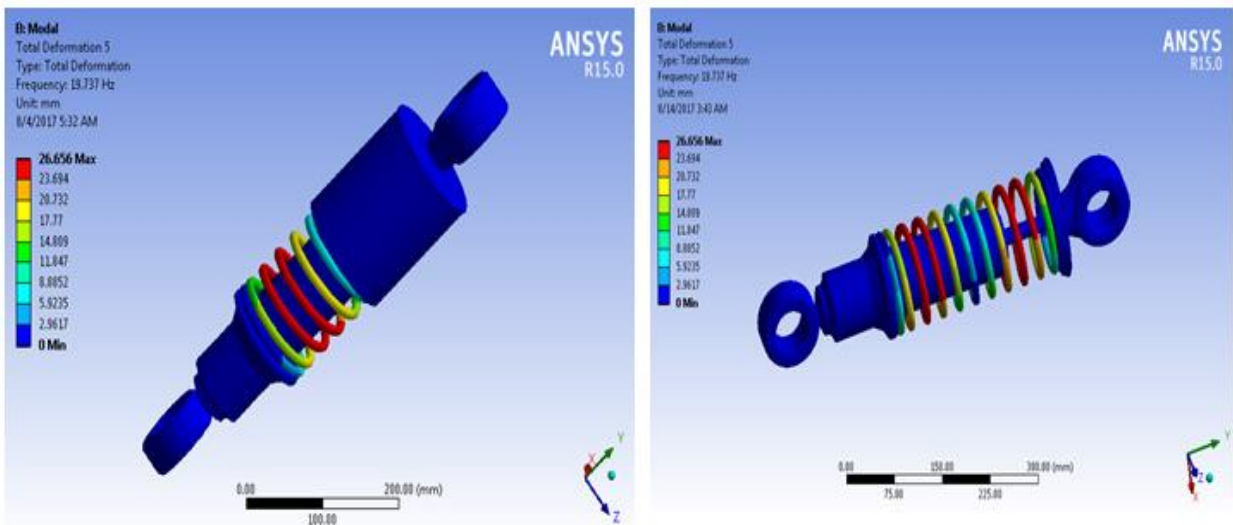


Figure 4-39: Fifth Mode Shape

- The maximum value (maximum deformation) is indicated with red color which is 26.65 mm and blue color indicates the minimum value as shown in figure 4-39.

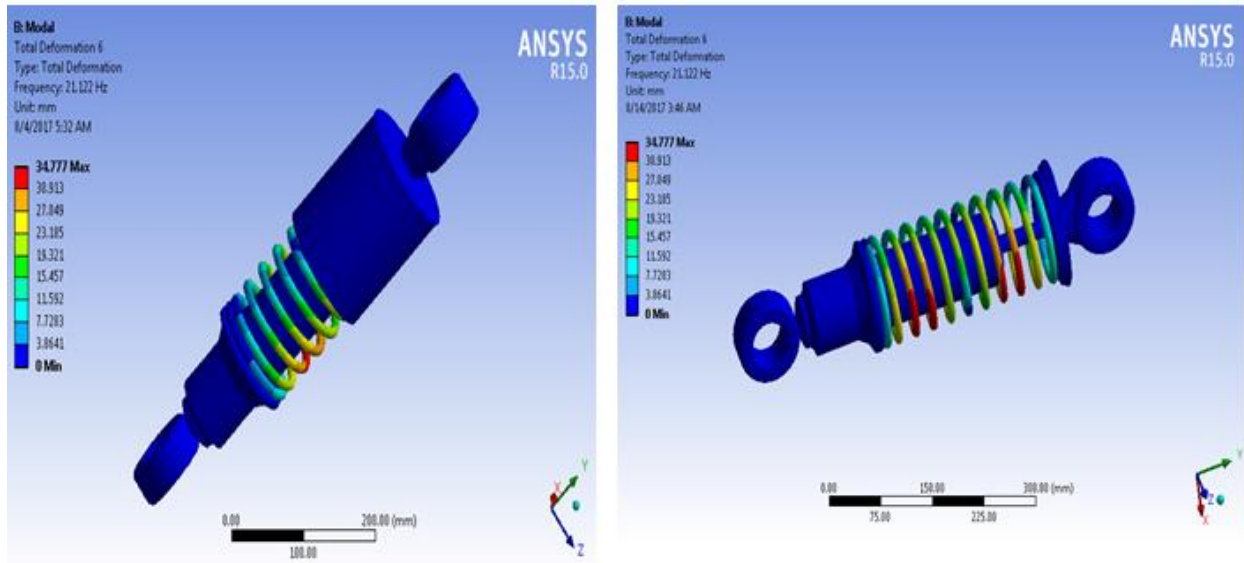


Figure 4-40: sixth Mode Shape

- The maximum value (maximum deformation) is indicated with red color which is 34.7 mm and blue color indicates the minimum value as shown in figure 4-40.

Table 4-4: Model > Modal > Solution

Mode	Frequency [Hz]
1	9.8
2	10.4
3	10.42
4	10.9
5	19.7

6	21.1
---	------

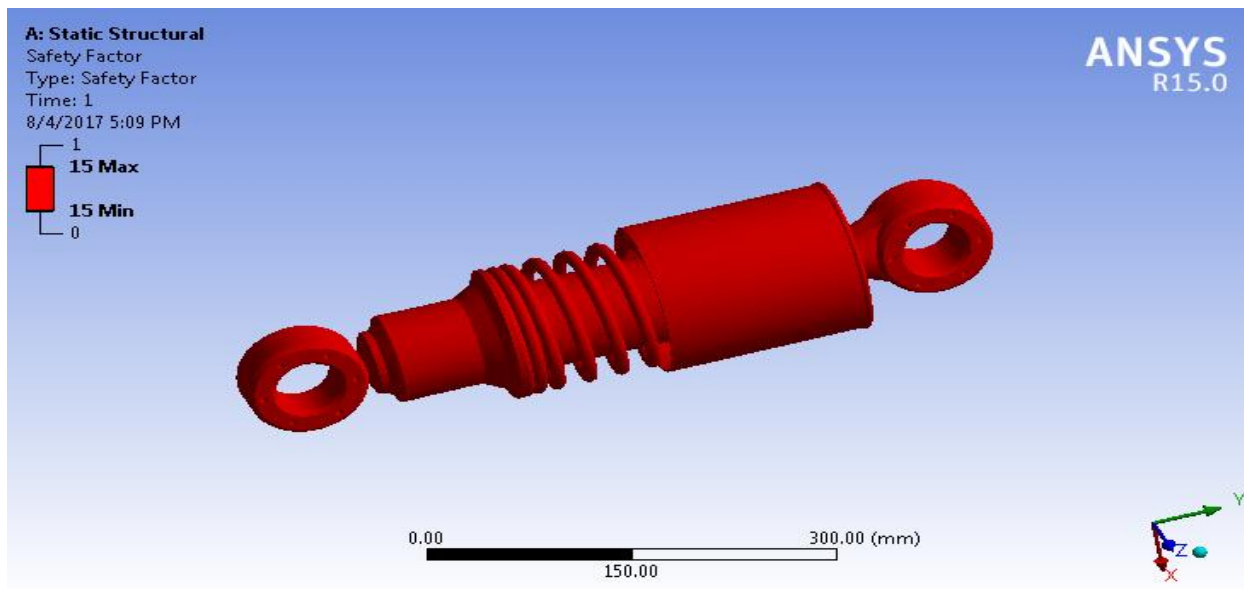


Figure 4-41: Safety factor

- In ANSYS the safety factor is always shown between 15 and -15. My design shows both maximum and minimum value as 15 with overall red colored model.
- In ANSYS if safety factor shows minimum and maximum value as 15 it mean that the design is safe.

4.7 Spring analysis

4.7.1 Spring analysis for 67sicr5 [Din 17221 Spring Steel Grade)) and (ASTM A228 [high carbon spring wire]

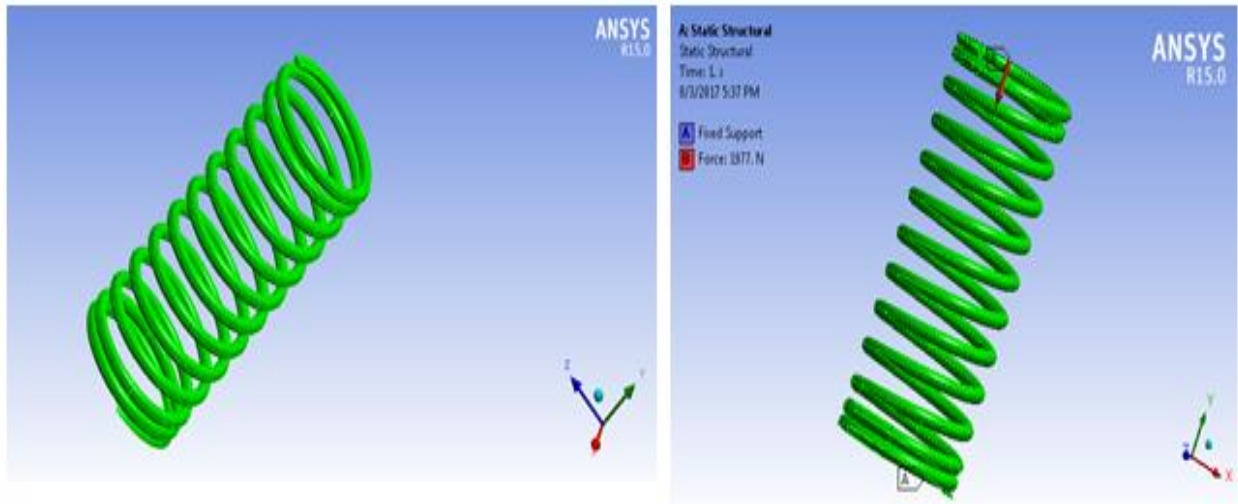


Figure 4-42: Spring model in ANSYS workbench and Applying load

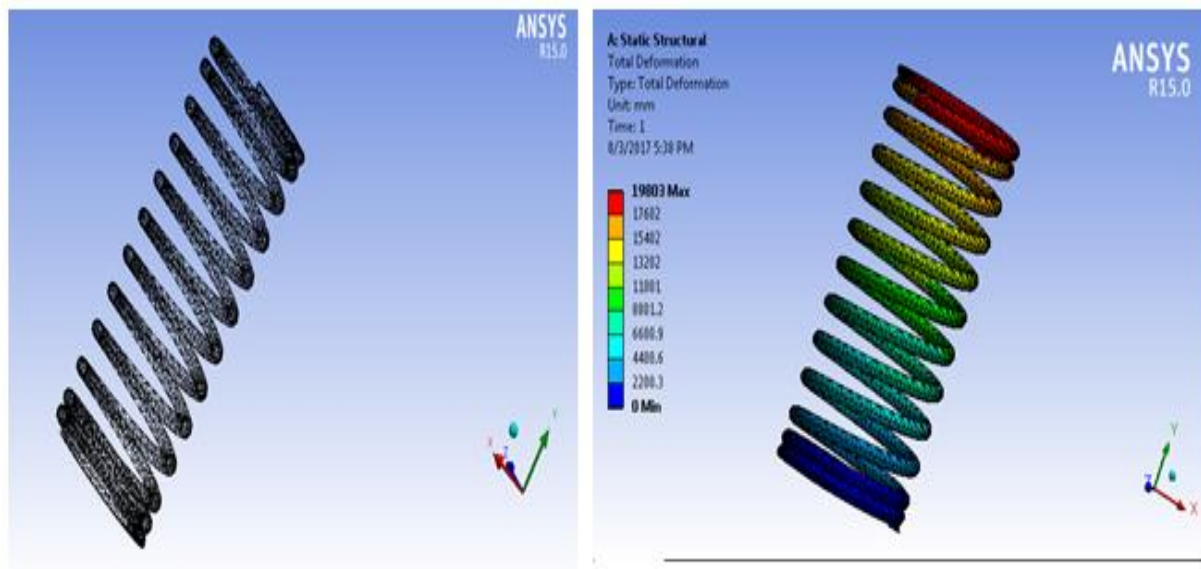


Figure 4-43: Meshing and Total deformation at (W= 1977N)

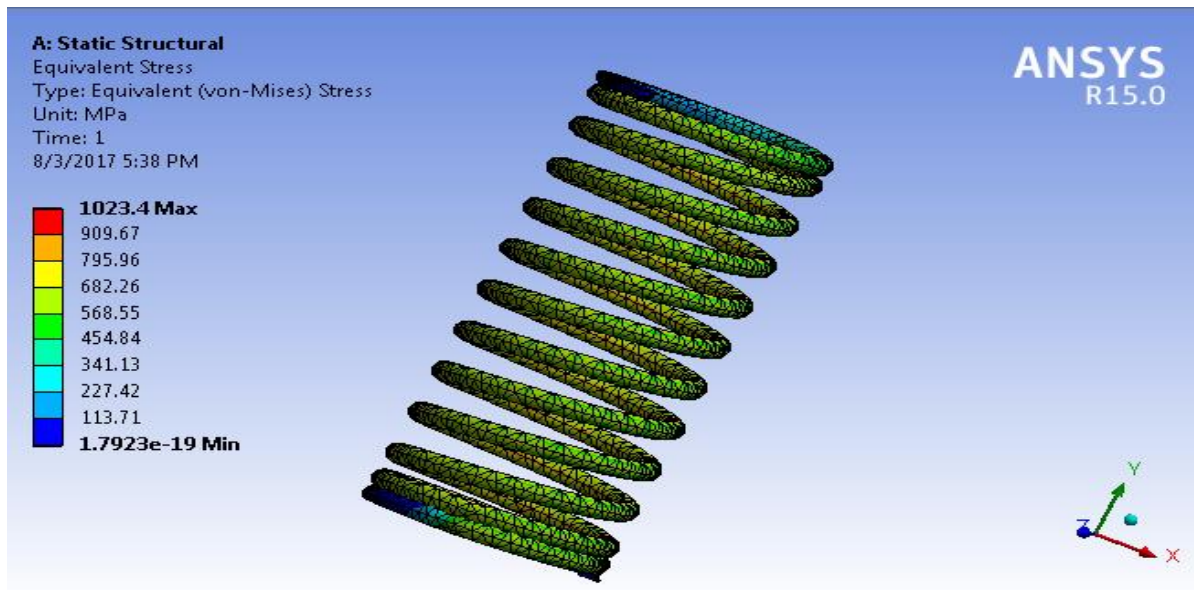


Figure 4-44: Equivalent (von-mises) stress at ($W= 1977N$)

4.7.2 (ASTM A228 [high carbon spring wire])

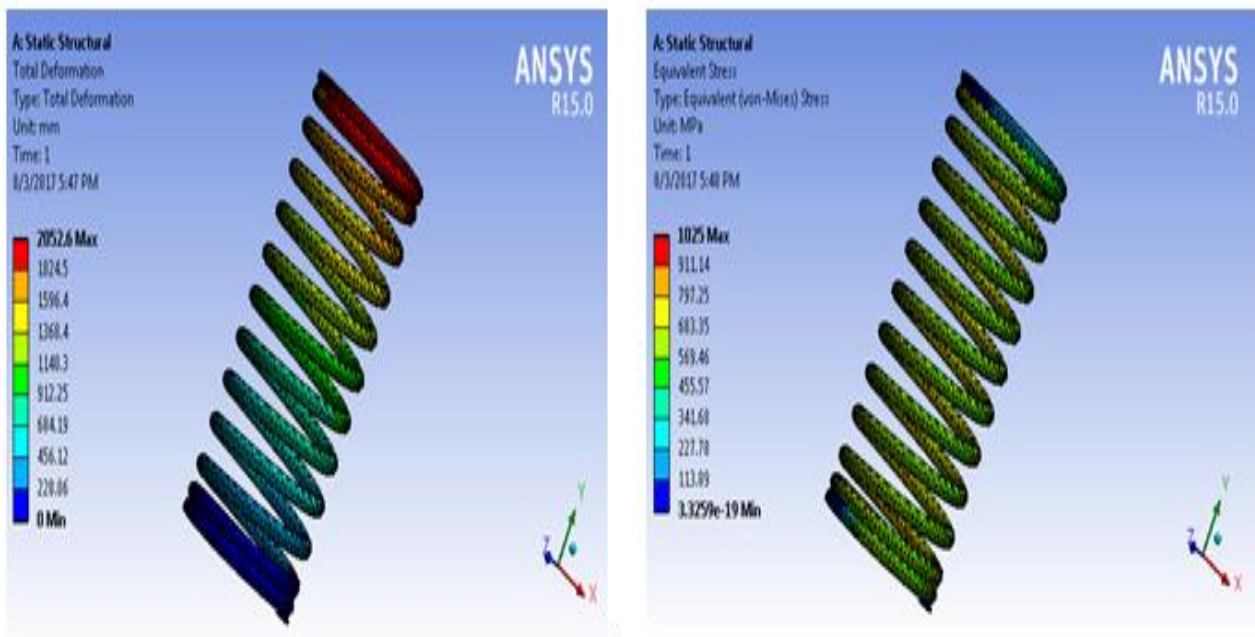


Figure 4-45: Total deformation and Equivalent (von-mises) stress at ($W=1977N$)

✚ It is very clear from the above obtained data that ASTM A228 [High carbon spring wire) has the minimum deformation at a particular load of 1977 N while the maximum shear stress is also less for ASTM A228 [High carbon spring wire).

4.8 Harmonic response for selected spring (ASTM A228 [high carbon spring wire])

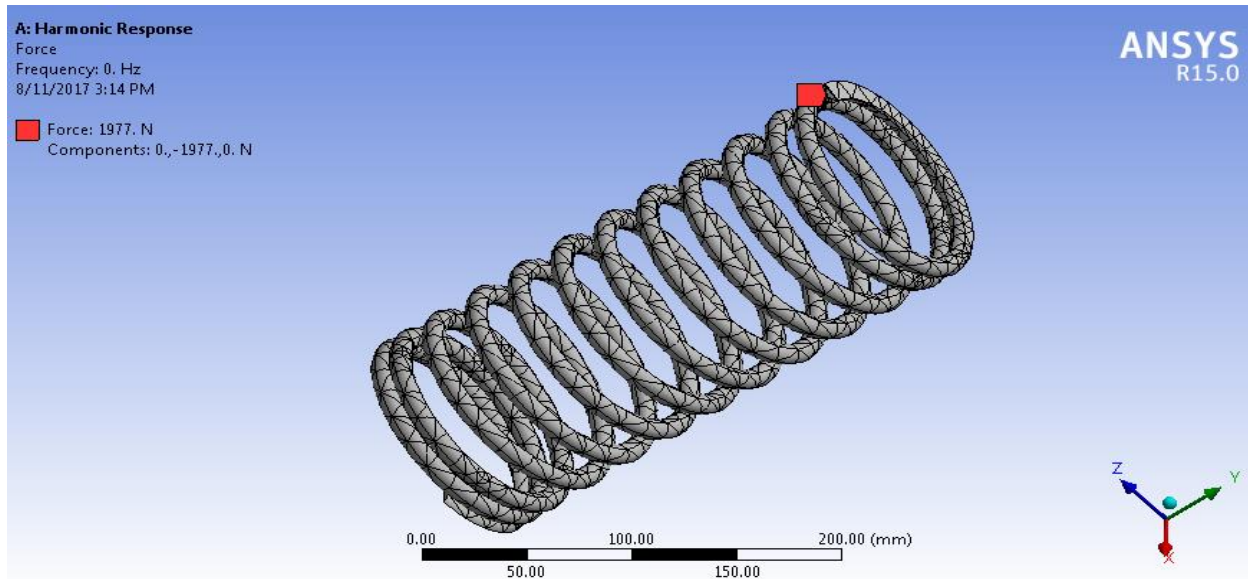


Figure 4-46: Applying load

4.8.1 Total deformation

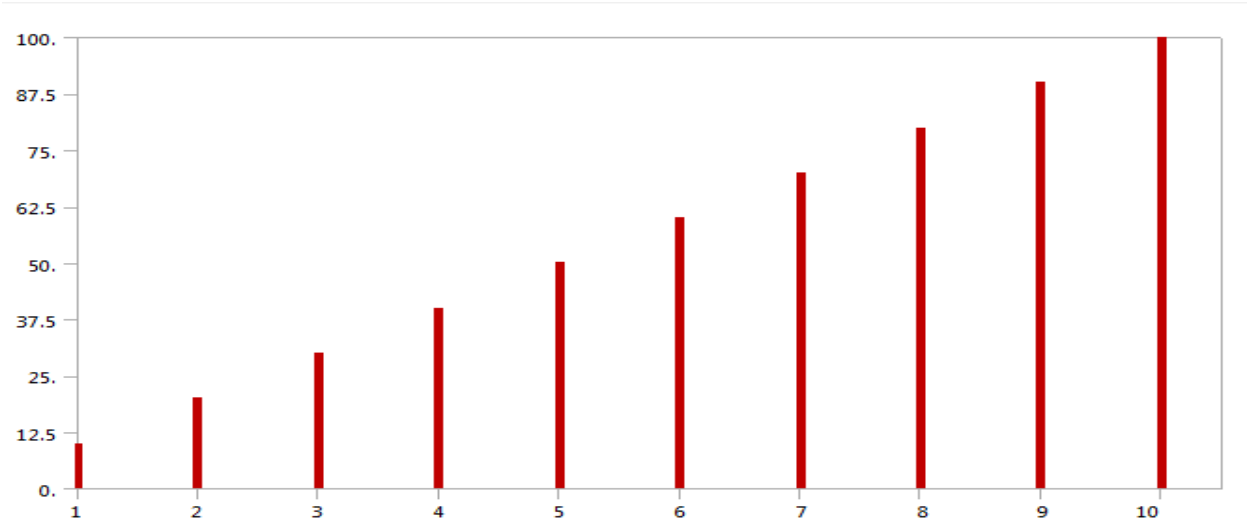


Figure 4-47 : Total deformation

Table 4-5: Frequency

	Set	Frequency[Hz]
1	1	10
2	2	20
3	3	30
4	4	40
5	5	50
6	6	60

7	7	70
8	8	80
9	9	90
10	10	100

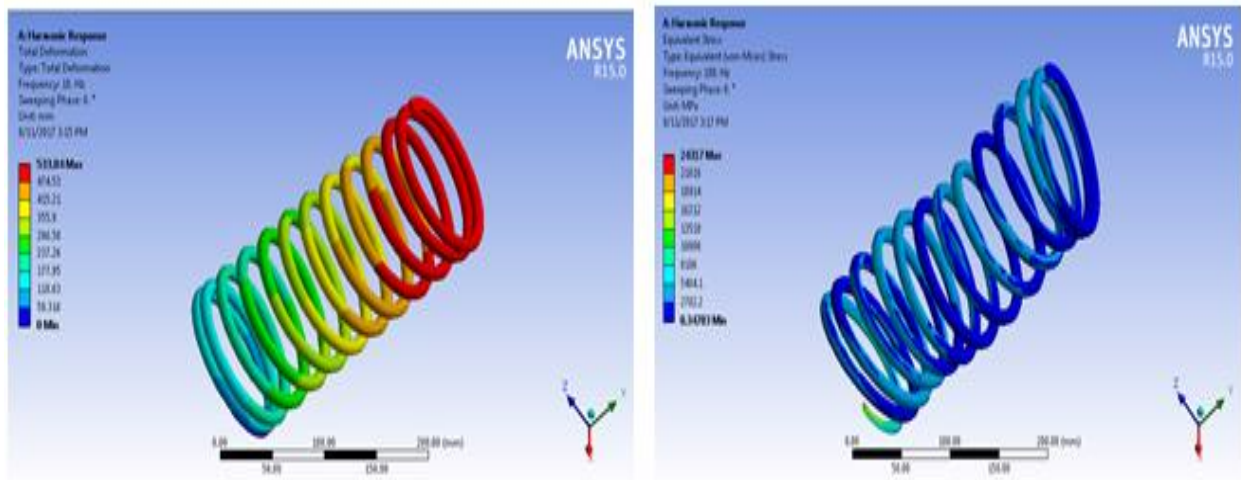


Figure 4-48: Total deformation and equivalence (von-mises),

4.8.2 Frequency response 1

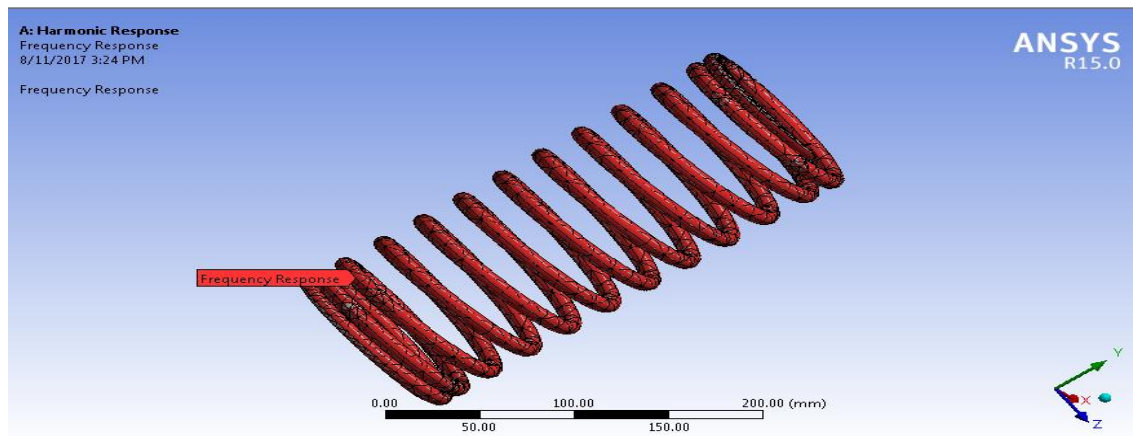


Figure 4-49: For stress frequency response

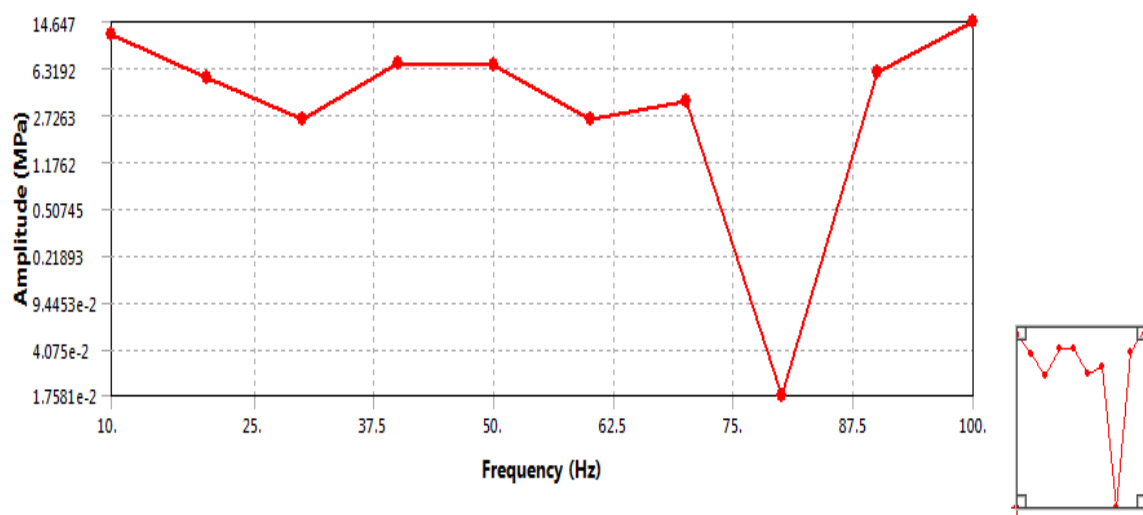


Figure 4-50: Amplitude vs frequency

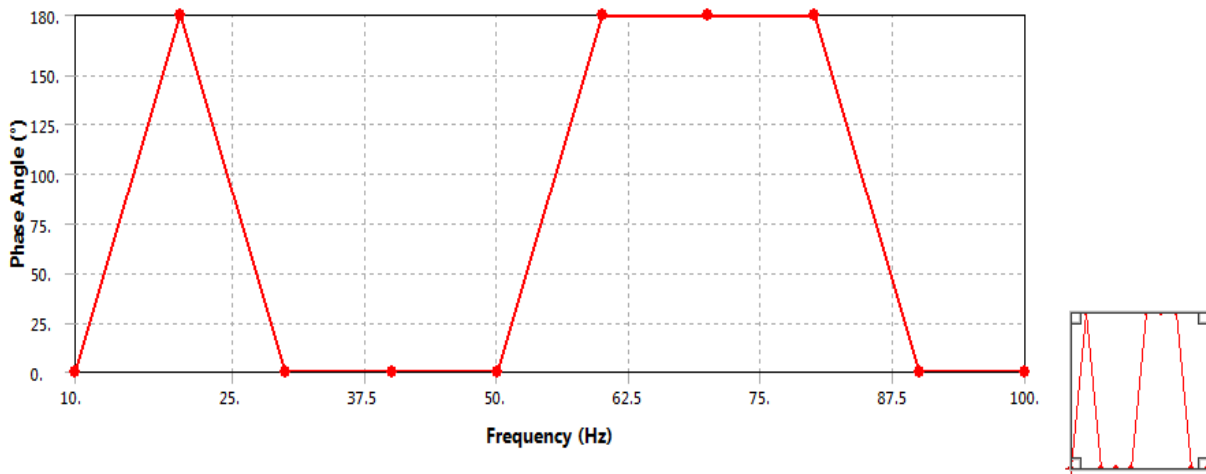


Figure 4-51: Phase angle vs frequency

Table 4-6: Frequency response 1

	Frequency [Hz]	Amplitude [MPa]	Phase angle[deg]
1	10	11.61	0
2	20	5.426	180
3	30	2.495	0
4	40	6.936	0
5	50	6.655	0
6	60	2.554	180
7	70	3.482	180
8	80	1.7581	180

9	90	5.993	0
10	100	14.647	0

4.8.3 Frequency response 2

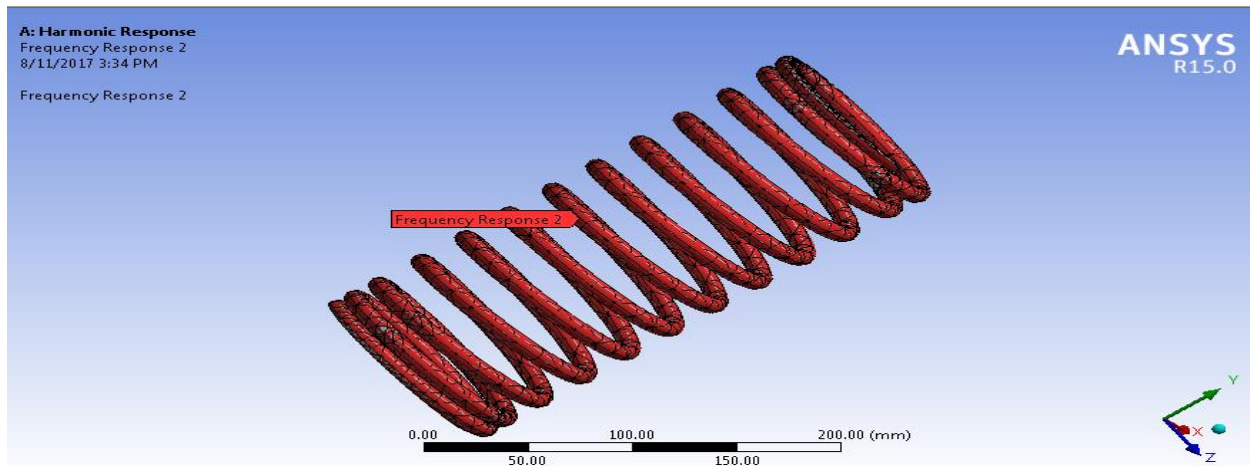


Figure 4-52: For deformation

4.8.4 Frequency response For deformation

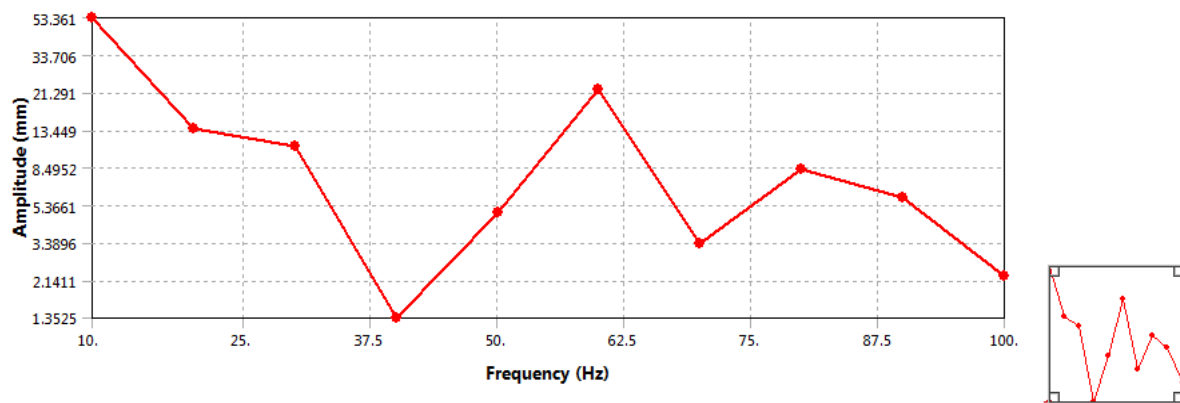


Figure 4-53: Amplitude vs frequency

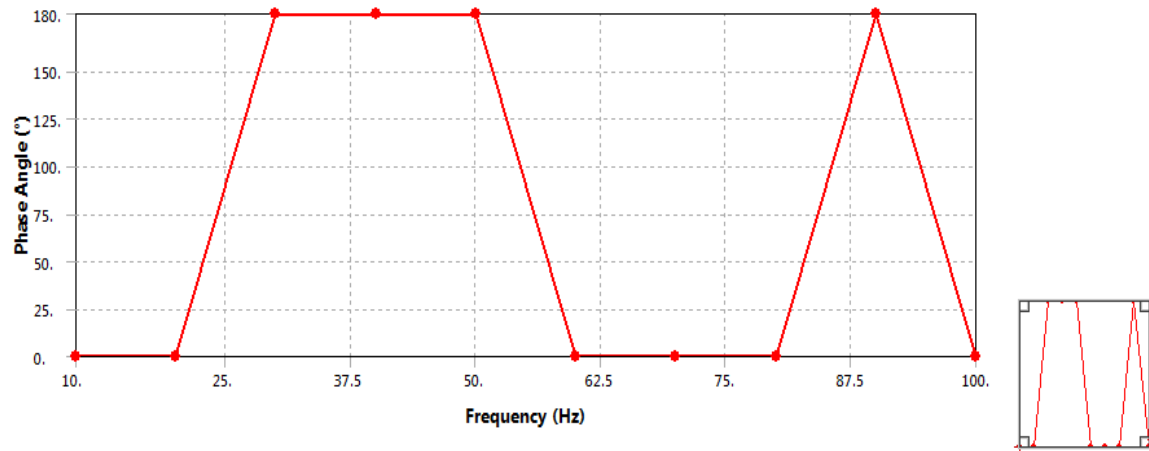


Figure 4-54: Phase angle vs frequency

Table 4-7: Frequency response 2

	Frequency [Hz]	Amplitude [MPa]	Phase angle[deg]
1	10	53.361	0
2	20	13.729	0
3	30	10.987	180
4	40	1.352	180
5	50	4.879	180
6	60	22.322	0
7	70	3.349	0
8	80	8.308	0
9	90	5.934	180

10	100	2.269	0
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4.9 BASE VALVE ANALYSIS

4.9.1 Modeling

A parametric 3D modelling software suite, CATIA was used to generate the 3D model of the valve assembly. The base valve consists of seven parts. The exploded view is shown by Figure 4.57.

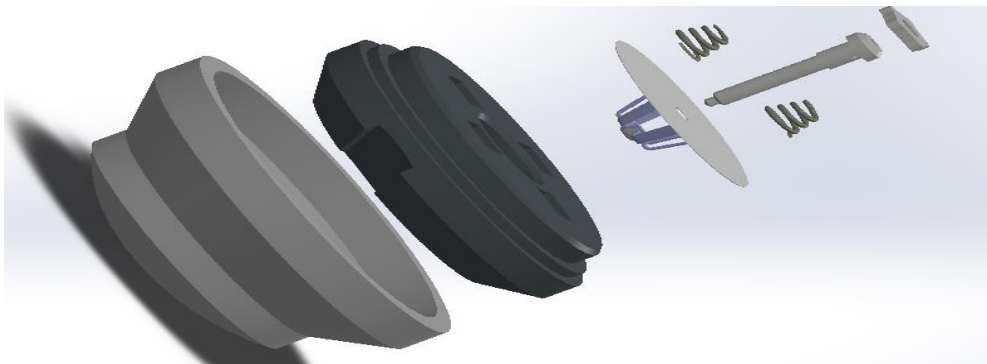


Figure 4-55: Exploded view of the assembly

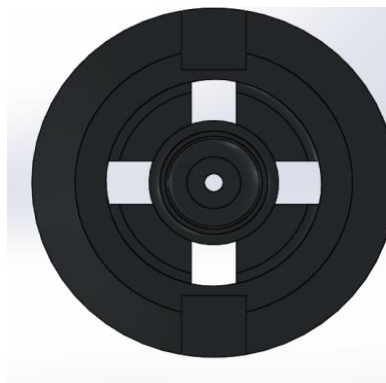


Figure 4-56: Base valve modelled in CATIA

4.9.2 Stress analysis:

The valve is pressure actuated. There are four orifices for the hydraulic fluid to flow through in open. The stress analysis of the valve was done in ANSYS 15.0 Workbench. The only moving part of the assembly is the diaphragm spring and it is observed that the failure of the valve is caused mainly by the failure of the spring. Hence the stress analysis has been done on the spring.

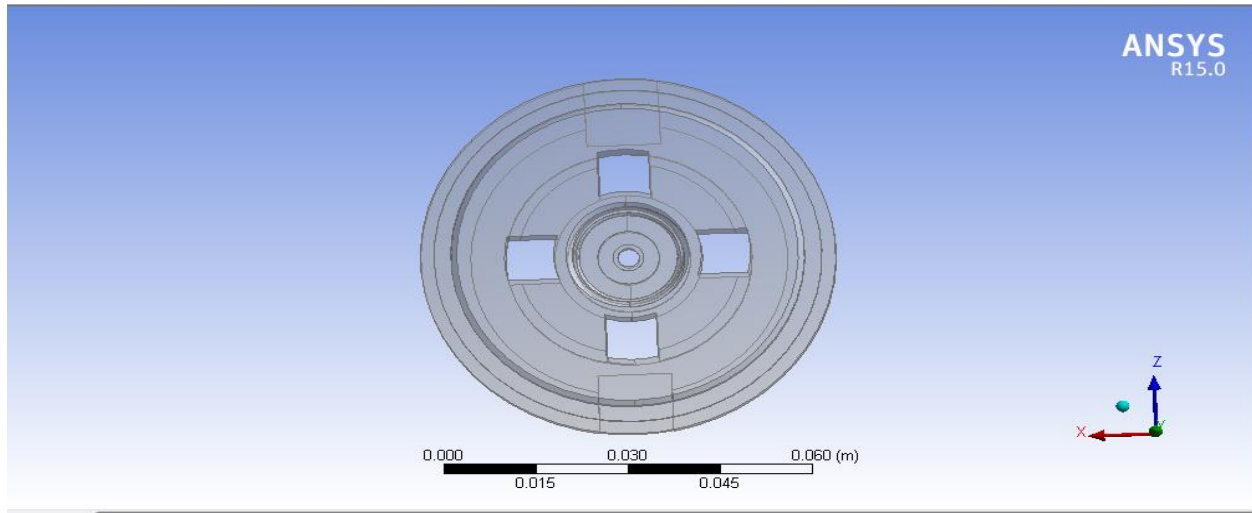


Figure 4-57 : Valve in ANSYS work bench

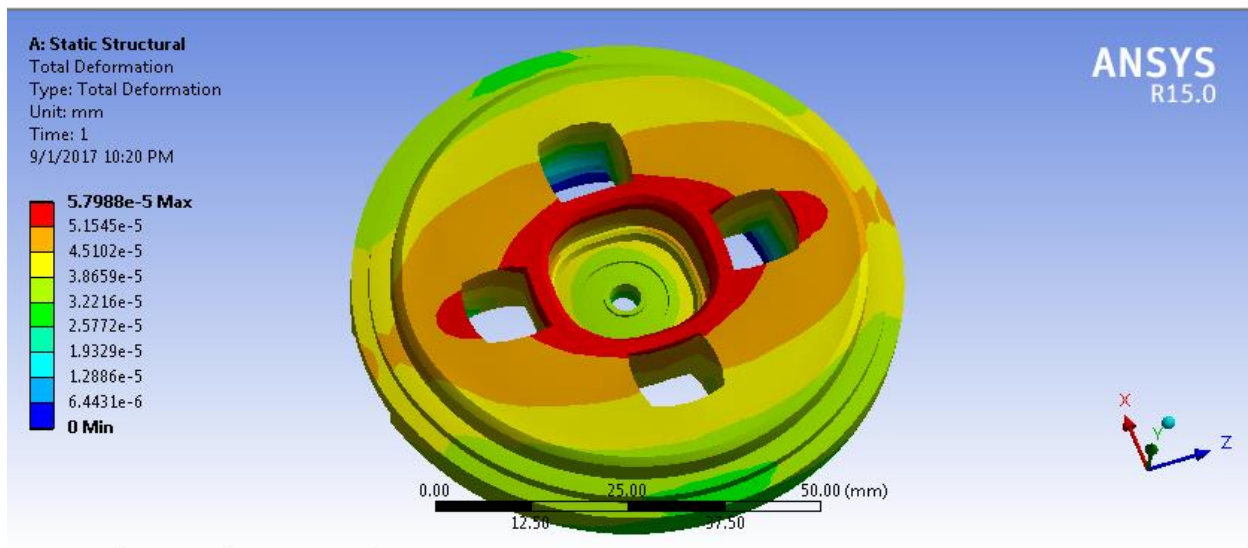


Figure 4-58 : Total deformation

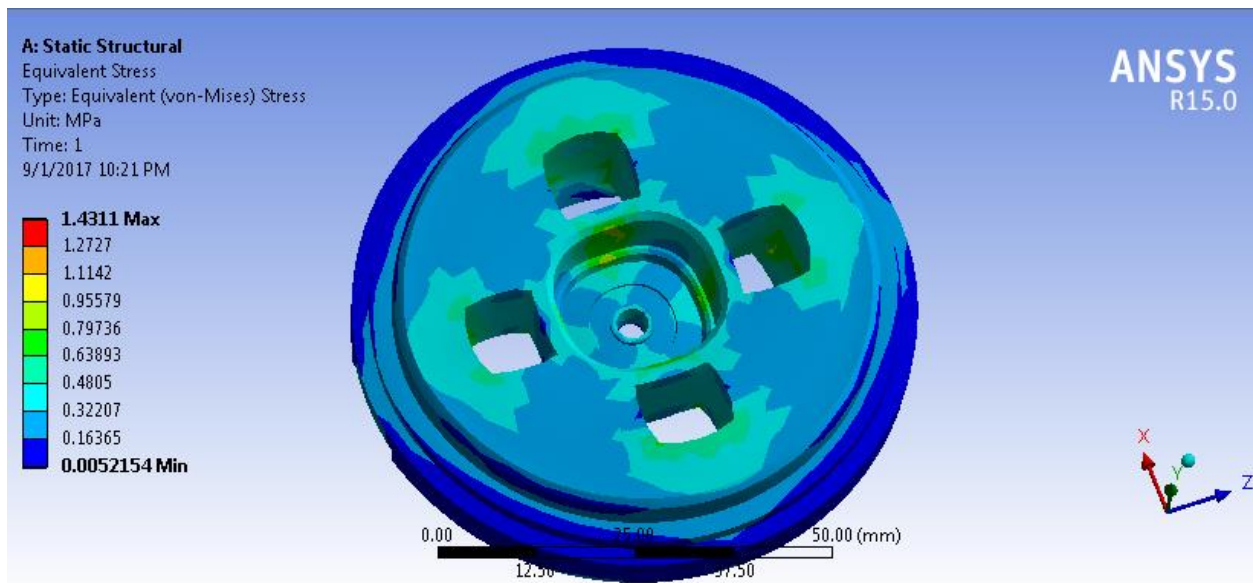


Figure 4-59: Equivalent (von-mises) stress

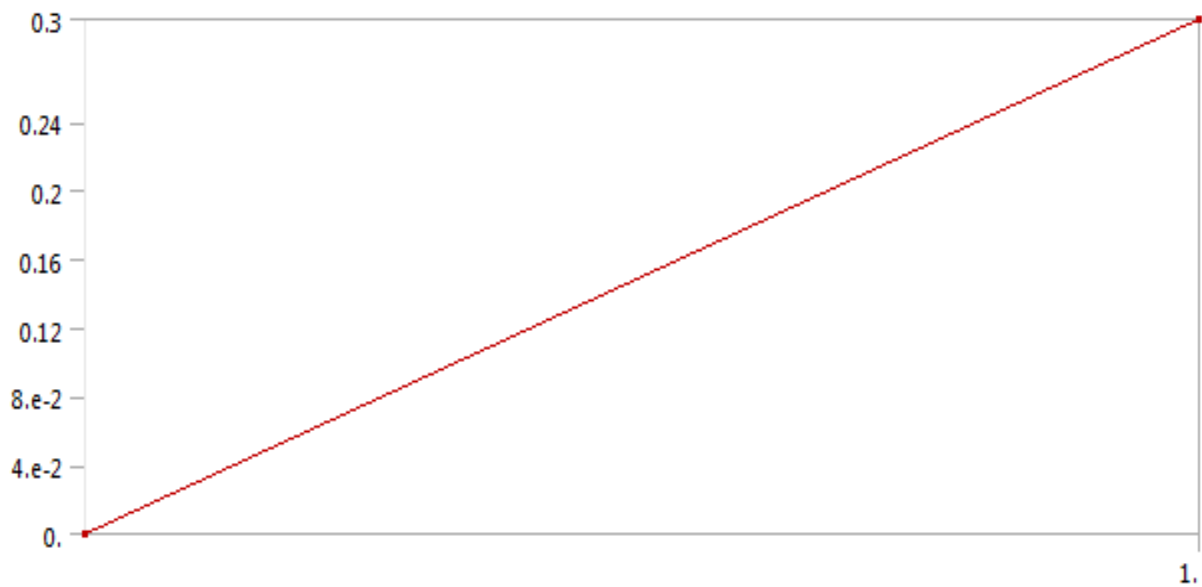


Figure 4-60: Pressure plot

4.10 Motion Analysis in Solid works

- ✓ Open SolidWorks Assembly Suspension Study.

Go to the Office Product Tab and click SolidWorks Motion Button.

Open Motion Study 1.

Select Motion Analysis in the drop down box.

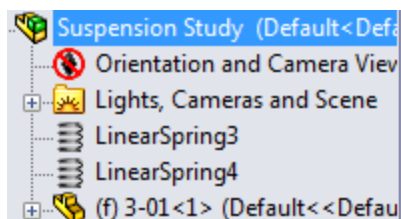
Motion Analysis



- ✓ We would like to analyze the forces of the system for 10 seconds, so select the black Diamond of the Suspension Study in the Motion Tab and drag it to 10 seconds.



- ✓ Now that the time has been determined, we would like to add two springs to the system to Model the shocks. The springs will have both damping and spring forces, to accurately Model actual shocks. This can be done with the **spring** button. 
- ✓ For the Spring Endpoints select the inner face of the upper disk of the shock and the Lower edge of the shock.
- ✓ Input the spring and damper parameters as shown to the right. These values were from experimentation with the car and from SolidWorks.
- ✓ Do this for both sides of the suspension, so that they are mirrored. Your Motion Tree should look something like the image below.



After both of the springs and forces have been determined for the simple suspension analysis, hit the **calculate** button  to simulate the motion and reactions. Once the motion had been

calculated the simulation can be ran a multiple times with the play button . Then we can calculate the wanted results of the spring force over the period and the spring displacement over the period. Click the Results and Plots button . Now we would like to calculate the reaction force of the spring over the time period, so follow the image to the right. The plot should look something like the lower figures 4-68.

The road profile is approximated by a sine wave represented by

$$q = Y \sin (\omega t) \quad (4-1)$$

Where,

q = Road surface excitation at time t in m.

Y = Amplitude of sine wave = 200 mm. and

l = Wavelength of road surface = 6000 mm.

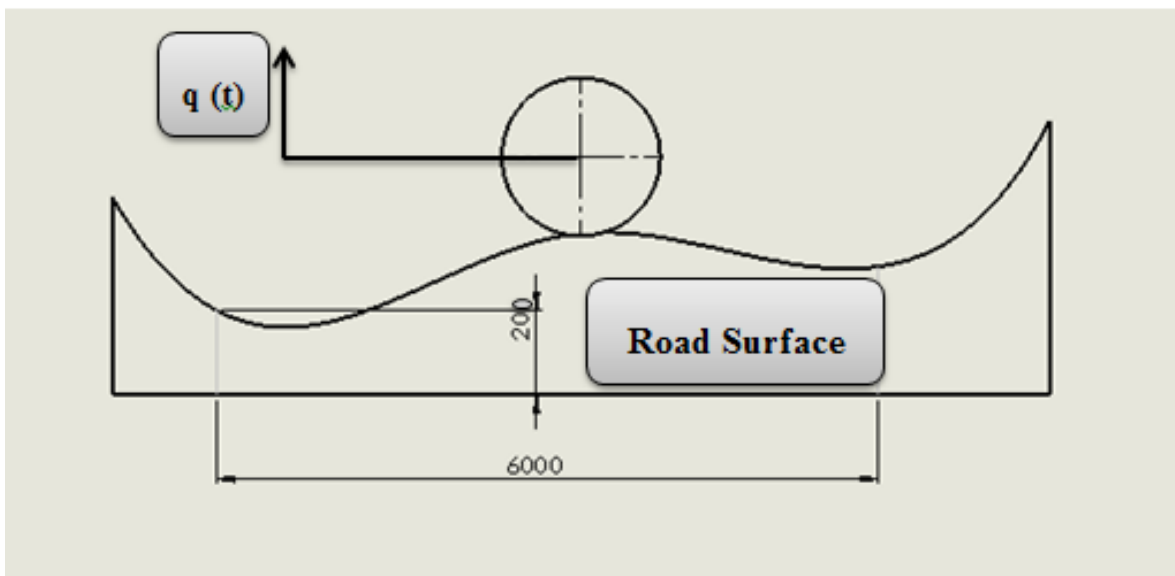


Figure 4-61: Road profile

4.10.1 Input variables:

Time=10sec The natural frequency of the vehicle dynamic system is given by,

$$\omega_n = \sqrt{\frac{k}{m}} \quad (4-2)$$

Where,

K= stiffness of spring (N/m) and

m= sprung mass (kg)

The value of K, m and damping coefficient (c) were obtained from analytical calculation.

These values are K=10983N/m. M=1890 kg, and c = 8200.8 N sec /m.

Therefore, $\omega_n = 2.4 \text{ rad/sec}$

Similarly damping ratio (ζ) for the front suspension is calculated by the equation

$$\zeta = \frac{c}{2\sqrt{km}} = \zeta = \frac{c}{c_c} \quad (4-3)$$

$$= \sqrt{\frac{\delta^2}{4\pi^2 + \delta^2}} = 0.99$$

$$C_c = 2\sqrt{km} \quad (4-4)$$

$$C = 8200.8 \text{ N sec /m} = 8.2 \text{ N sec/mm}$$

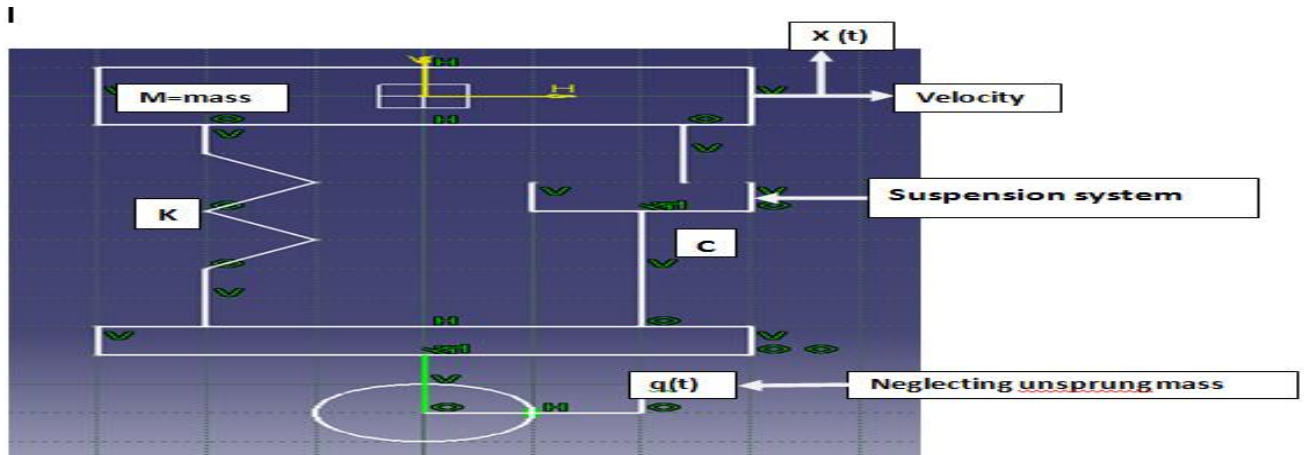


Figure 4-62 : Single degree damping system

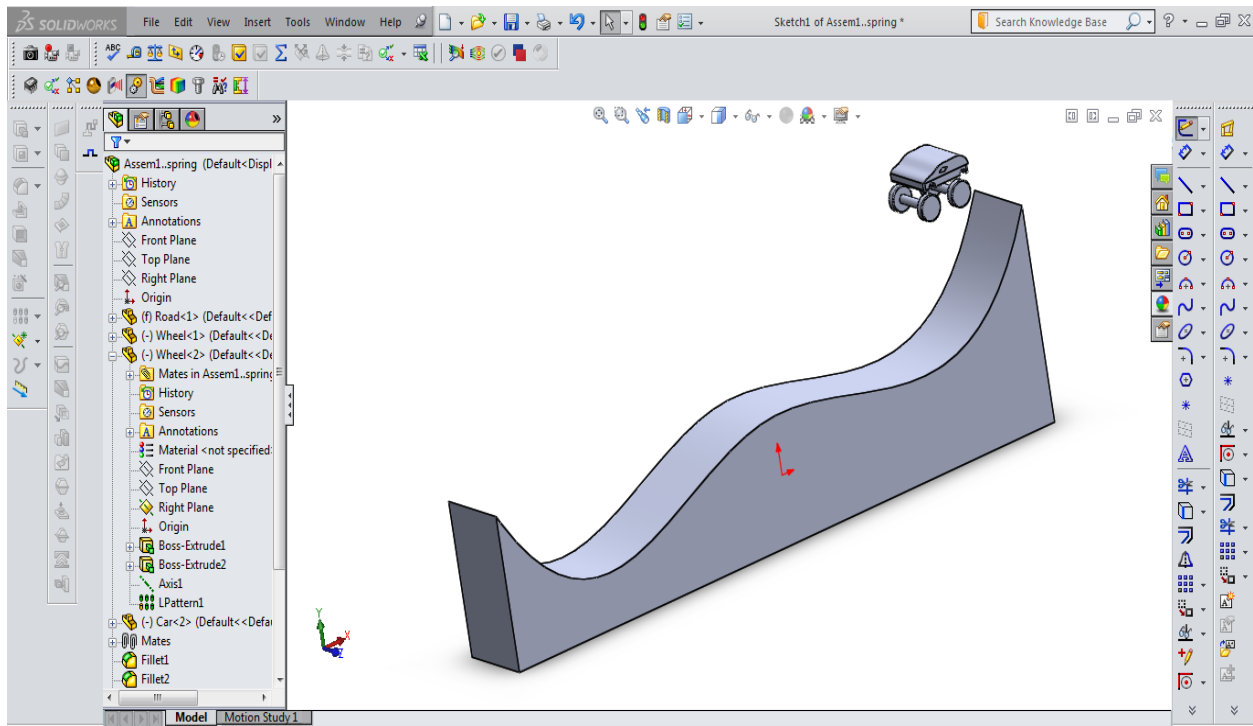


Figure 4-63 : Car model in solid work motion

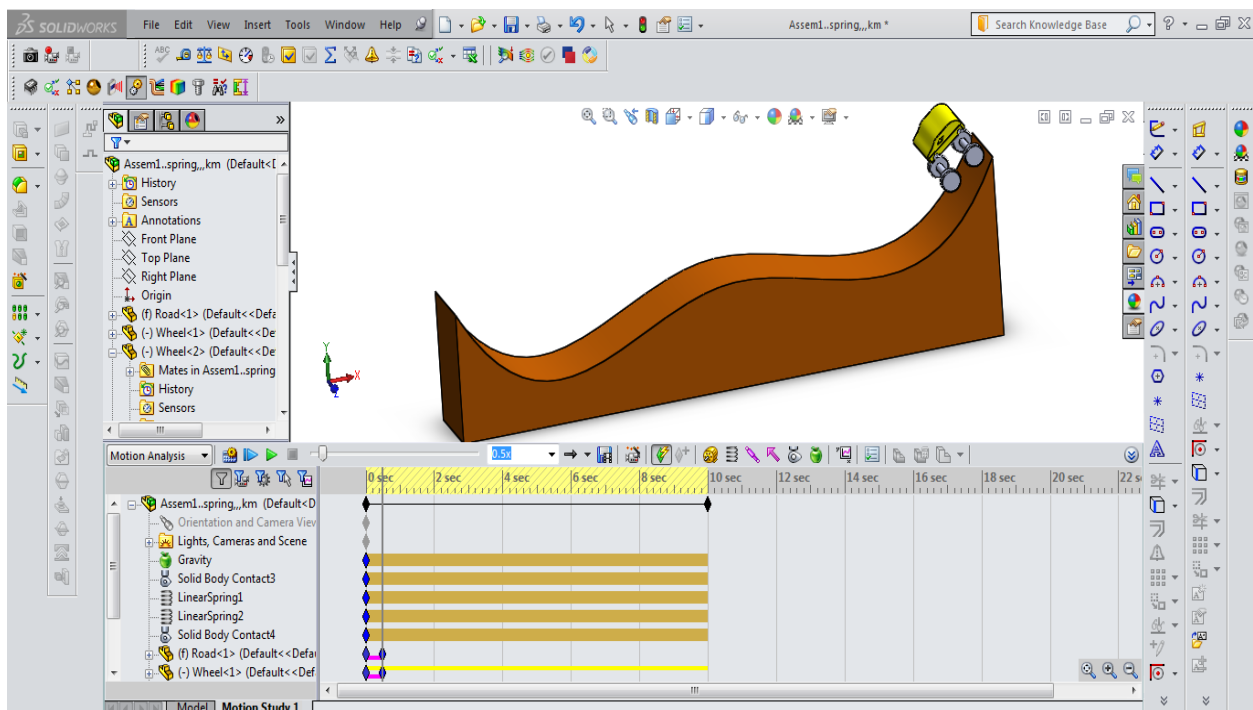


Figure 4-64: Analysis of spring characteristics on an uneven road

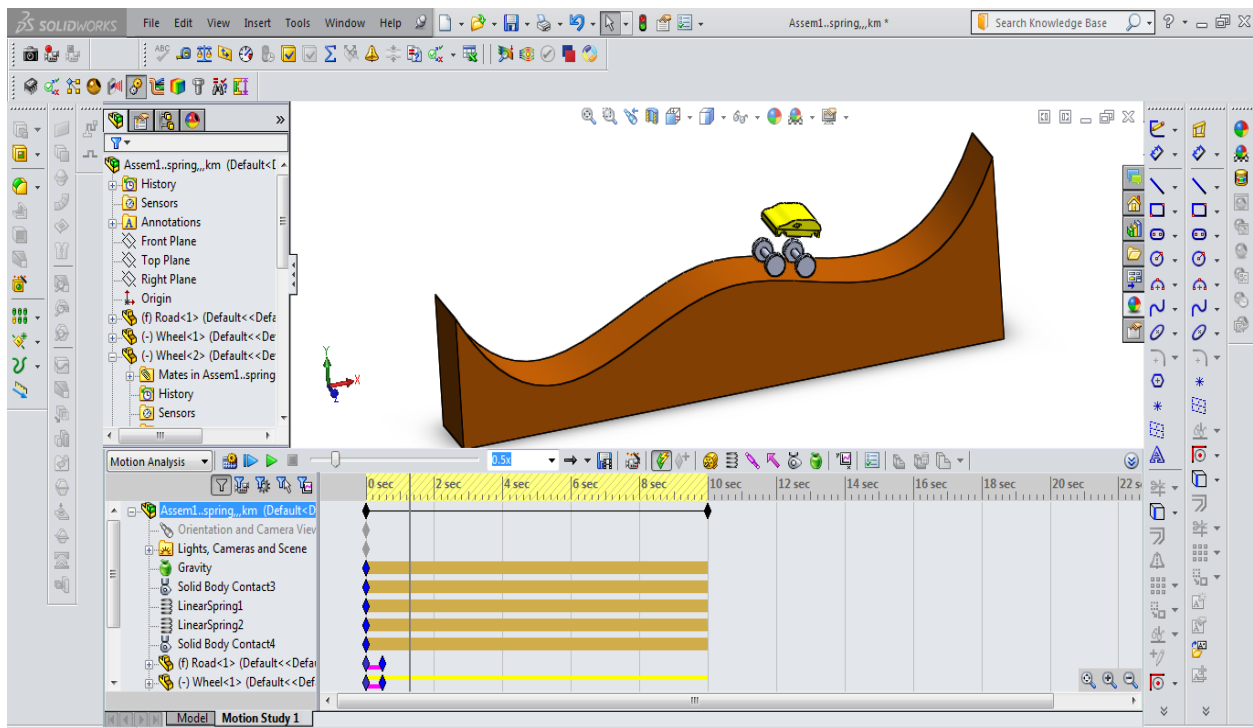


Figure 4-65: Analysis of spring characteristics on an uneven road

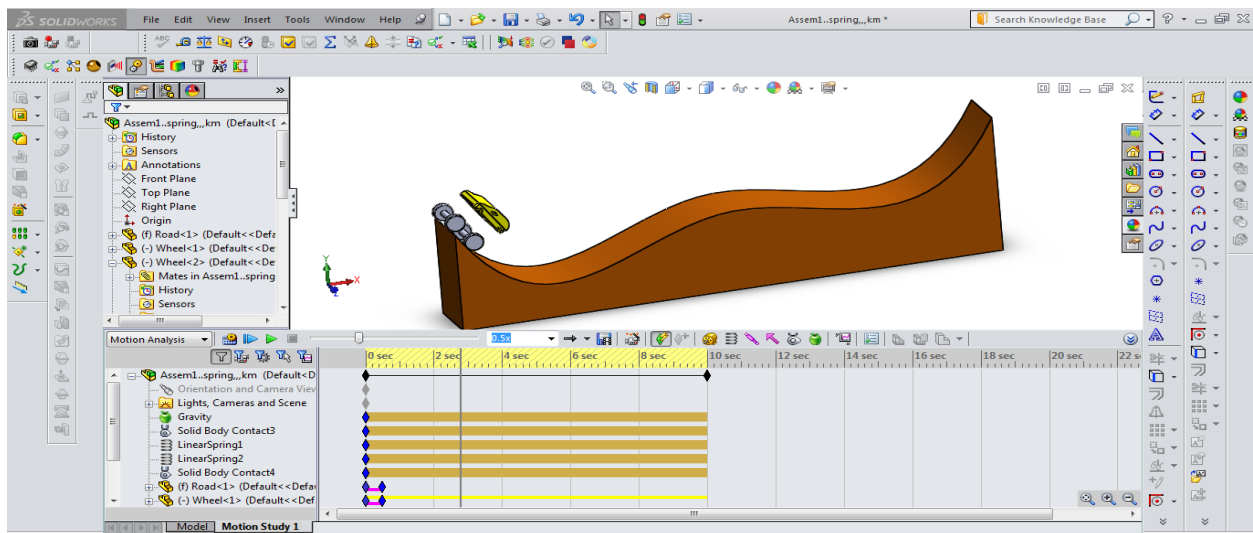


Figure 4-66 : Analysis of spring characteristics on an uneven road

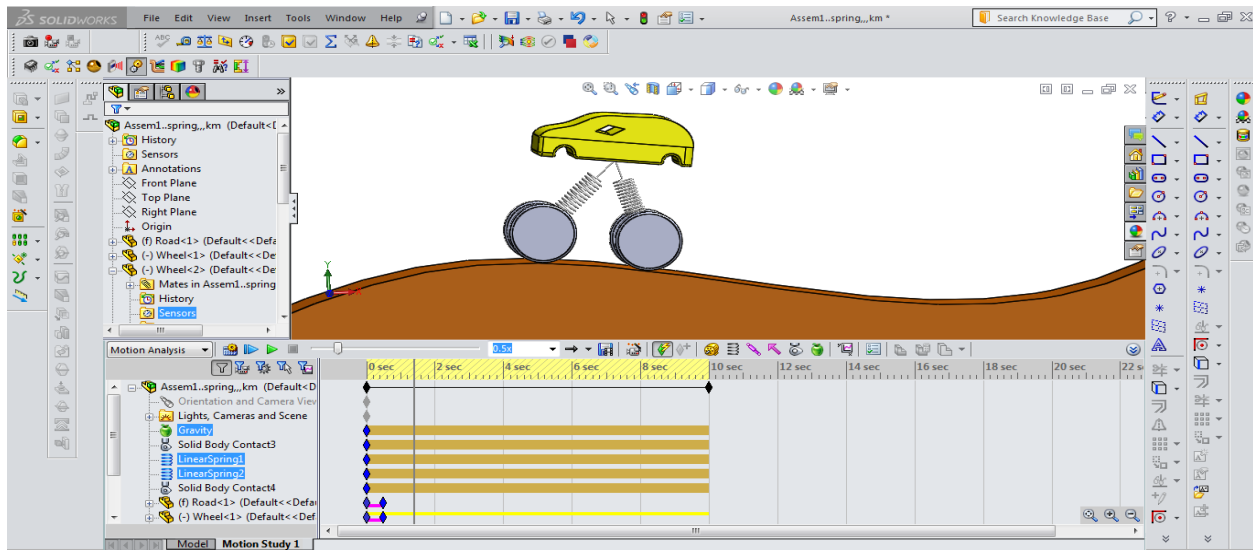


Figure 4-67: Analysis of spring characteristics on an uneven road

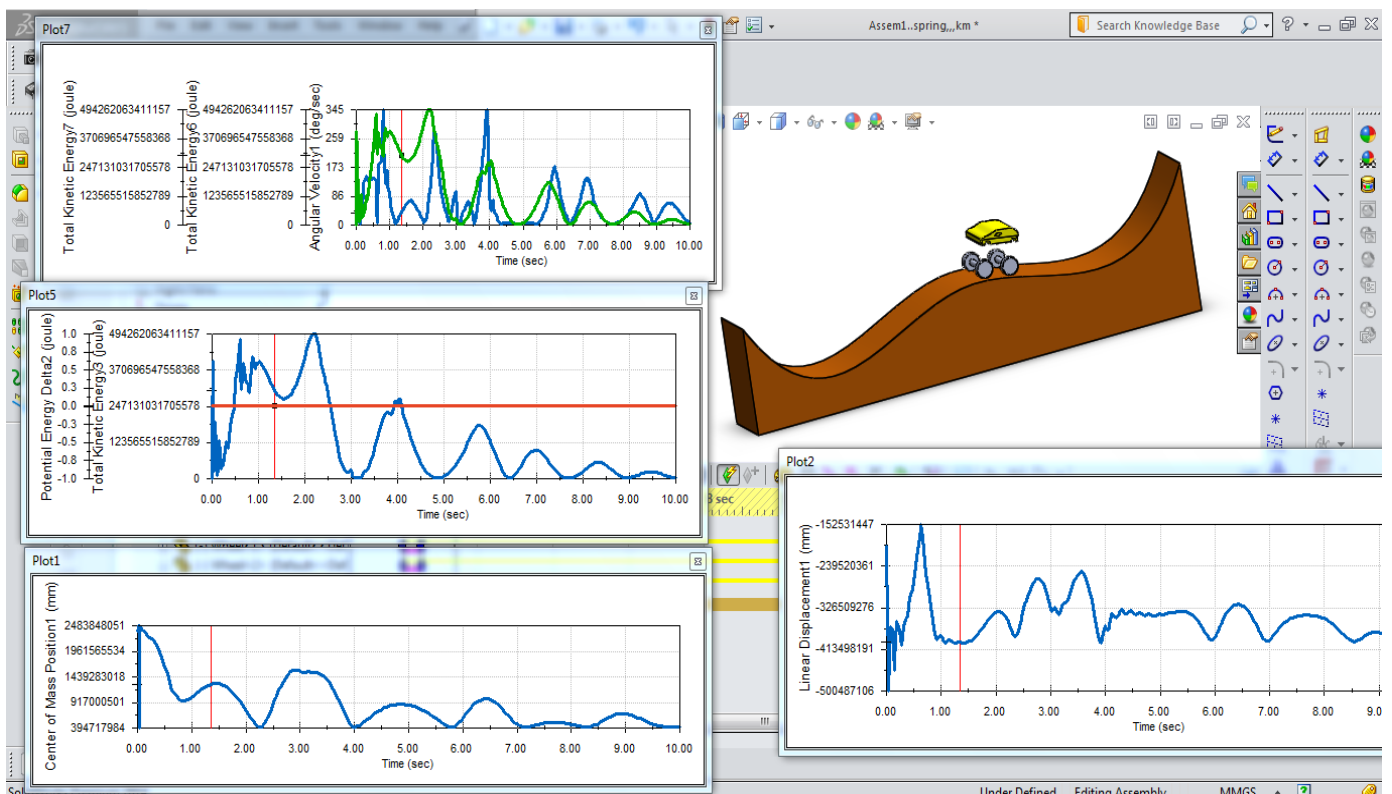


Figure 4-68: Responses of model

4.11 Responses of model

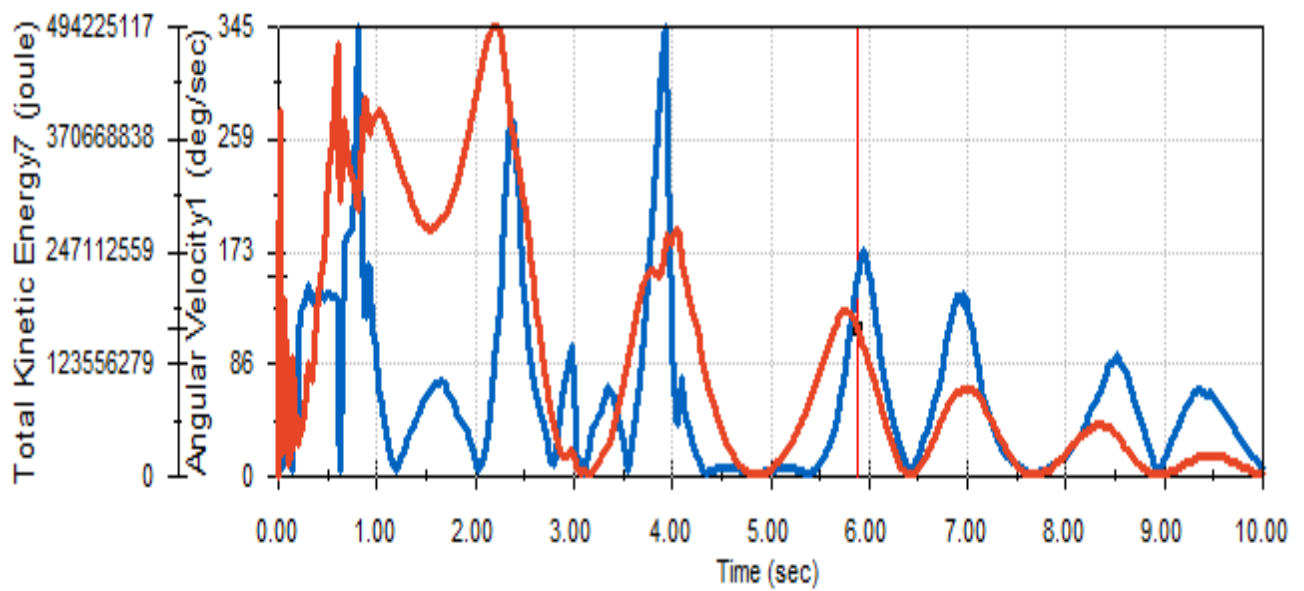


Figure 4-69: Total kinetic energy and angular velocity vs Time

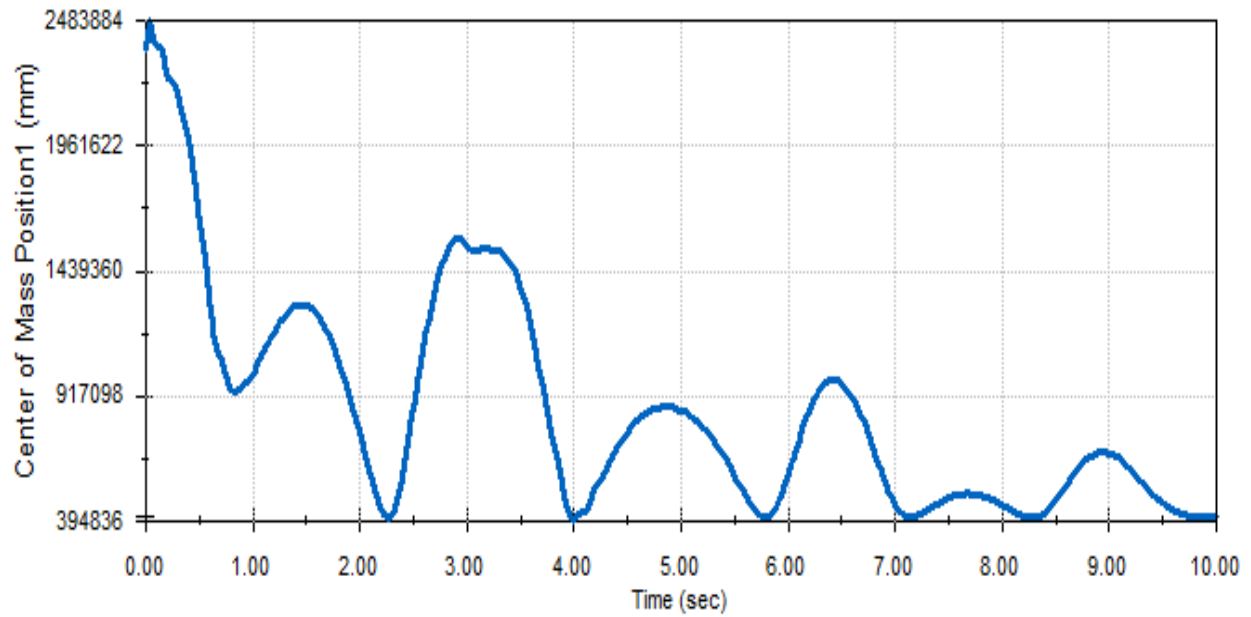


Figure 4-70: Center of mass position vs Time

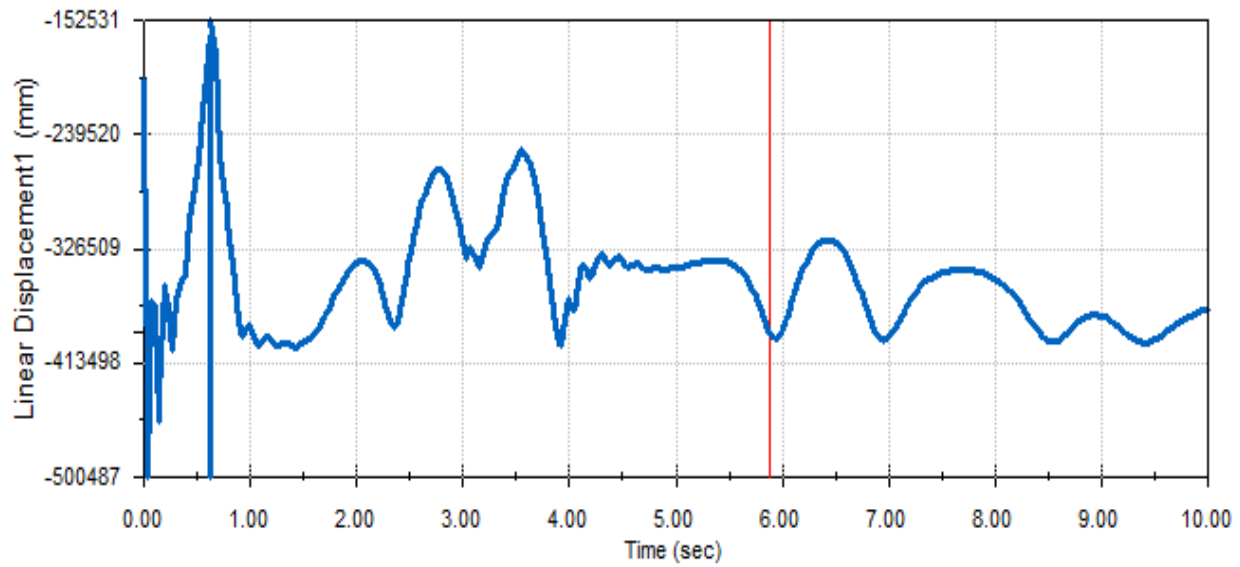


Figure 4-71: Liner displacement vs Time

- ✚ The displacement output for Solid works is the distance between the two endpoints of the spring.

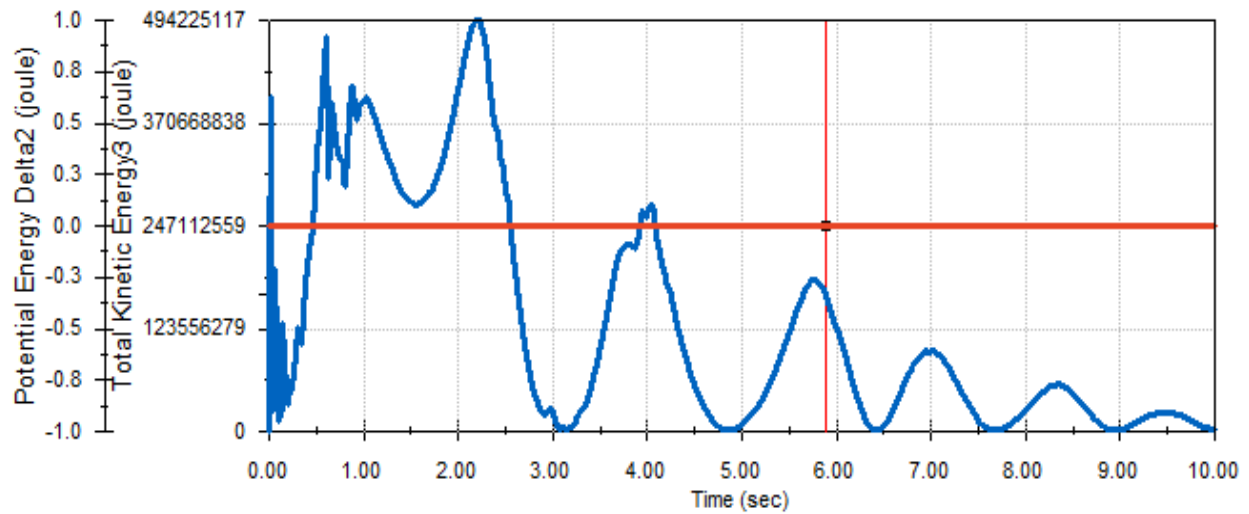


Figure 4-72: Potential energy and total kinetic energy vs Time

4.11.1 Deformation Result

Table 4-8: Total Deformation in mm

Material	ANSYS Workbench	
	2 Person Load [W=5542 N]	4 Persons Load [W=6020 N]
1) Existing shock absorber	0.032139	0.03499
2) Modified shock absorber Material for spring :(ASTM A228 [HIGH CARBON SPRING WIRE])	0.030383	0.033003

4.11.2 Reduction in Deflection (mm)

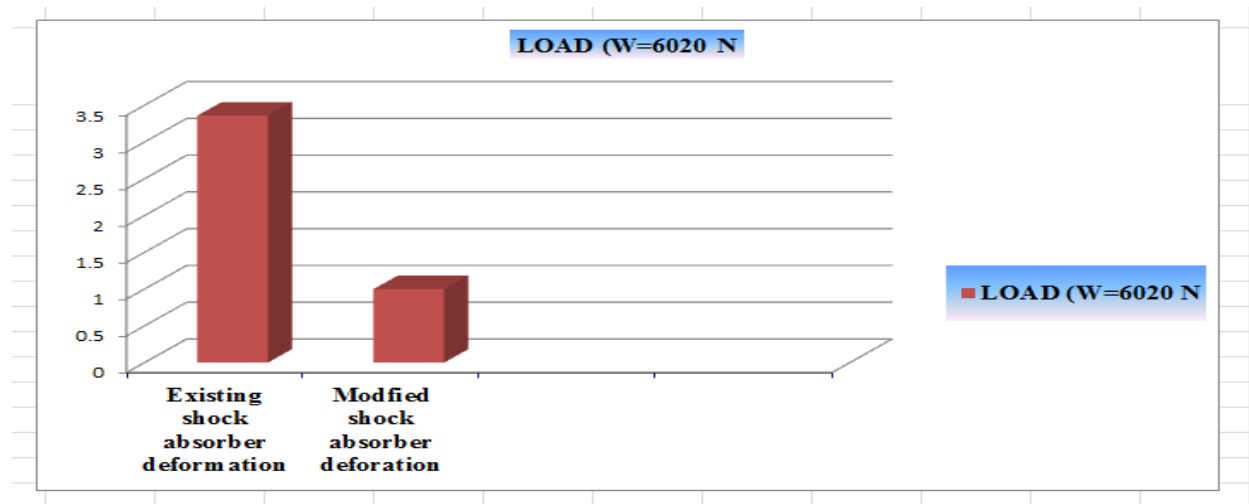


Figure 4-73: Reduction in deflection for load of (W= 5542N)

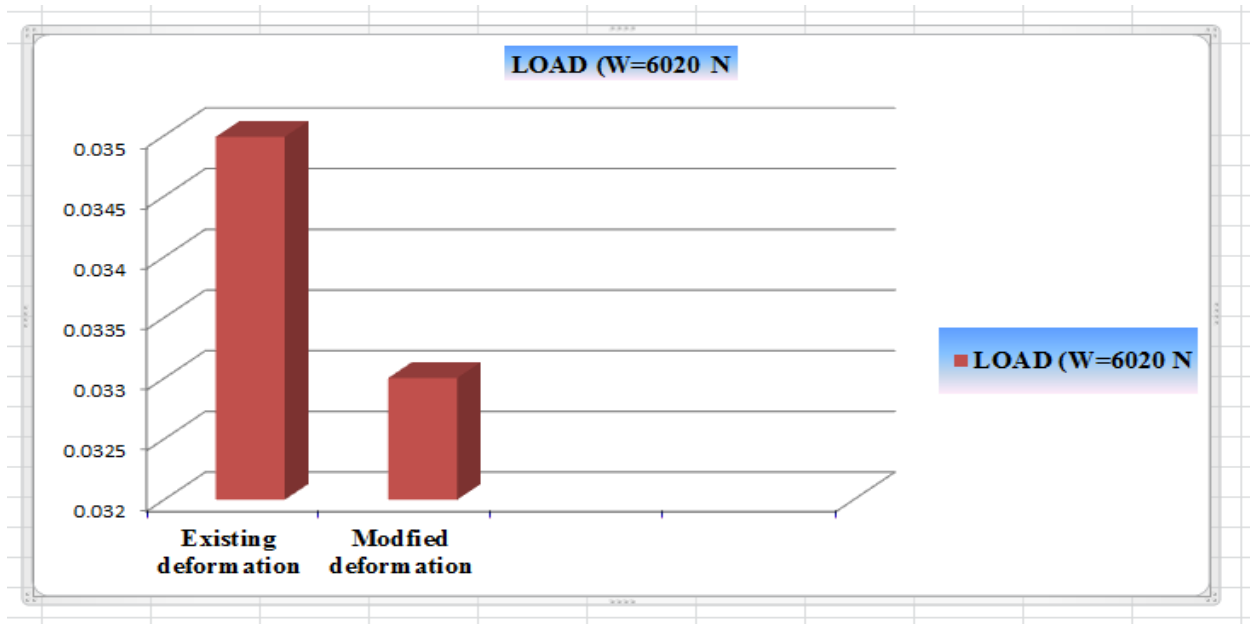


Figure 4-74 : Reduction in deflection for load of (W= 6020N0

4.11.3 Shear Stress Result

Table 4-9: Maximum Shear Stress in Mpa

Material	ANSYS Workbench	
	2 Person Load [W=5542 N]	4 Persons Load [W=6020 N]
1) Existing shock absorber	134.61	146.22
2) Modified shock absorber Material for spring :(ASTM A228 [HIGH CARBON SPRING WIRE])	98.204	106.67

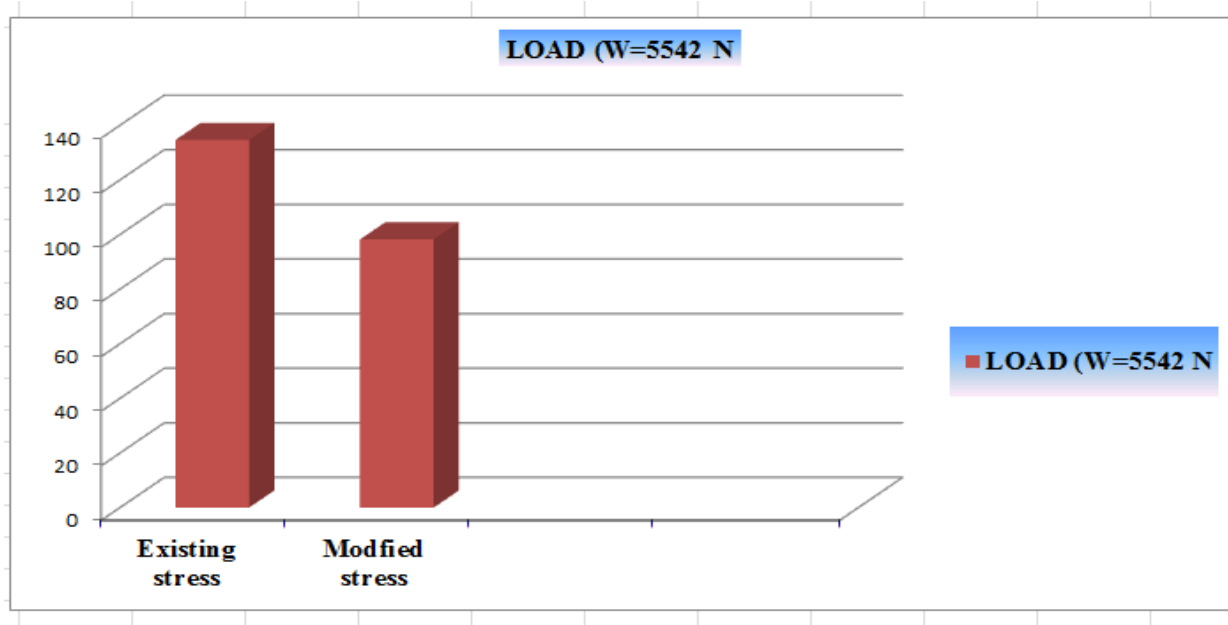


Figure 4-75: Reduction in Stress for load of (W=5542N)

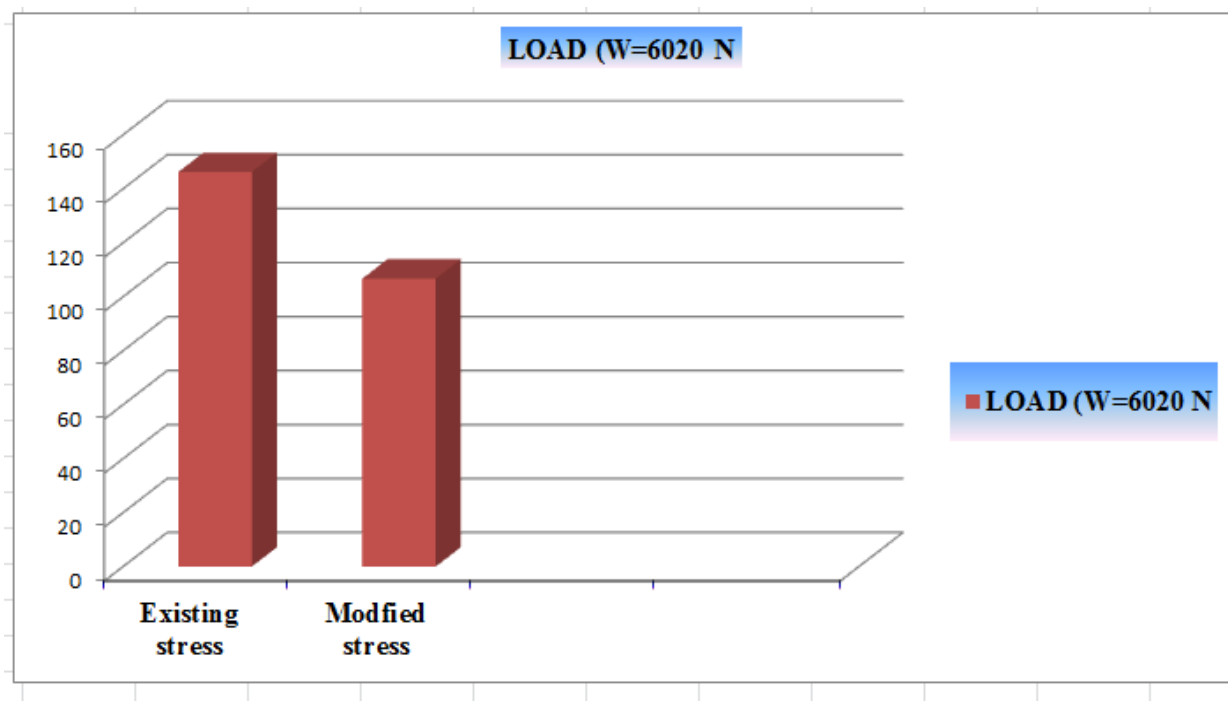


Figure 4-76: Reduction in Stress for load of (W= 6020N)

Total Deflection in mm

Table 4-10: Reduction in deflection

Material	ANSYS Workbench	
	2 Person Load [W=5542 N]	4 Persons Load [W=6020 N]
1) Existing shock absorber	0.032139	0.03499
2) Modified shock absorber Material for spring :(ASTM A228 [HIGH CARBON SPRING WIRE])	0.030383	0.033003

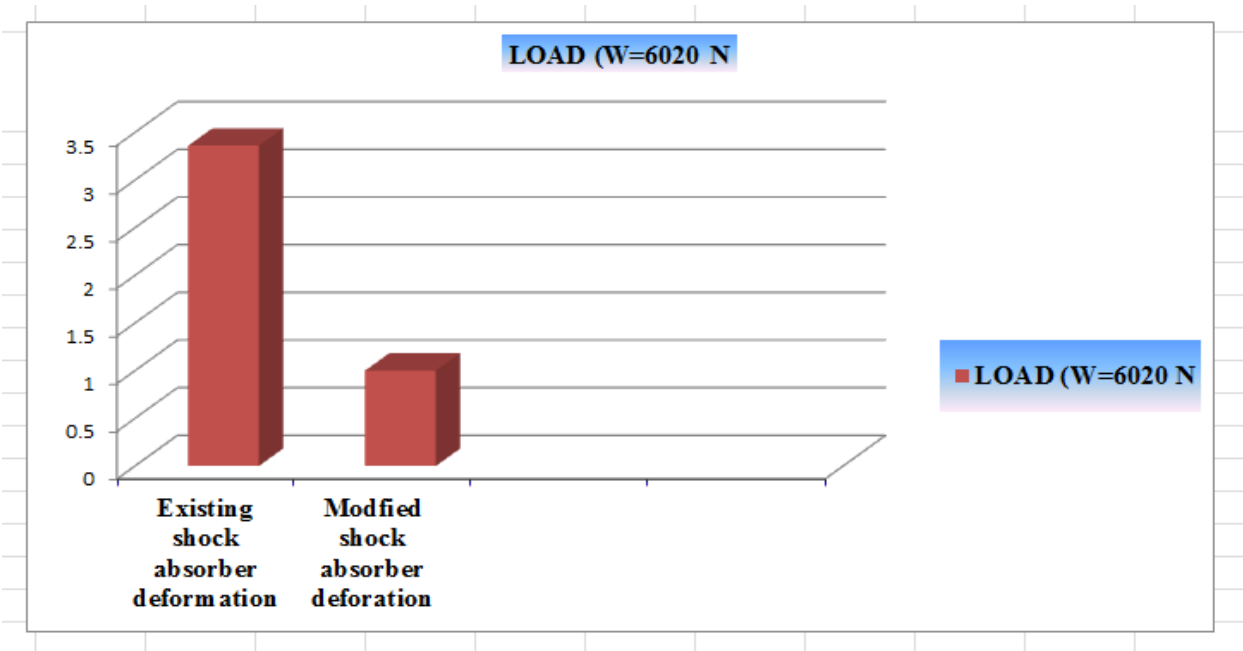


Figure 4-77: Reduction in deflection for load of (W=5542N)

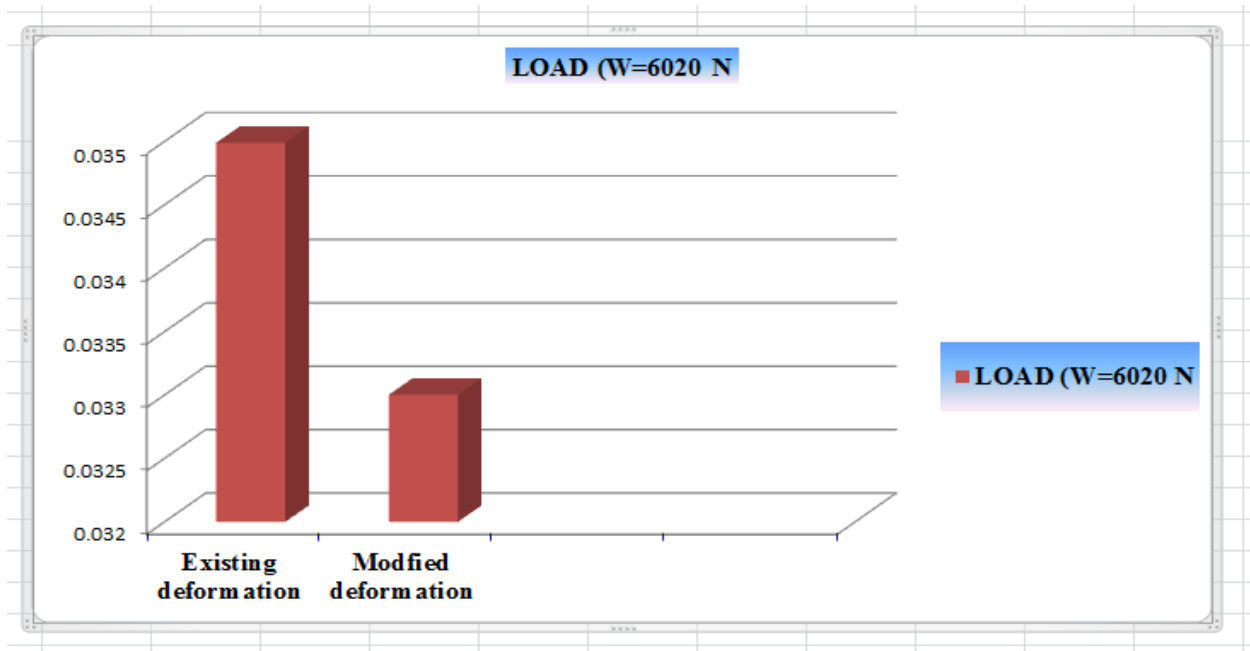


Figure 4-78: Reduction in deflection for load of (W= 6020N)

Maximum Shear Stress in Mpa

Table 4-11: Reduction in stress

Material	ANSYS Workbench	
	1 Person Load [W=5542 N]	2 Persons Load [6020 N]
1) Existing shock absorber	134.61	146.22
2) Modified shock absorber material for spring : (ASTM A228 [HIGH CARBON SPRING WIRE])	98.204	106.67

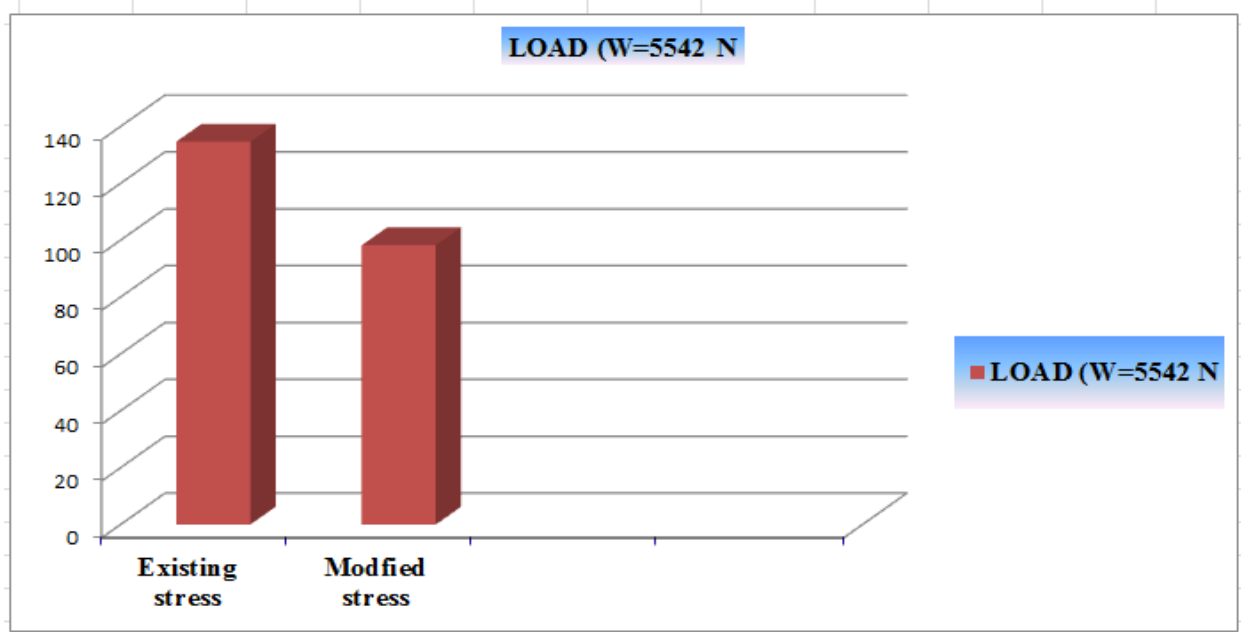


Figure 4-79: Reduction in Stress for load of (W=5542N)

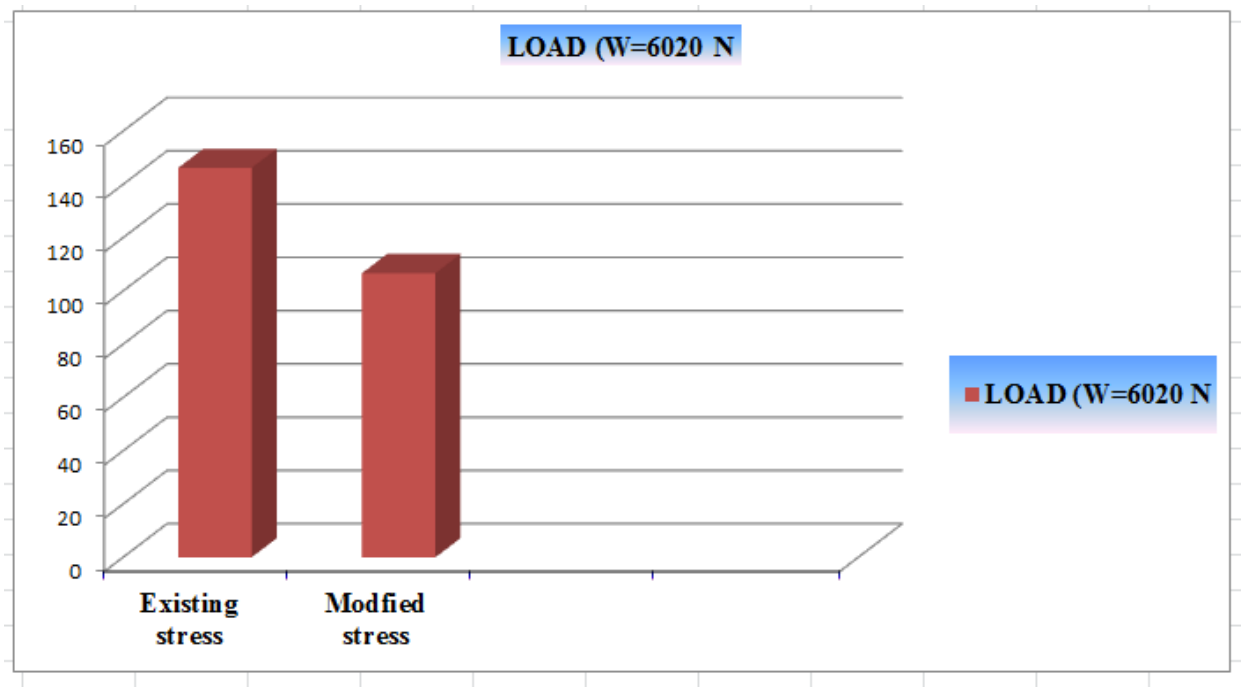


Figure 4-80: Reduction in Stress for load of (W=6020N)

CHAPTER FIVE

5 SUMMARY, CONCLUSION, RECOMMENDATION AND FUTURE WORK

5.1 SUMMARY

In this thesis work structural and modal analysis of small four wheel vehicle is done. The analysis is performed on the existing shock absorber and modified models. Both the existing and modified models are modeled using CATIA V5 and imported to ANSYS workbench in the form of, IGES, STEP file for detail analysis .Then this, IGES or STEP file are analyzed using static structural in ANSYS and Motion analysis in SOLIDWORK successfully.

The modified shock absorber model which contains spring and existing shock absorber without spring deformation and stress analysis performed by using static and modal analysis in the ANSYS workbench. And the spring of the modified shock absorber also analyzed by ANSYS and SOLIDWORK

5.2 CONCLUSION

- ✓ In this project we have designed and modify a shock absorber of TOYOTA HILUX 1999. We have modeled the shock absorber by using 3D parametric software CATIA.
- ✓ To validate the strength of our design, we have done structural analysis and modal analysis for existing and modified shock absorber.
- ✓ By comparing the results for shock absorber models, the total deflection and stress value is less for modified shock absorber than existing one.
- ✓ So we can conclude that as per our analysis the modified shock absorber has better performance than existing one.
- ✓ We have done analysis for modified shock absorber by selecting for sprig material (ASTM A228 [high carbon spring wire]).

- ✓ To validate the strength of our design, we have done structural analysis and modal analysis for spring material. We have done analysis between 67SiCr5 and ((ASTM A228 [high carbon spring wire])).
- ✓ By observing the analysis results, the analyzed stress values (106.67MPa) are less than their respective yield stress values (250MPa). So our design is safe.
- ✓ By comparing the results for both materials, the total deflection value is less for ASTM A228 than 67SiCr5. So stiffness is more for Spring Steel.
- ✓ So we can conclude that as per our analysis using ASTM A228 [high carbon spring wire] for spring is best.
- ✓ In this paper small car model is developed for analysis of vibrational effect when it is subjected to harmonic excitation by road profile
- ✓ From the results it can be seen that the maximum stresses generated in the valve range from 0.00521 MPa to 1.4311 MPa. These stresses are well within the allowable limits for Structural steel, which has yield strength of 250 MPa. It can hence be safely stated that structural steel can be used to manufacture the base valve without excessive risk.

5.3 RECOMMENDATION

- During shock absorber installation on the vehicle must be considered structural comfort; crashworthiness enhancement, good aerodynamics and ergonomic features.
- Material effect is considered in this study. The shock absorber mechanism or all newly added systems are made from stiff, higher in absorbing vibration and light in weight material.

5.4 FUTURE WORK

- A numerical model using ADAMS has to be developed in order to simulate the dynamic behavior of the shock absorber and to describe and evaluate its damping coefficient in compression and rebound cycles.
- The design of interchangeable shock absorber test rig has to be developed and fabricated for the dynamics measurement system.

- A full vehicle simulation model has to be developed to get better results.

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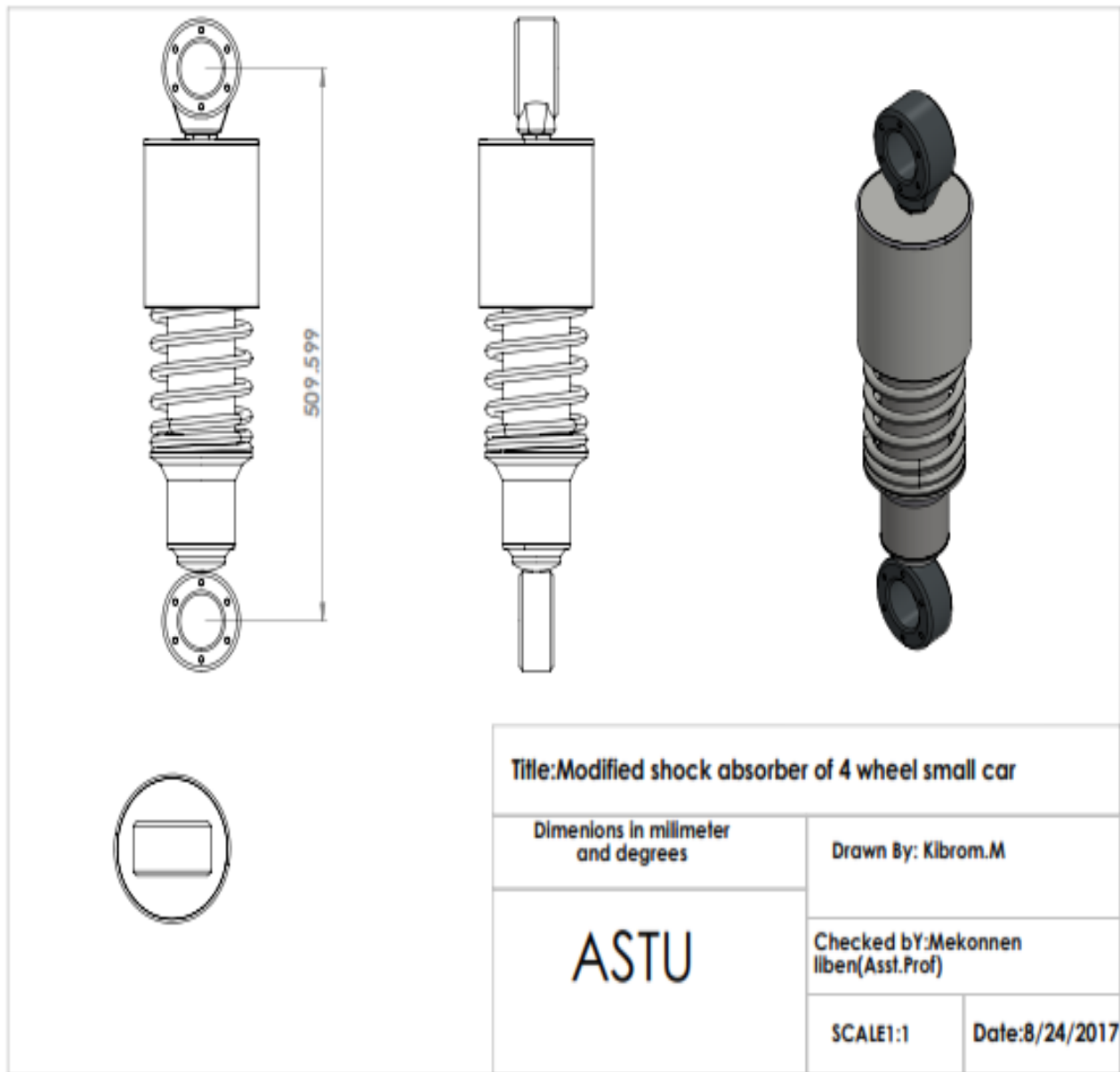
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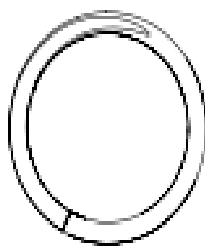
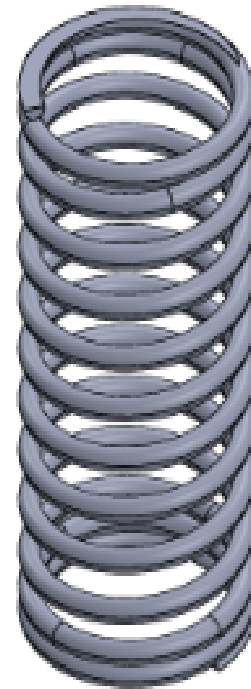
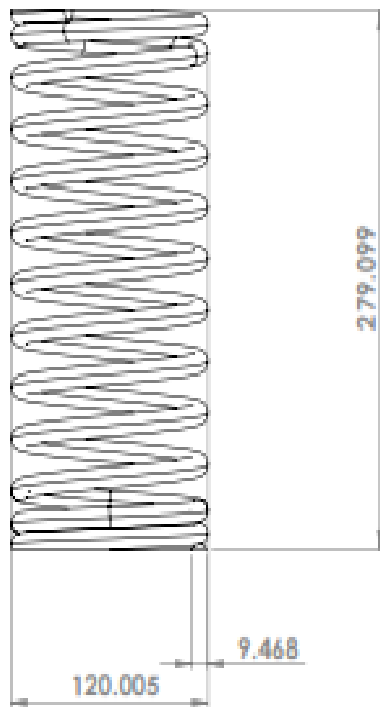
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APPENDIX-A



APPENDIX-B



Title:Compress Spring

Dimenlons in millimeter and
degrees

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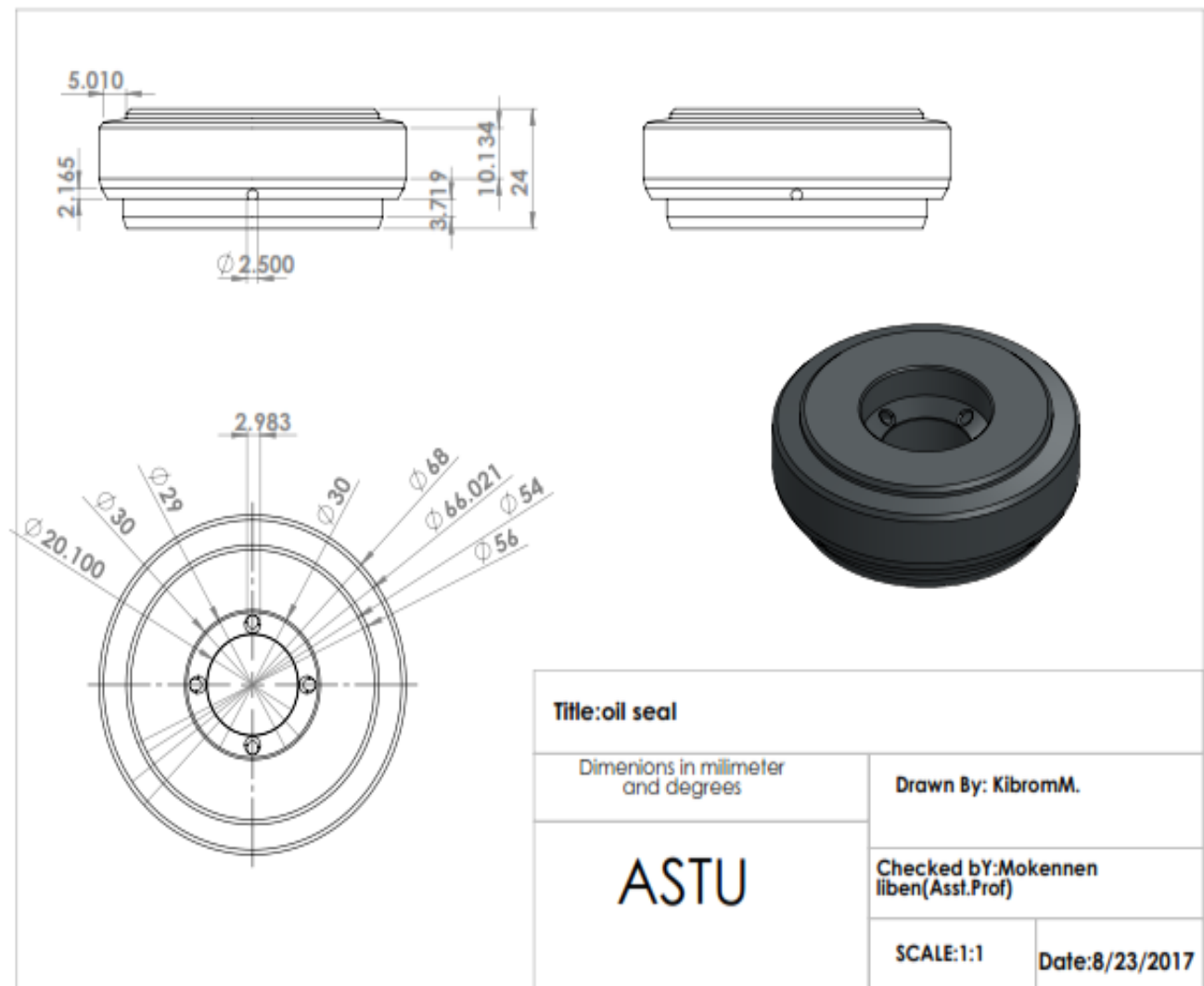
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Checked BY:Mekonnen
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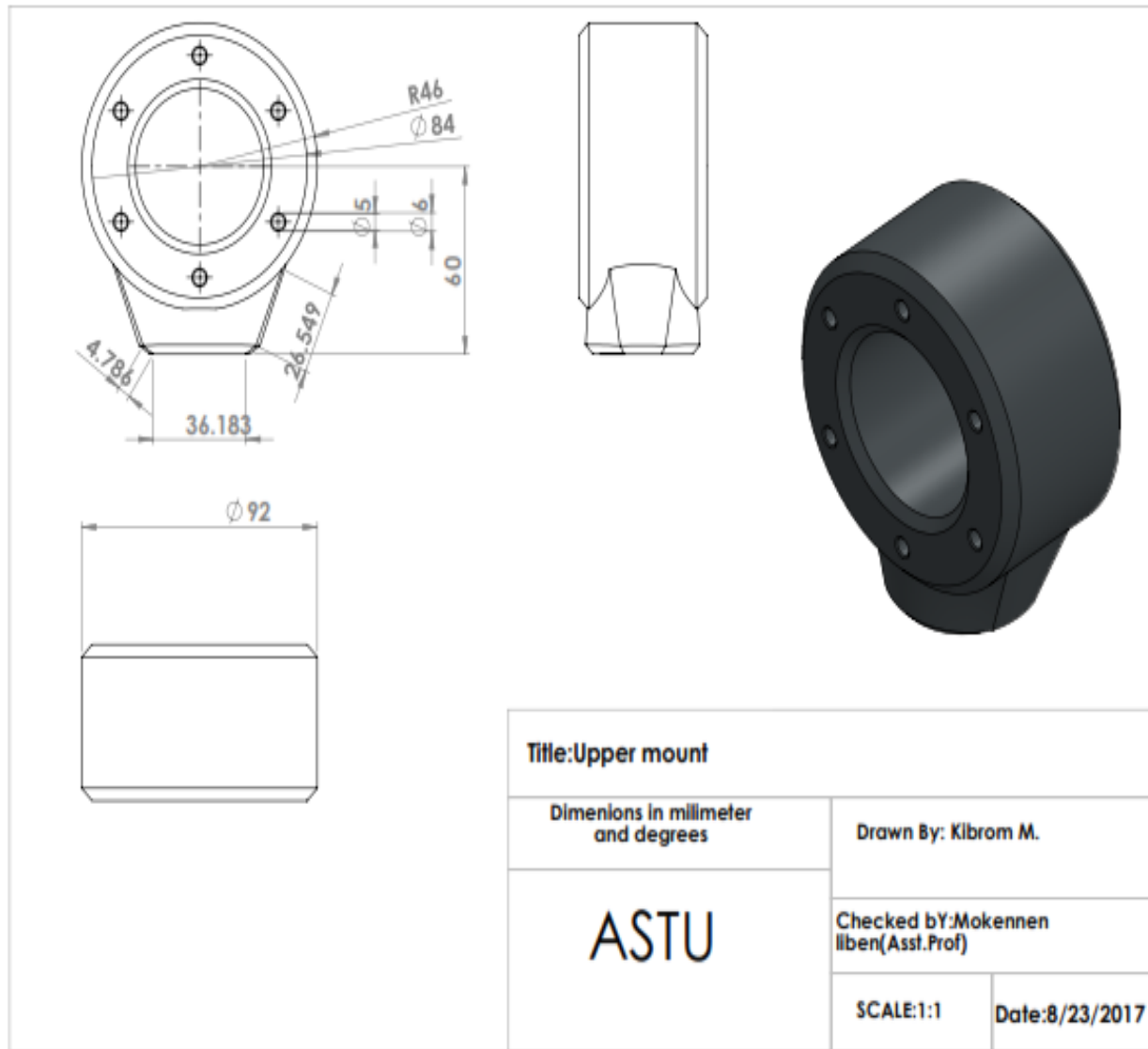
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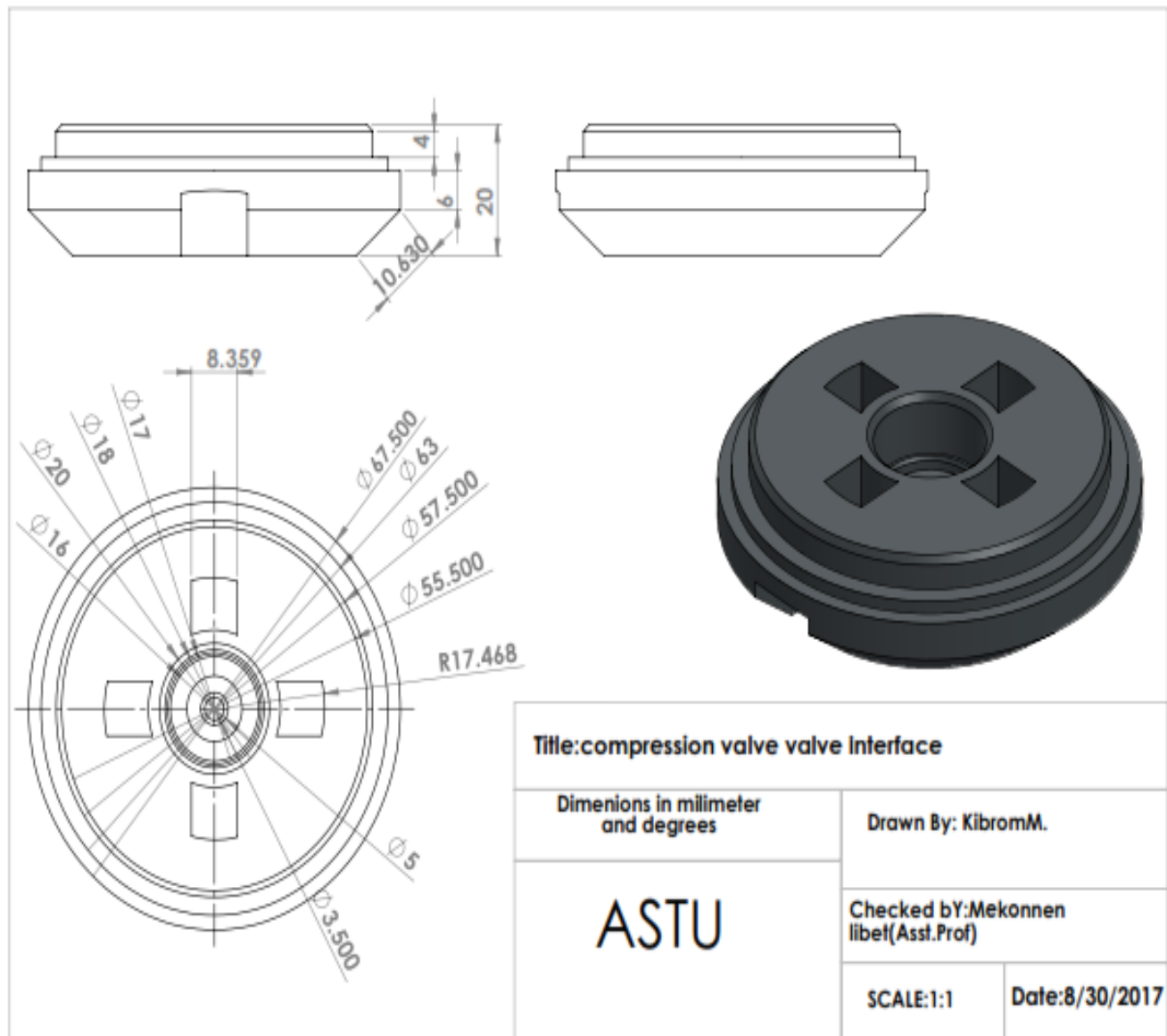
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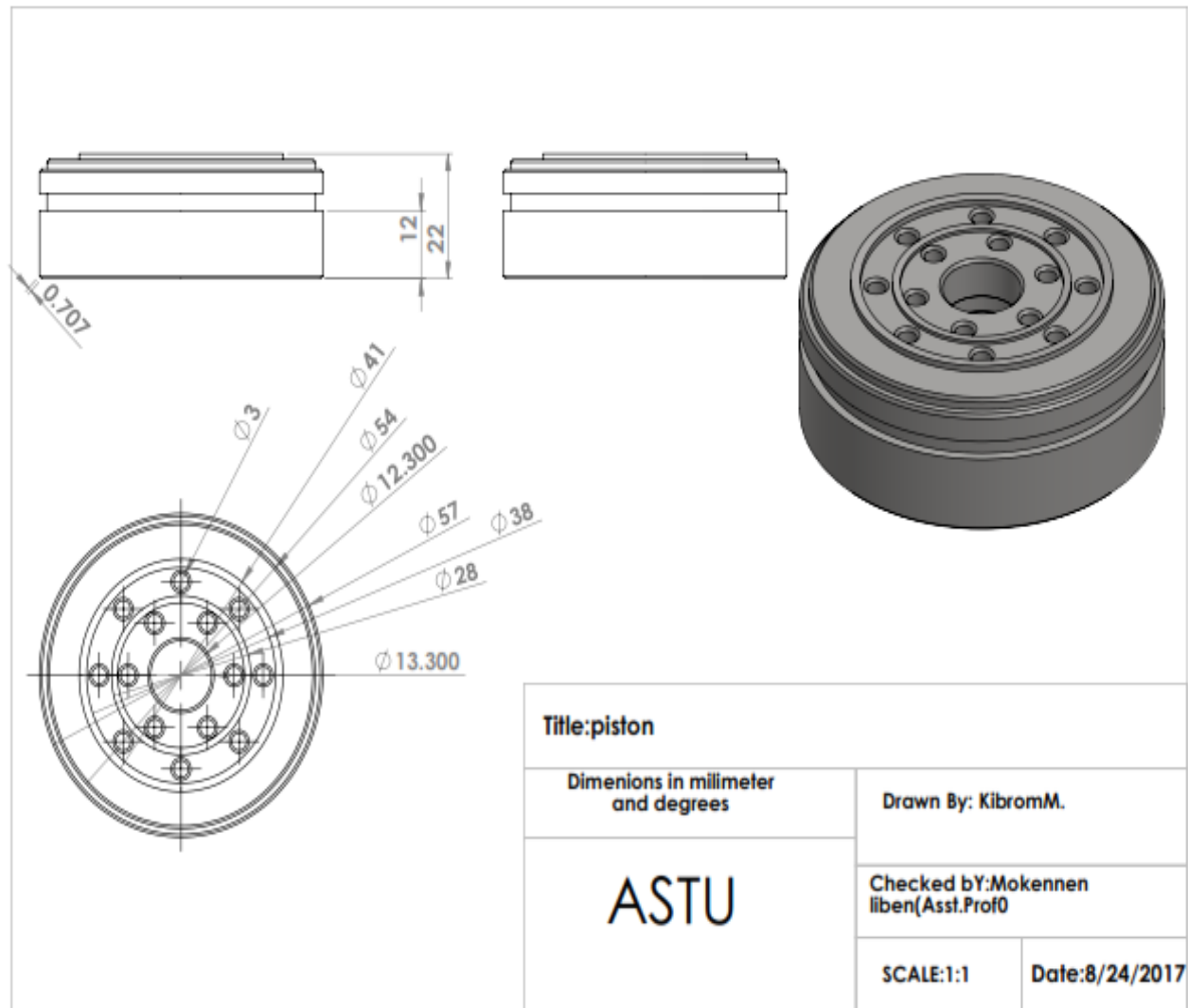
APPENDIX-D



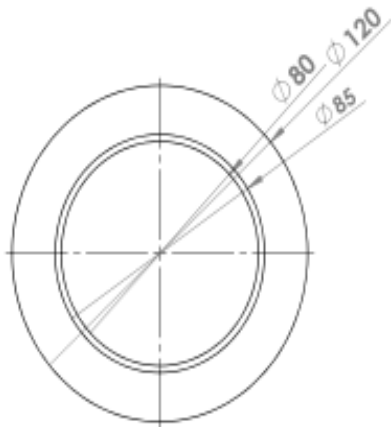
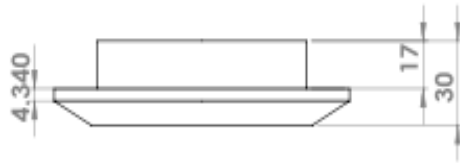
APPENDIX-E



APPENDIX-F



APPENDIX-G



Title: Spring seat

Dimensions in millimeter
and degrees

Drawn By: KibromM.

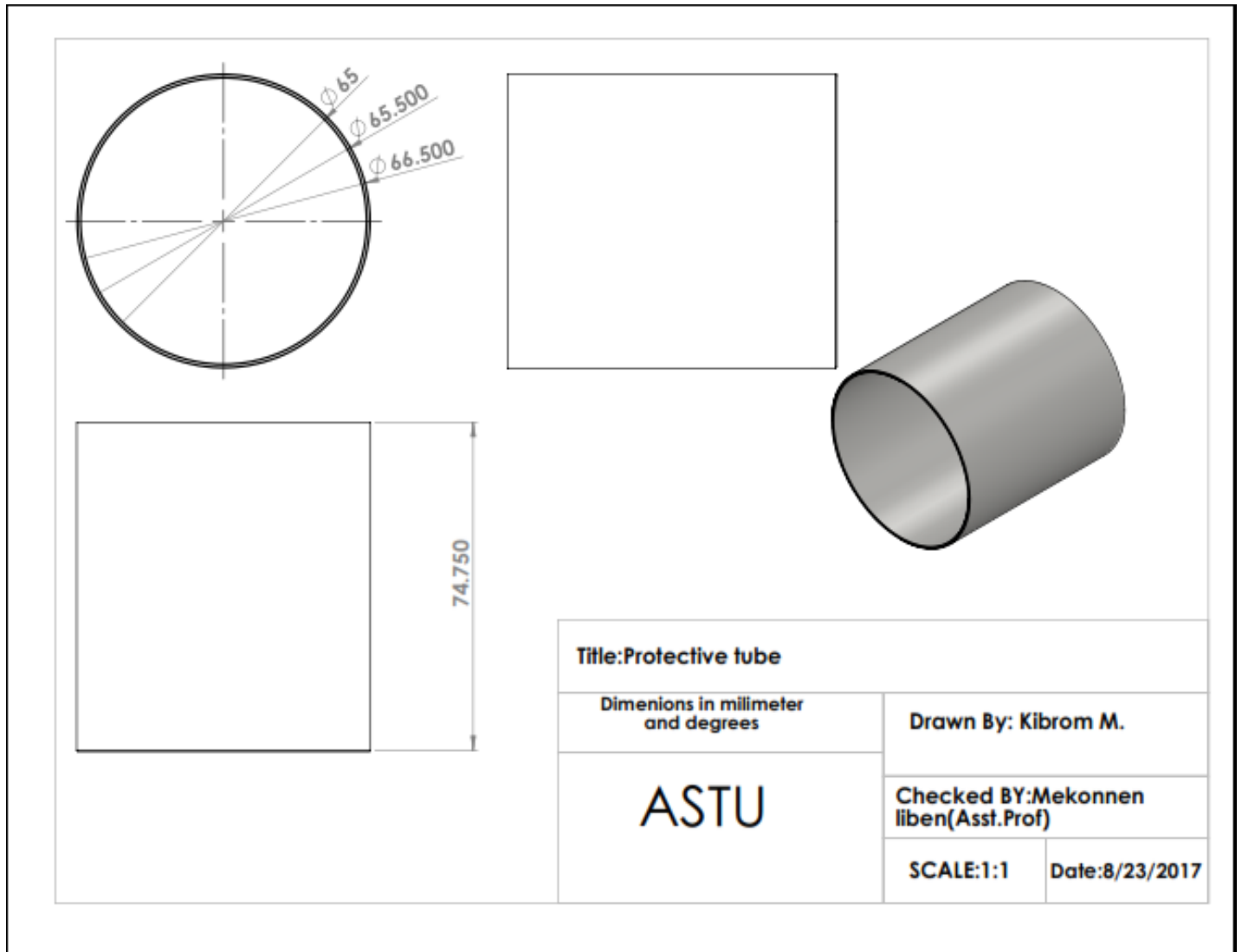
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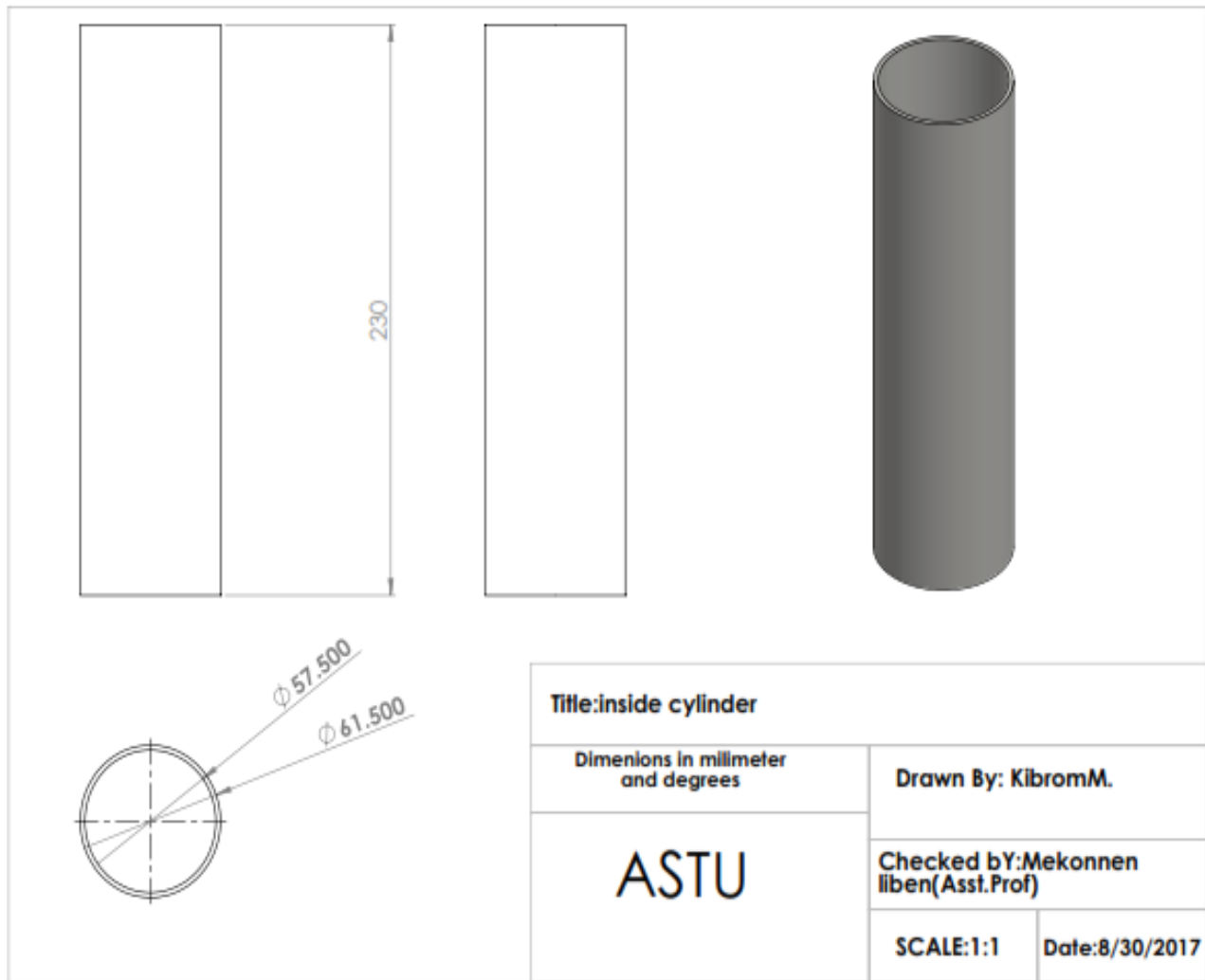
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Date: 8/23/2017

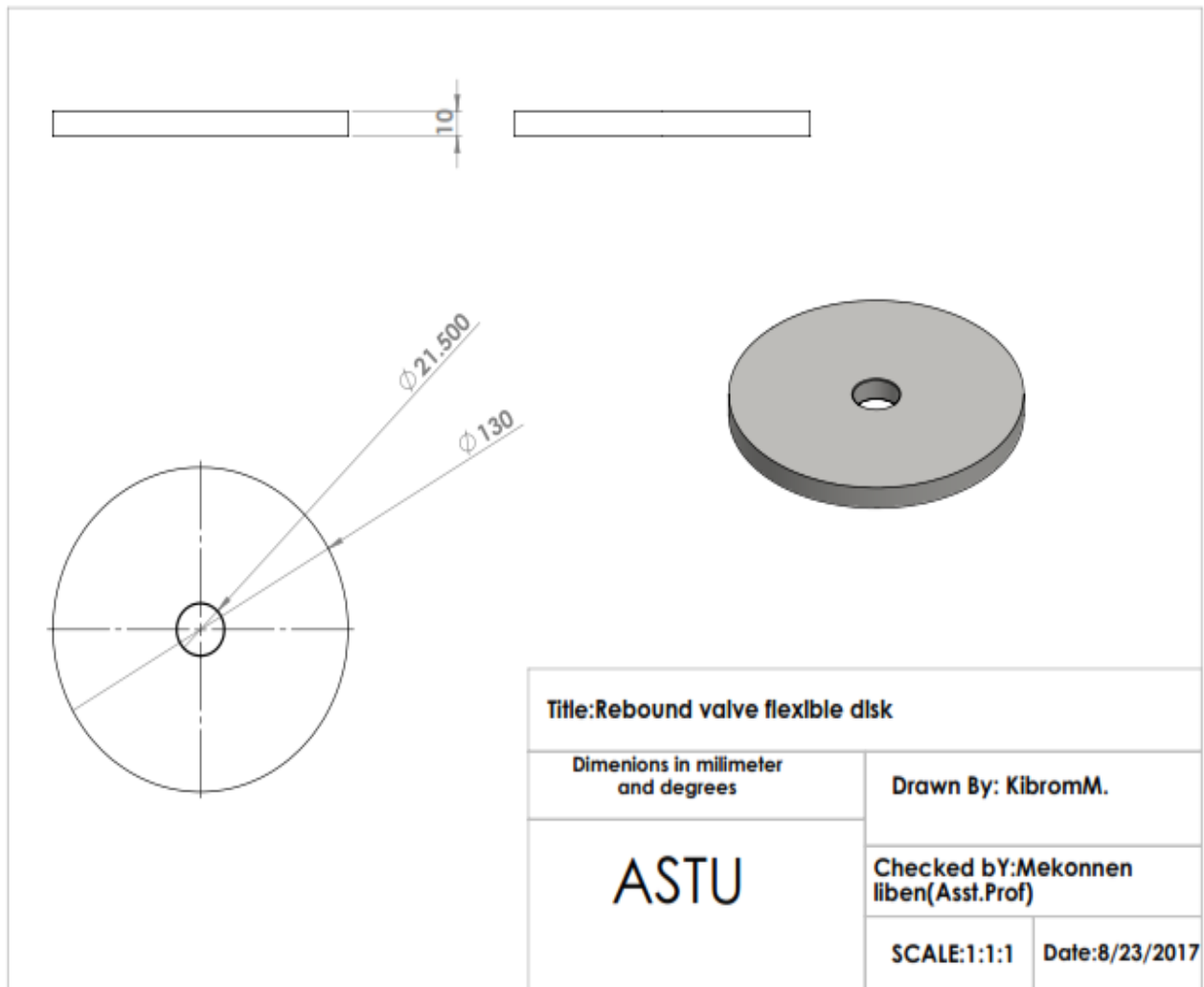
APPENDIX-H



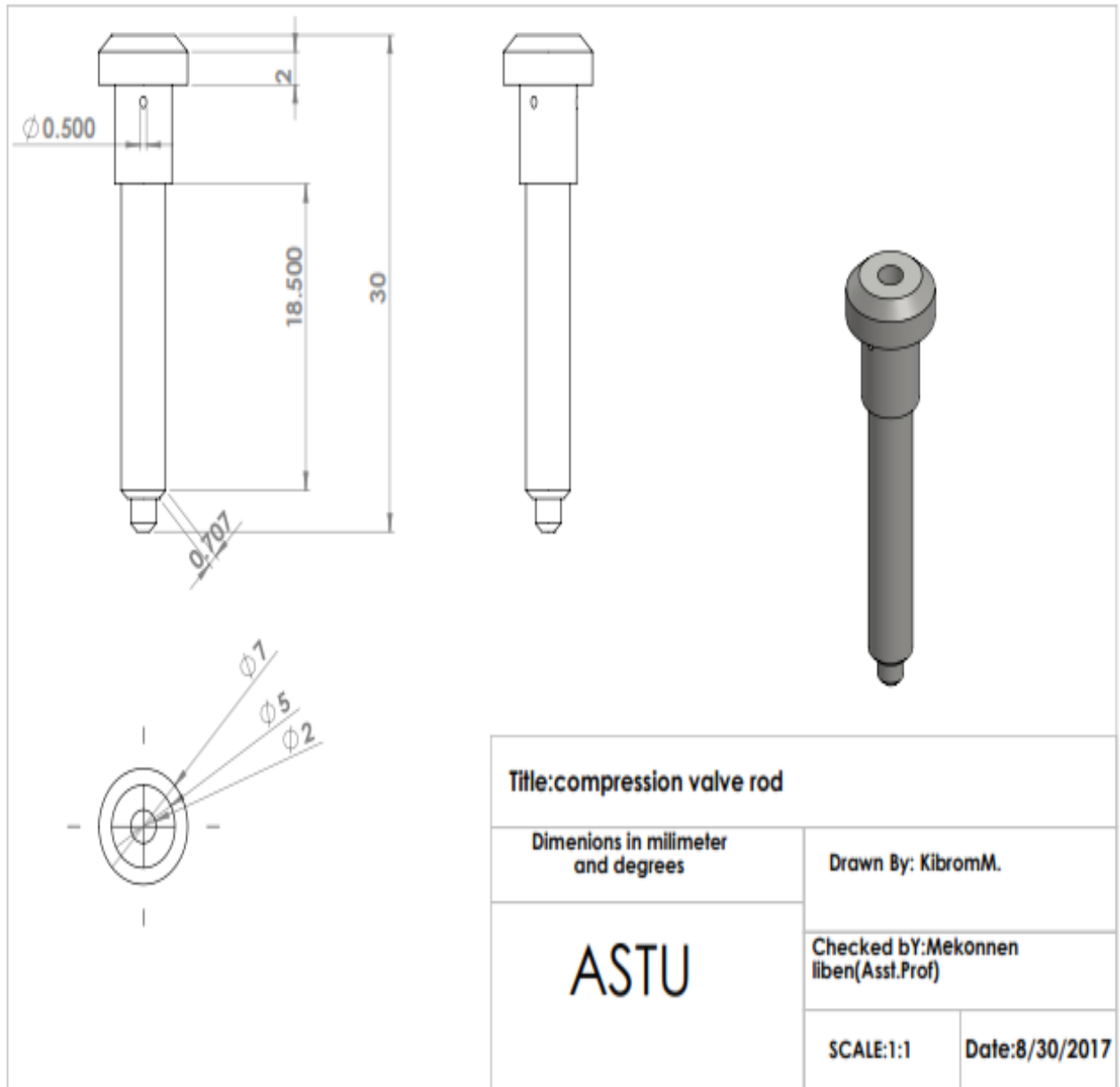
APPENDIX-I



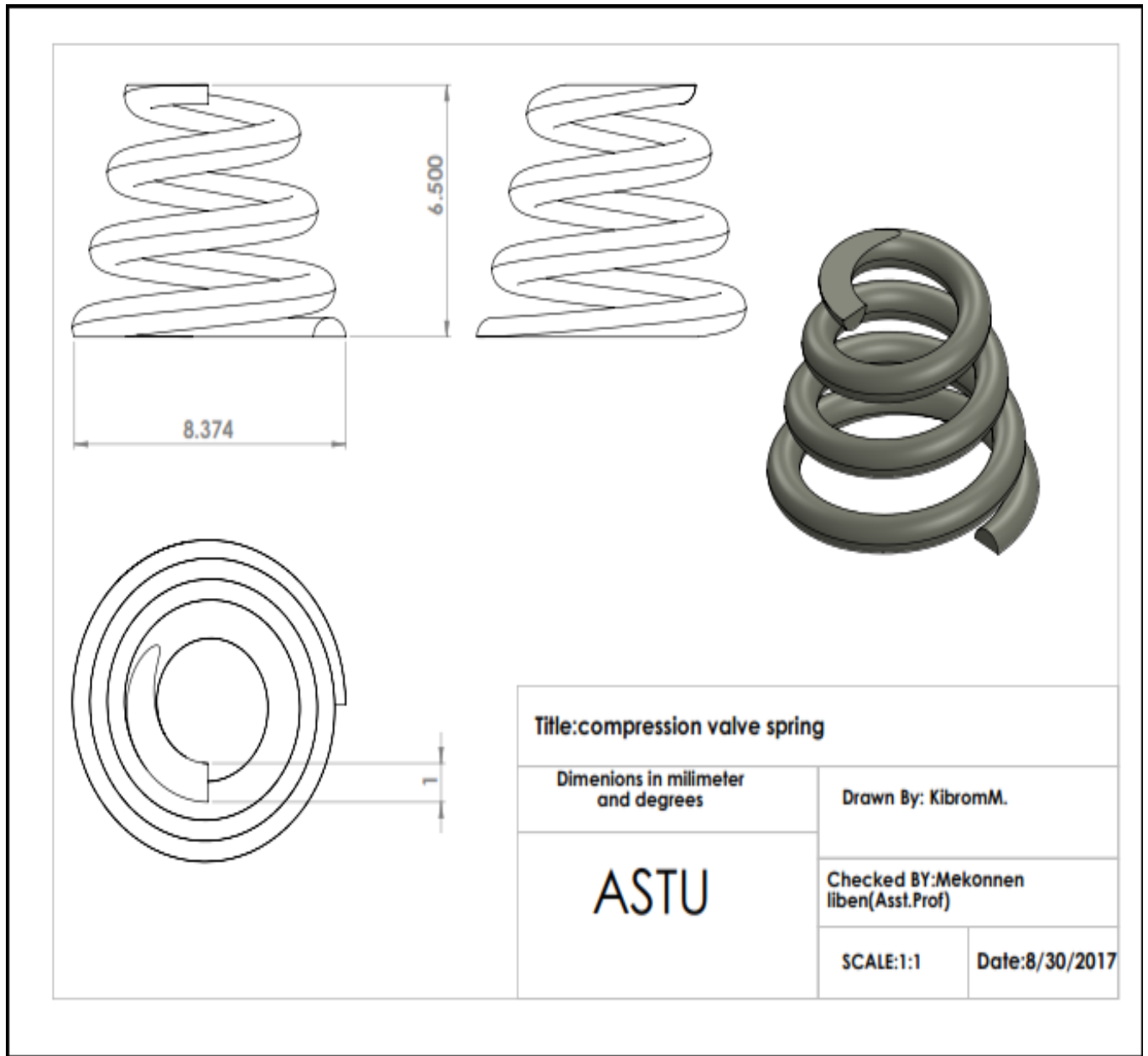
APPENDIX-J



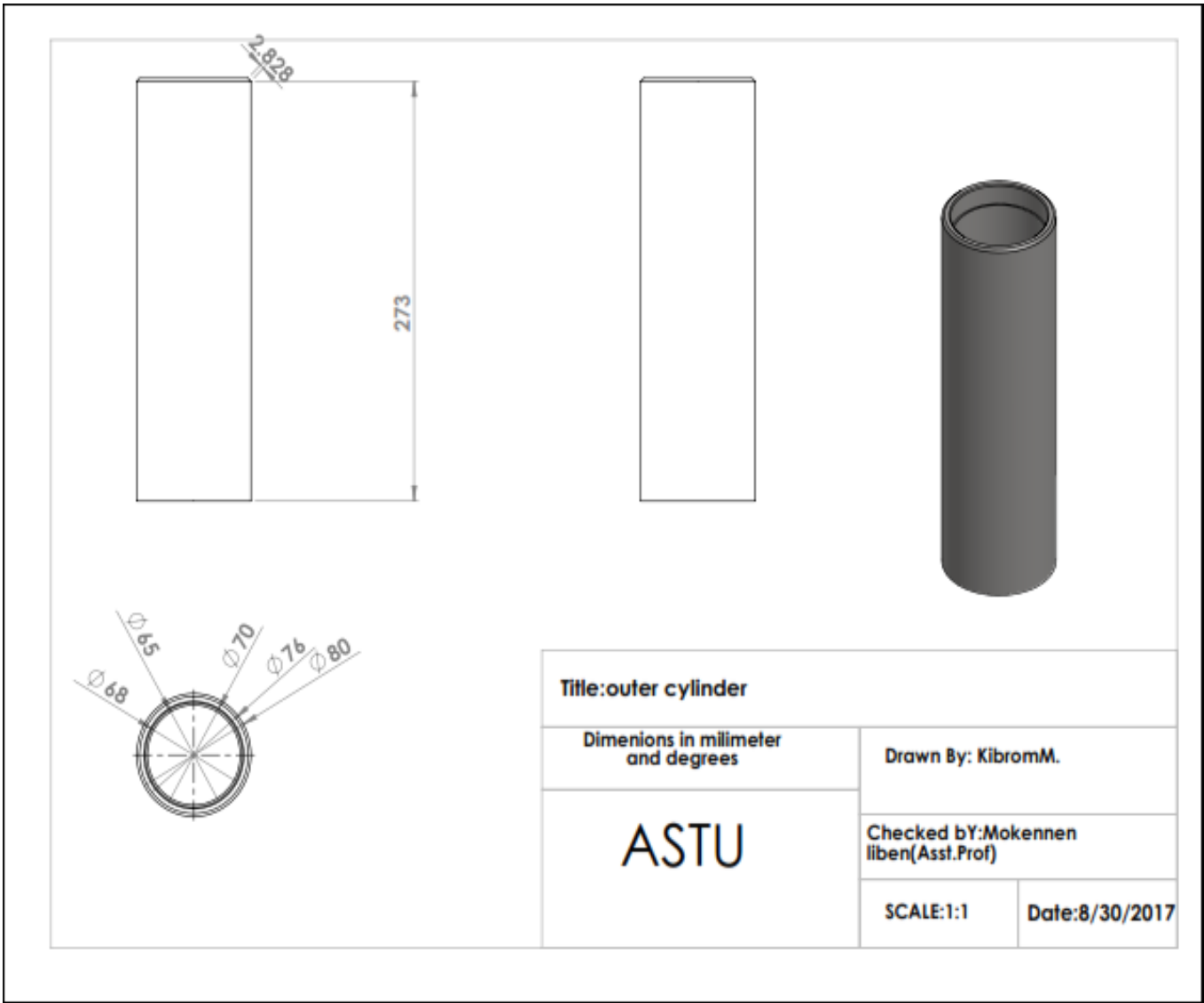
APPNDIXE-K



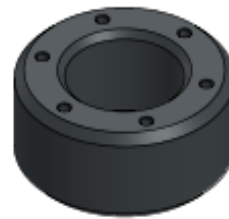
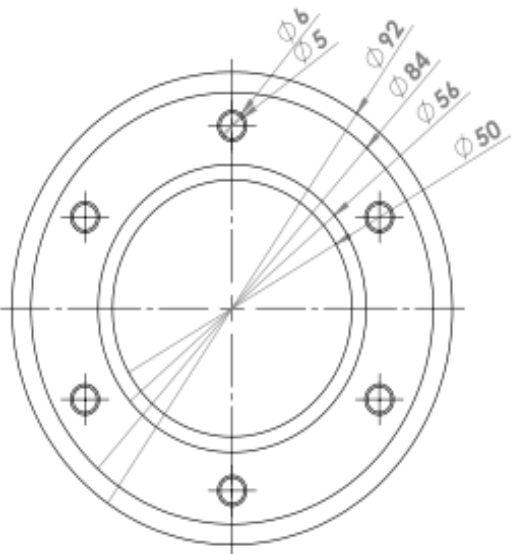
APPENDIX-L



APPENDIX-M



APPENDIX-N



Title: lower mount

Dimensions in millimeter
and degrees

Drawn By: KibromM.

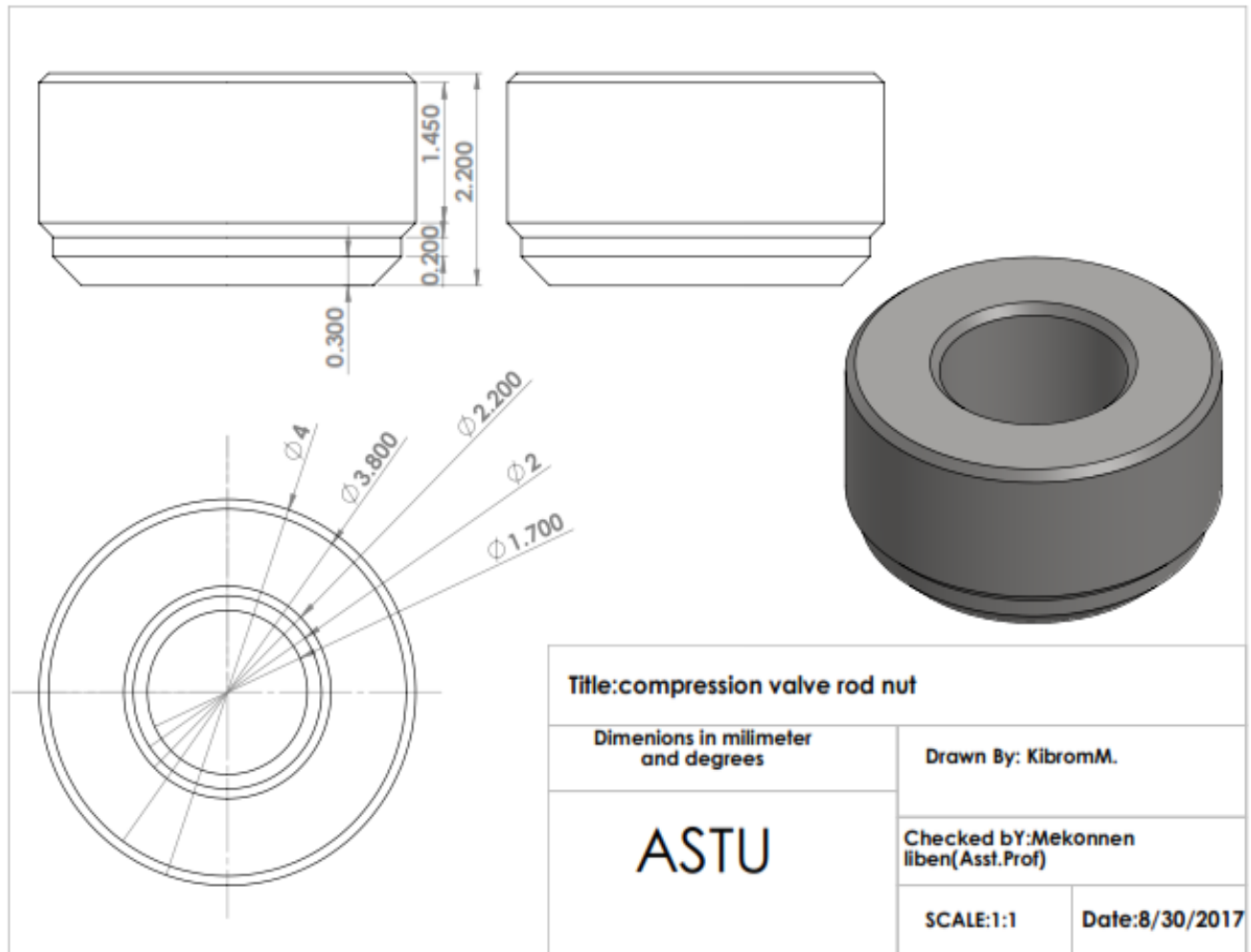
ASTU

Checked by: Mokenen
liben (Asst. Prof)

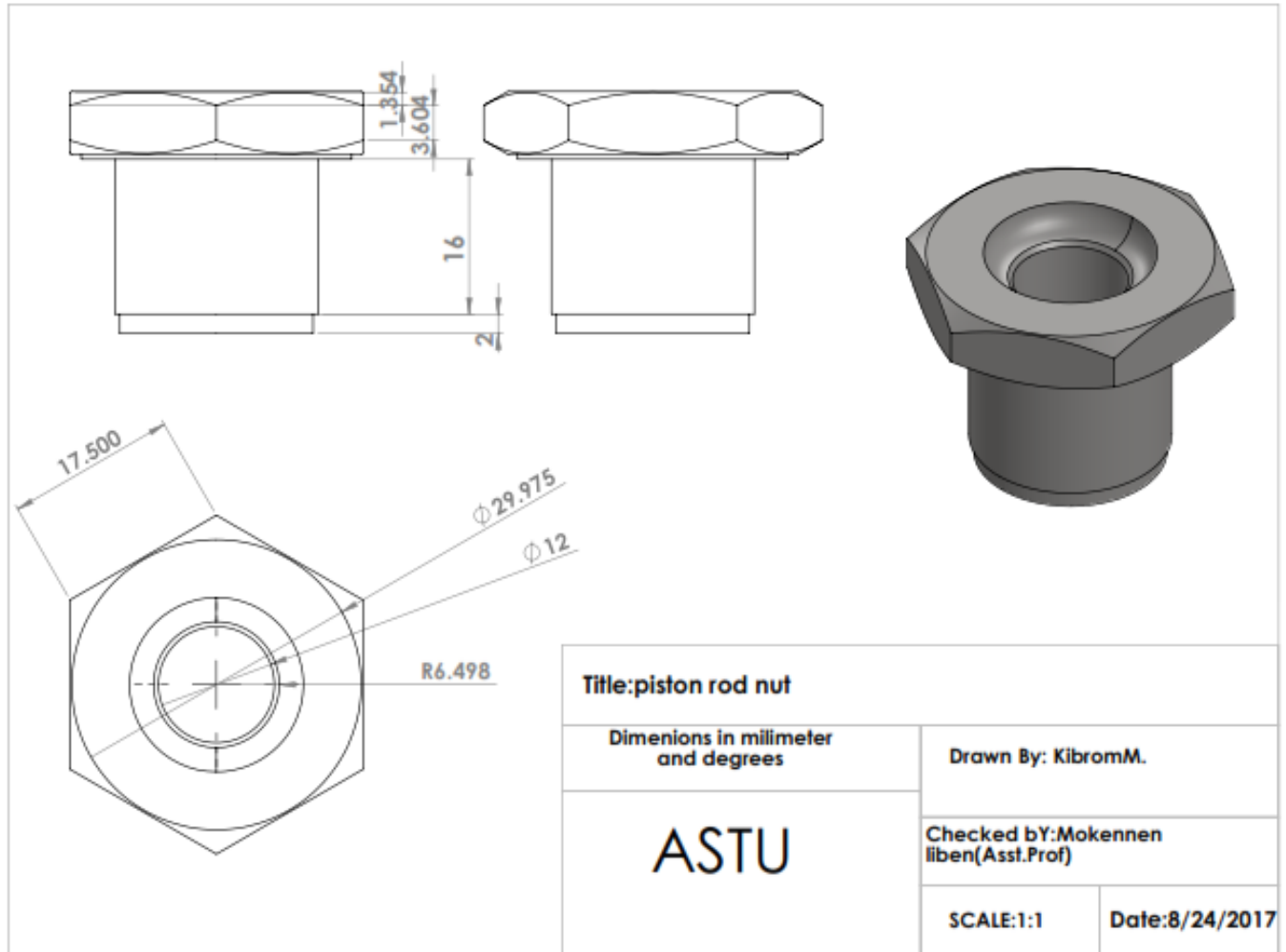
SCALE: 1:1

Date: 8/24/2017

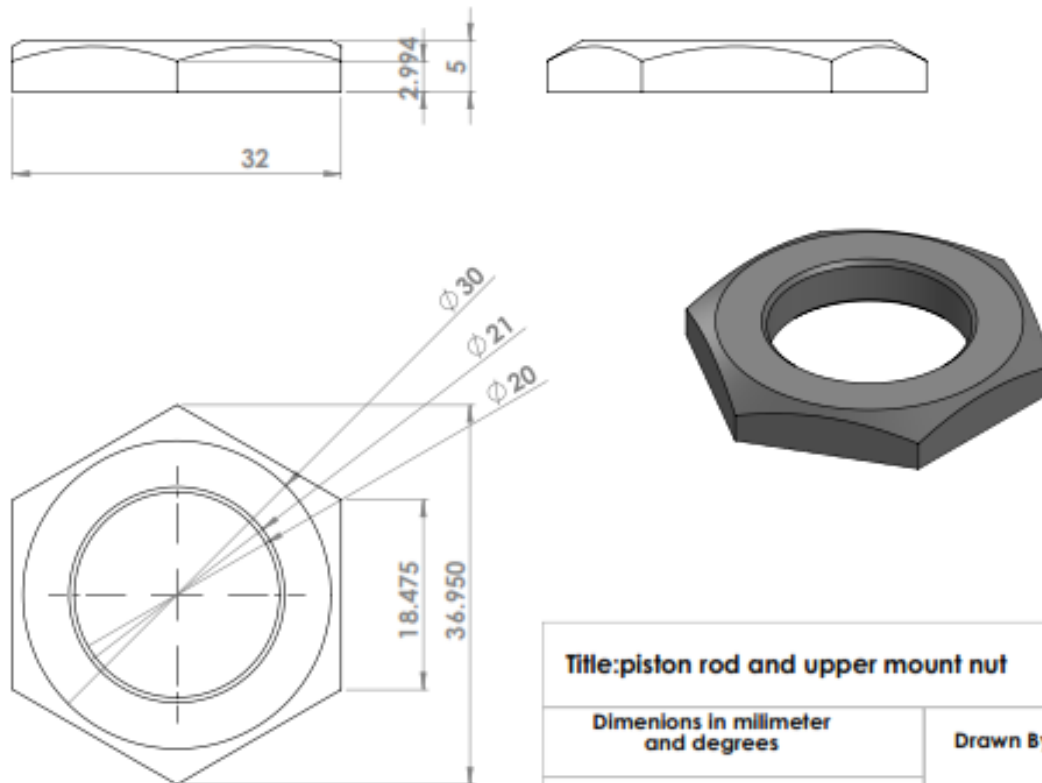
APPENDIX-O



APPENDIX-P



APPENDIX-Q



Title:piston rod and upper mount nut

Dimenions in millimeter
and degrees

Drawn By: KlbromM.

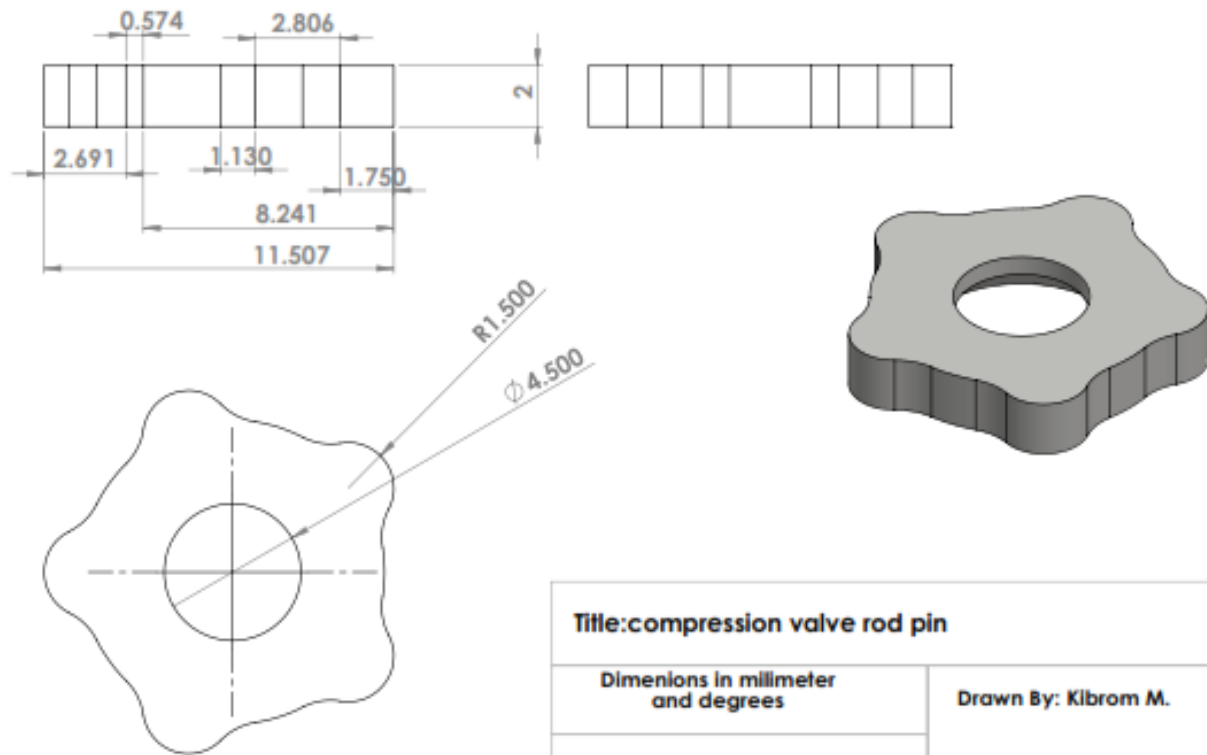
ASTU

Checked bY:Mokennen
liben(Asst.prof)

SCALE:1:1

Date:8/24/2017

APPENDIX-R



Title:compression valve rod pin

Dimenlons in millimeter
and degrees

Drawn By: Kibrom M.

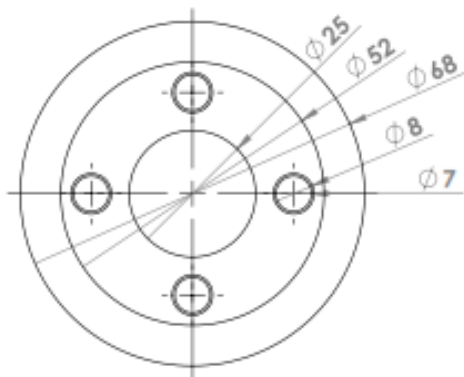
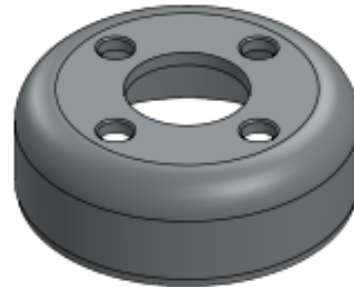
ASTU

Checked bY:Mokennen
liben(Asst.Prof)

SCALE:1:1

Date:8/30/2017

APPENDIX-S



Title:oil filter

**Dimenions in millimeter
and degrees**

Drawn By: KibromM.

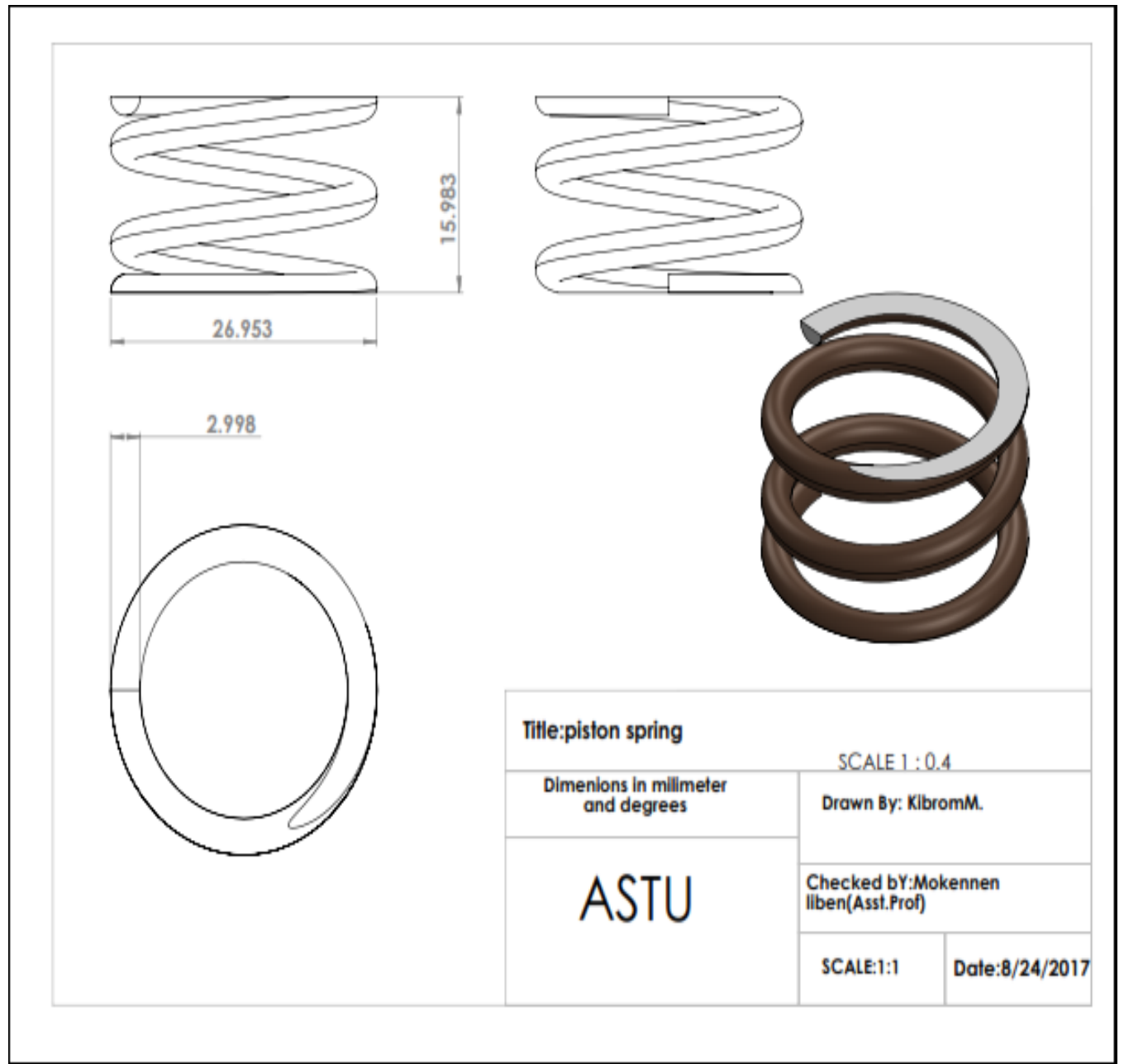
ASTU

**Checked bY:Mokennen
liben(Asst.Prof)**

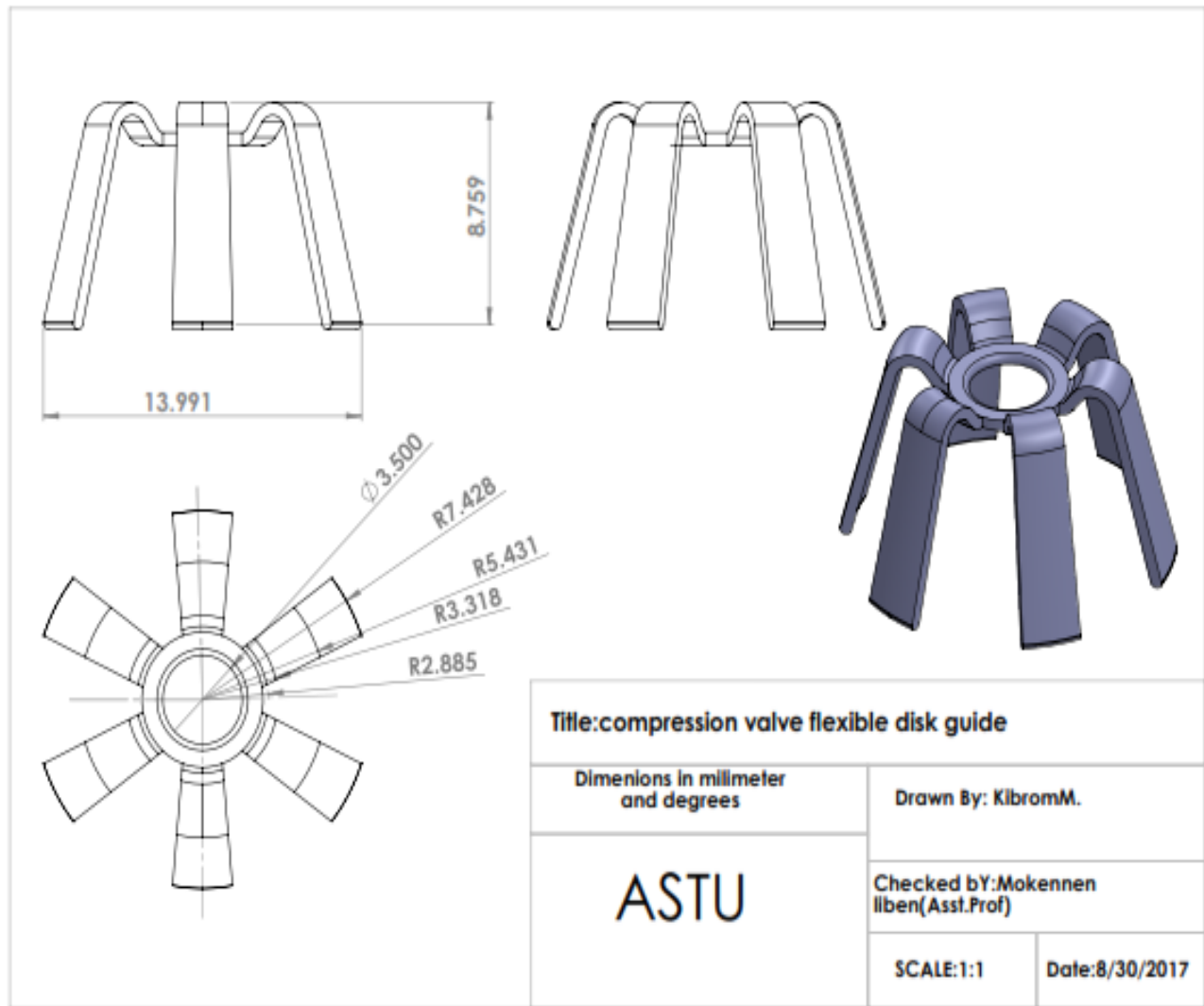
SCALE:1:1

Date:8/24/2017

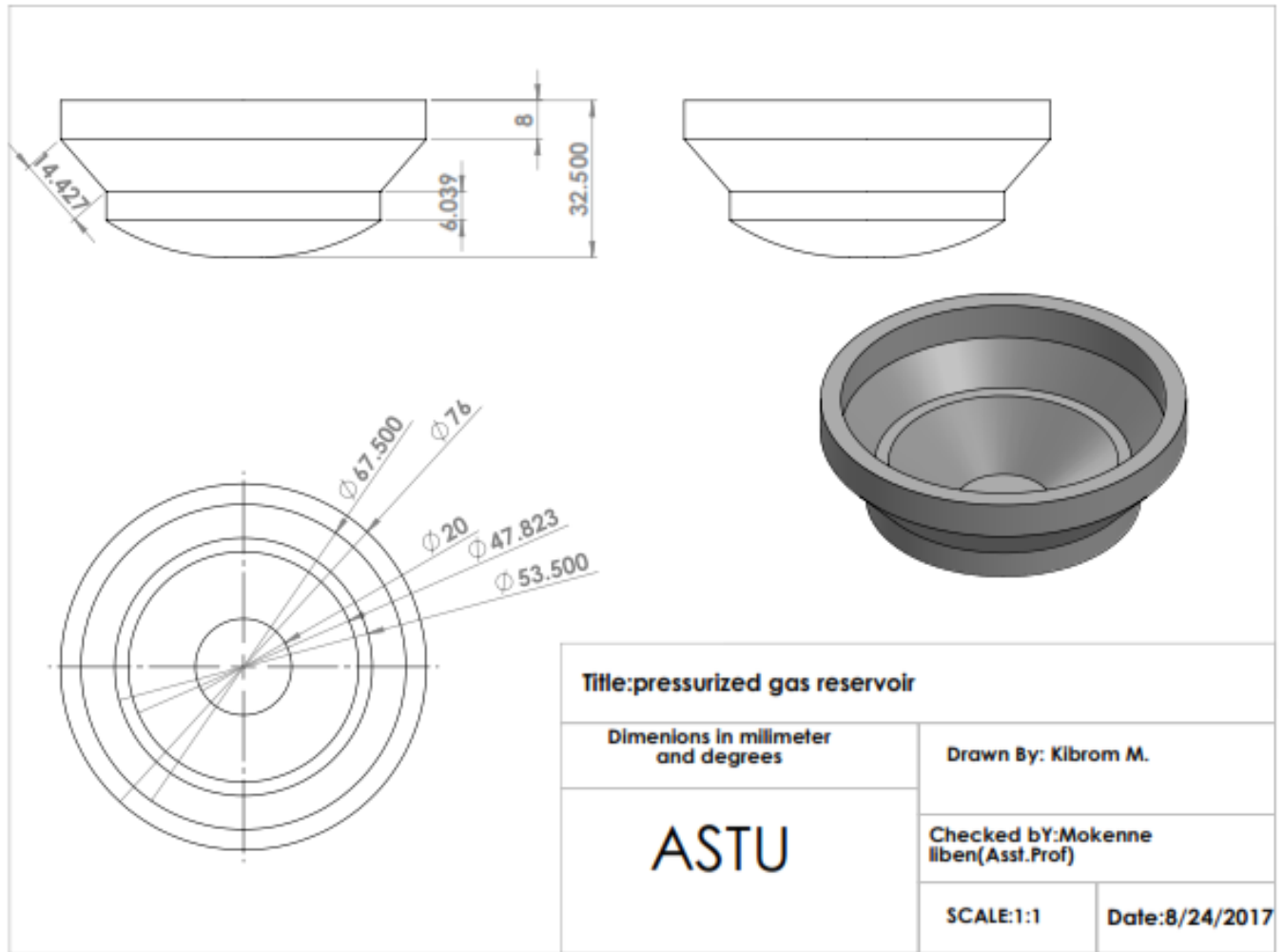
APPENDIX-T



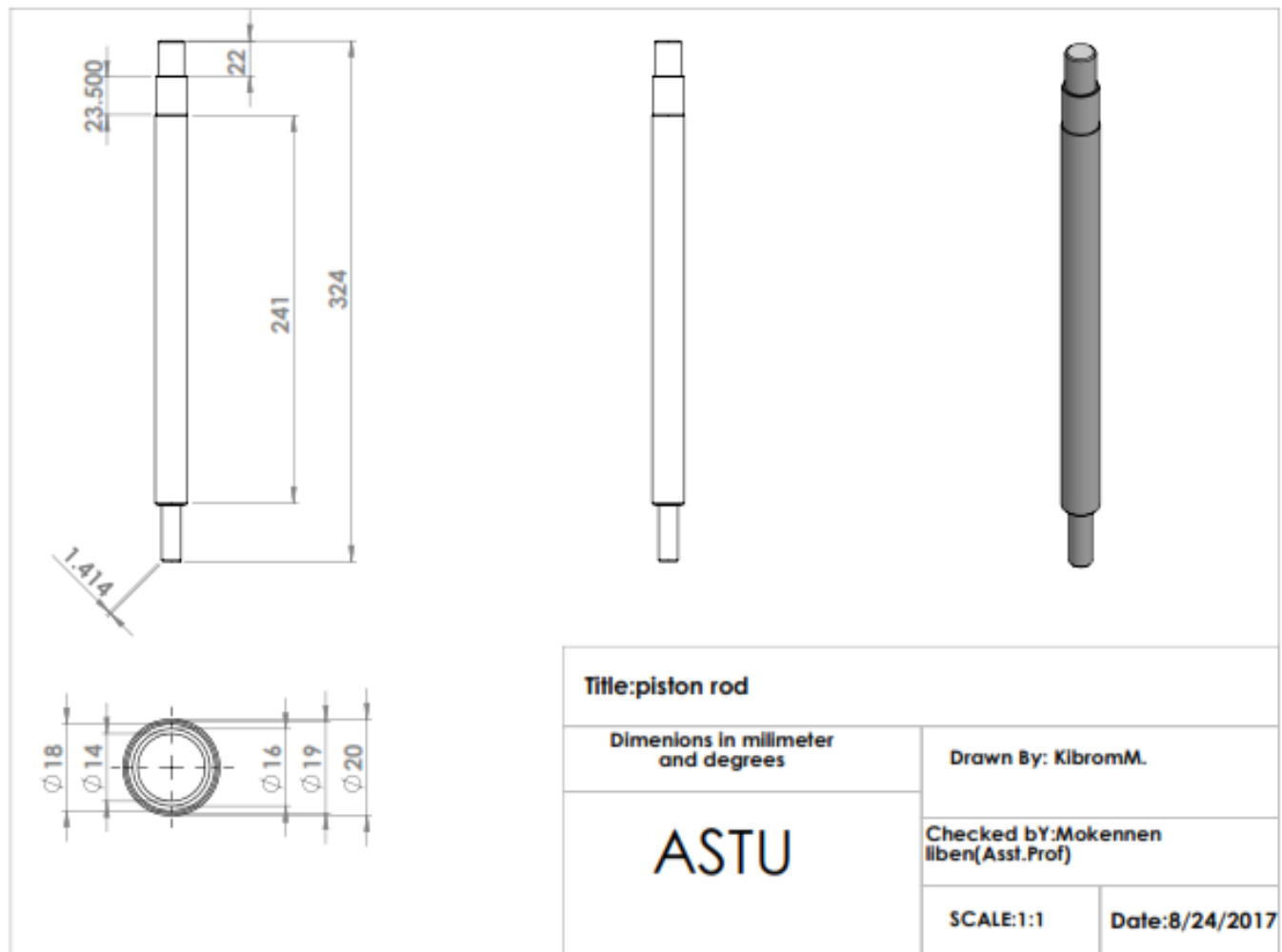
APPENDIX-U



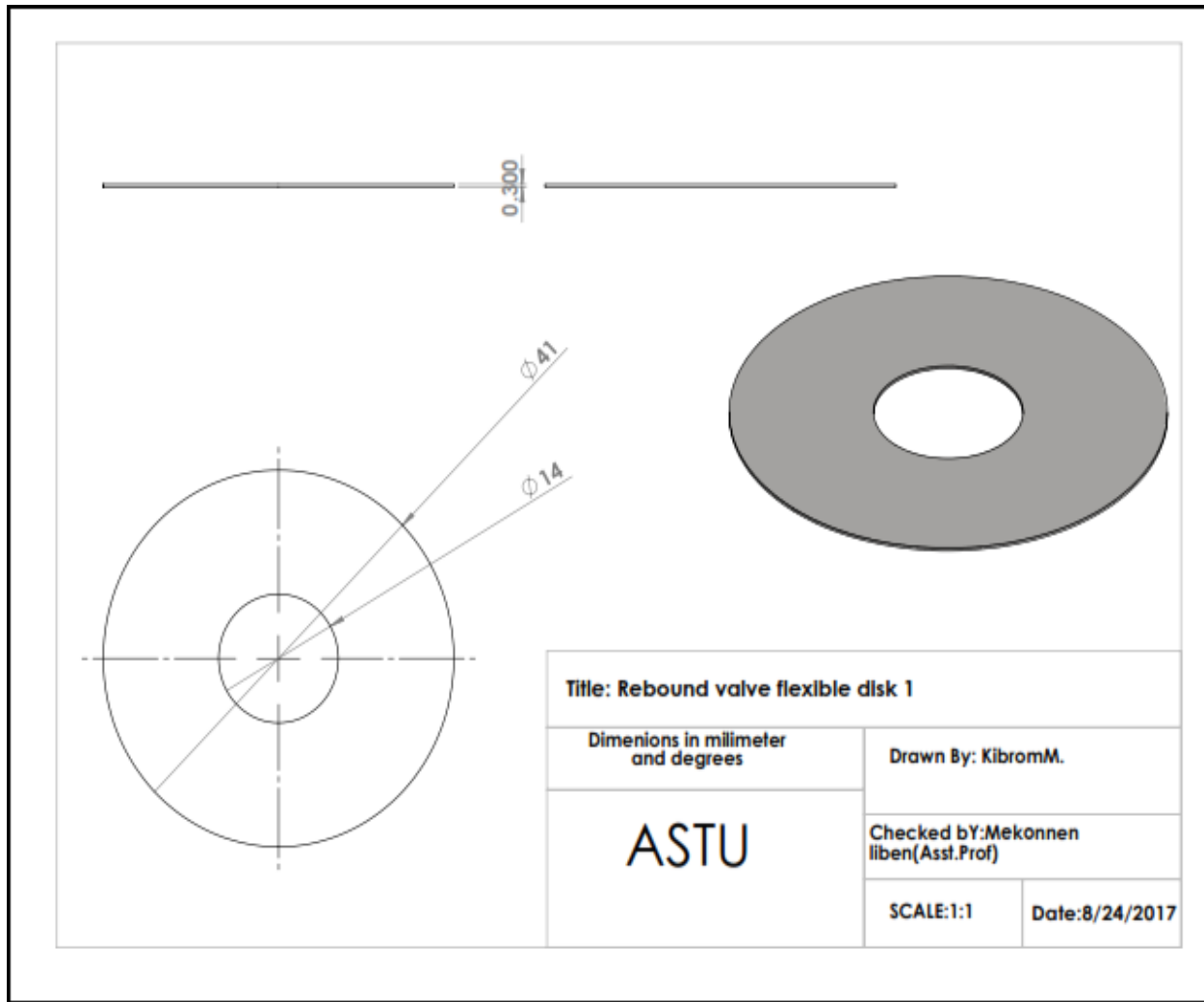
APPENDIX-V



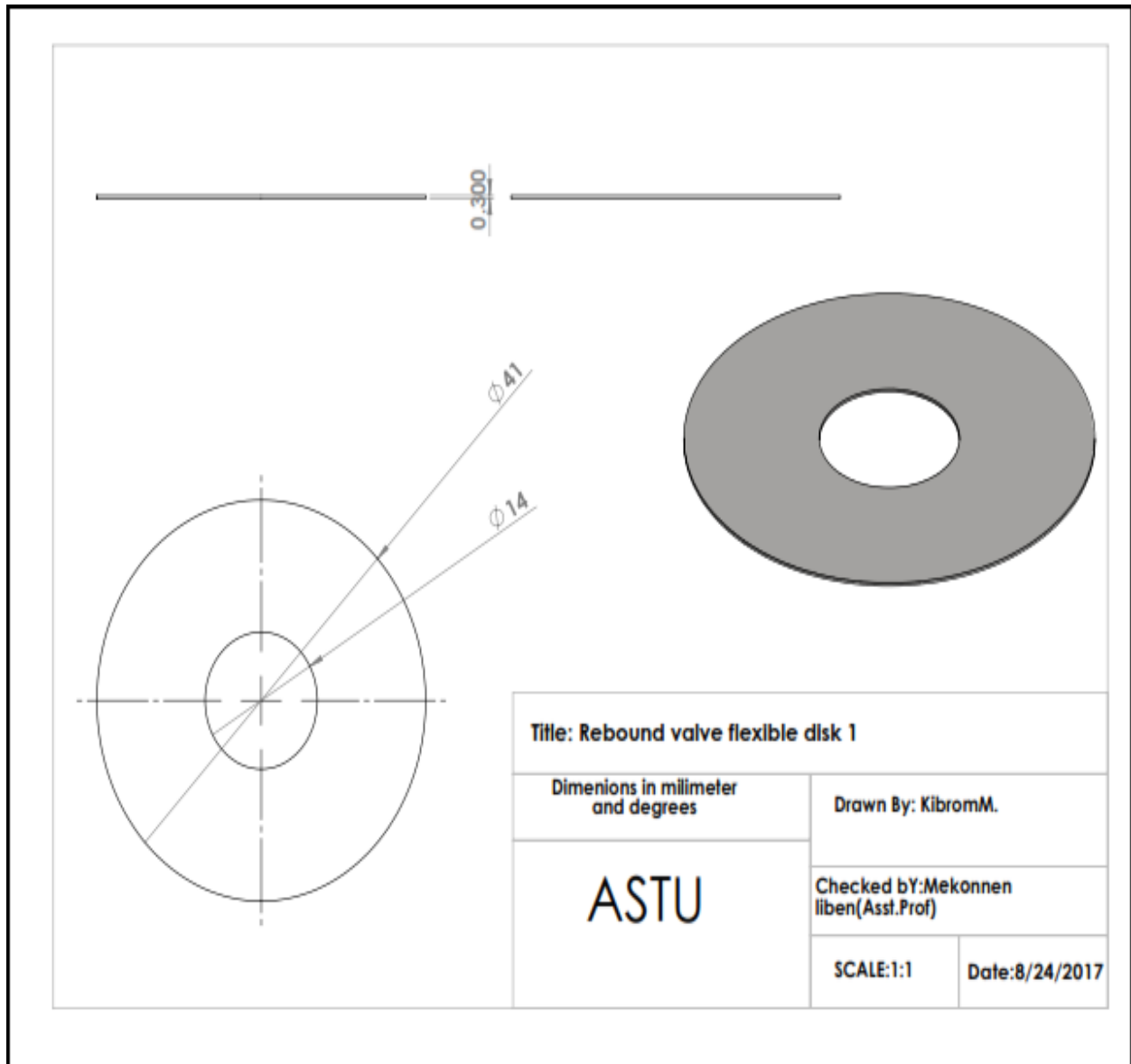
APPENDIX-W



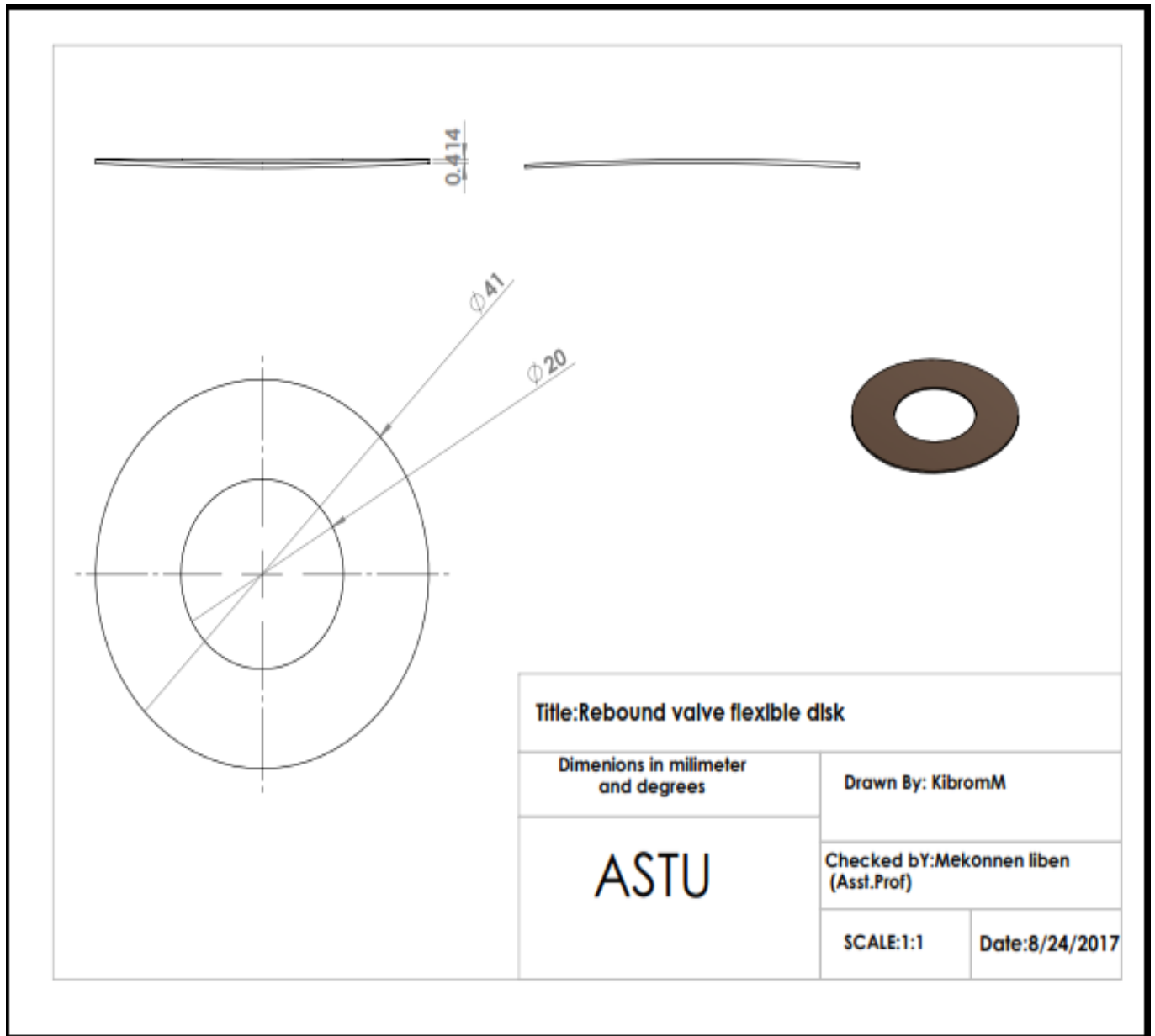
APPENDIX-X



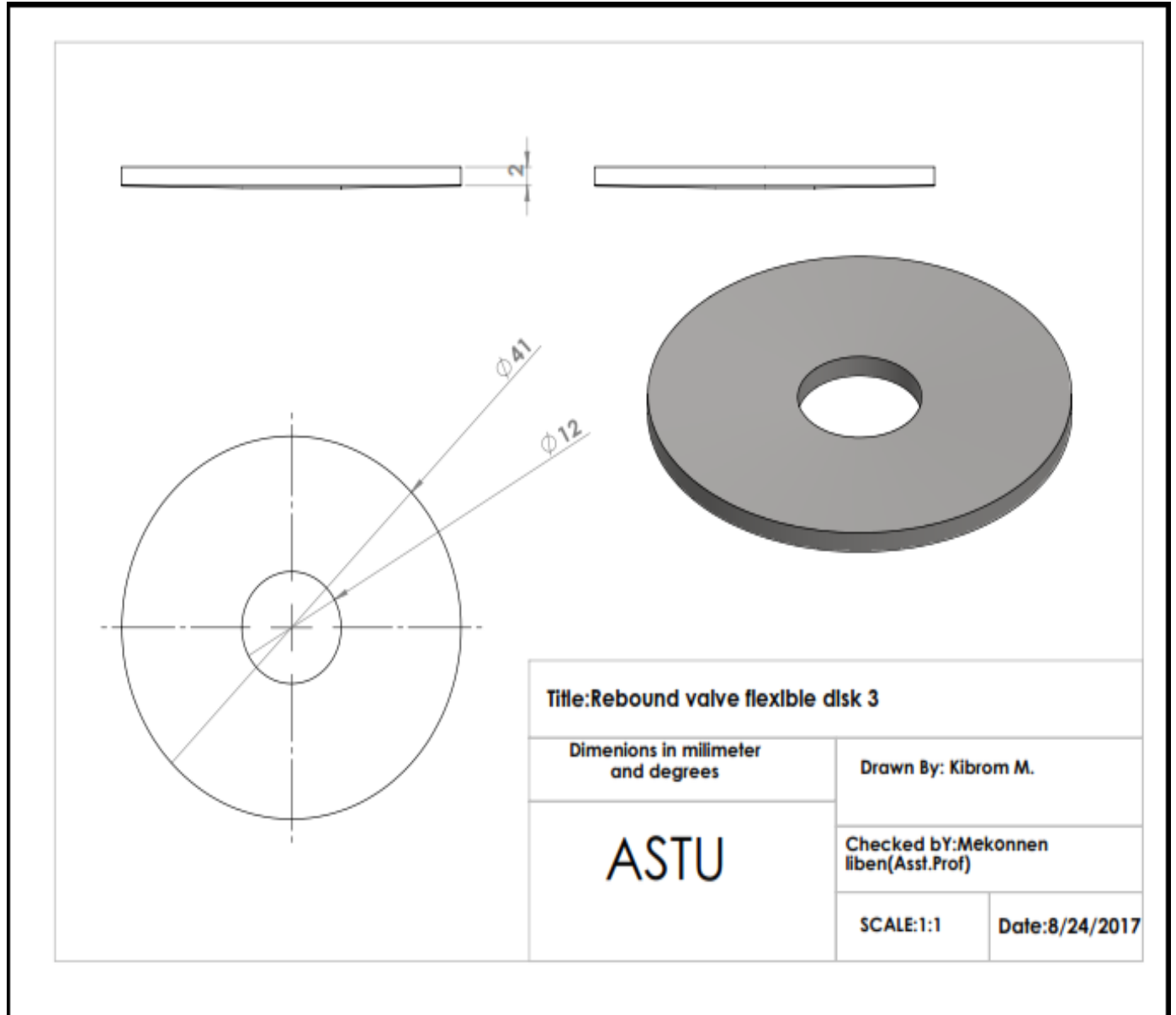
APPENDIX-Y



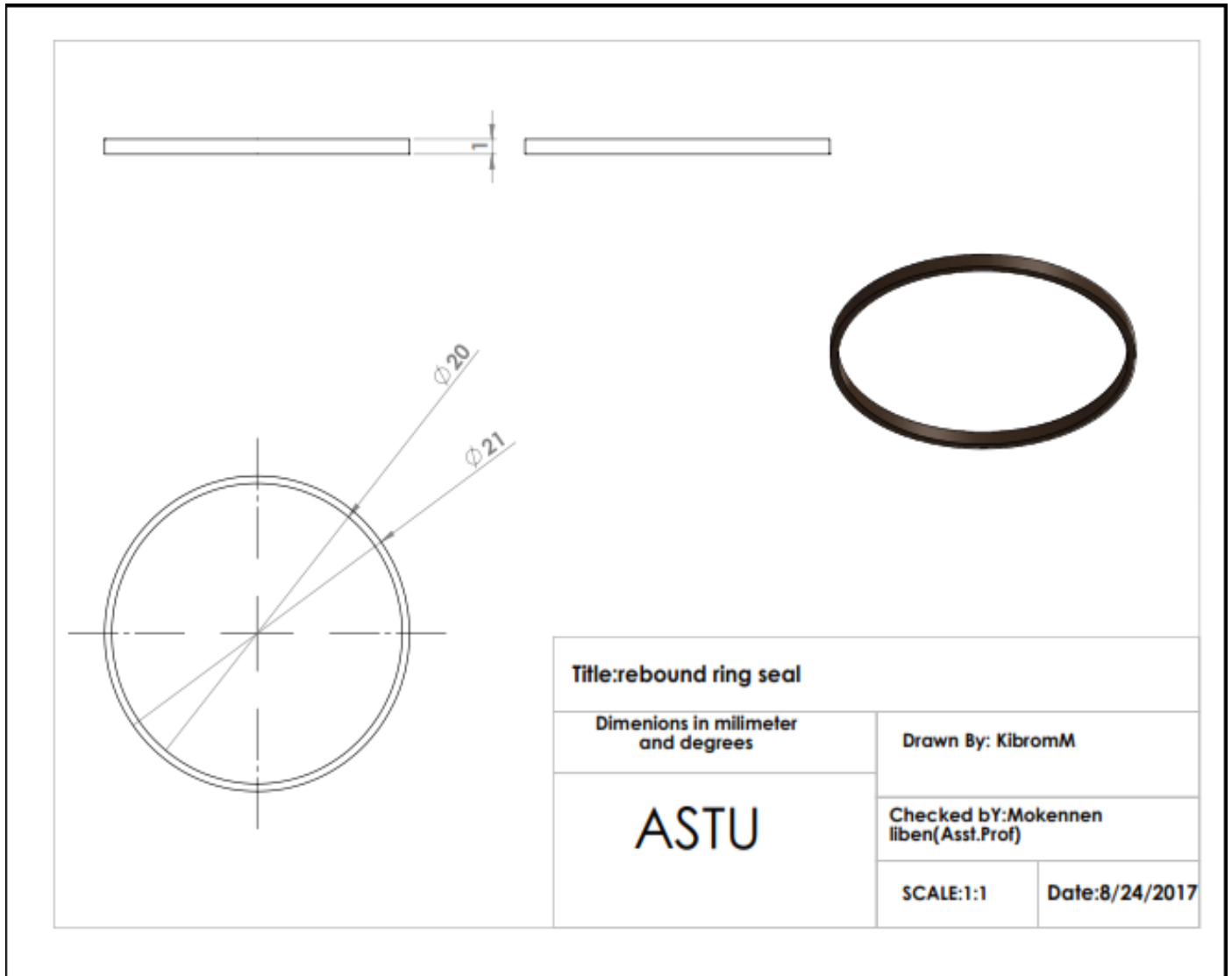
APPENDIX-Z



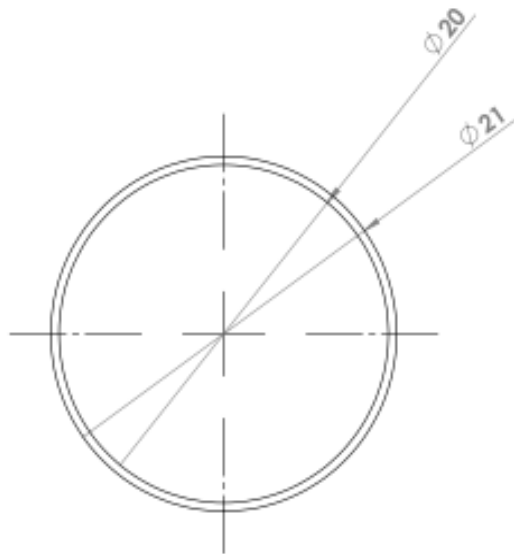
APPENDIX-A2



APPENDIX-B2



APPENDIX-C2



Title:rebound ring seal 2

**Dimenions in millimeter
and degrees**

Drawn By: KibromM

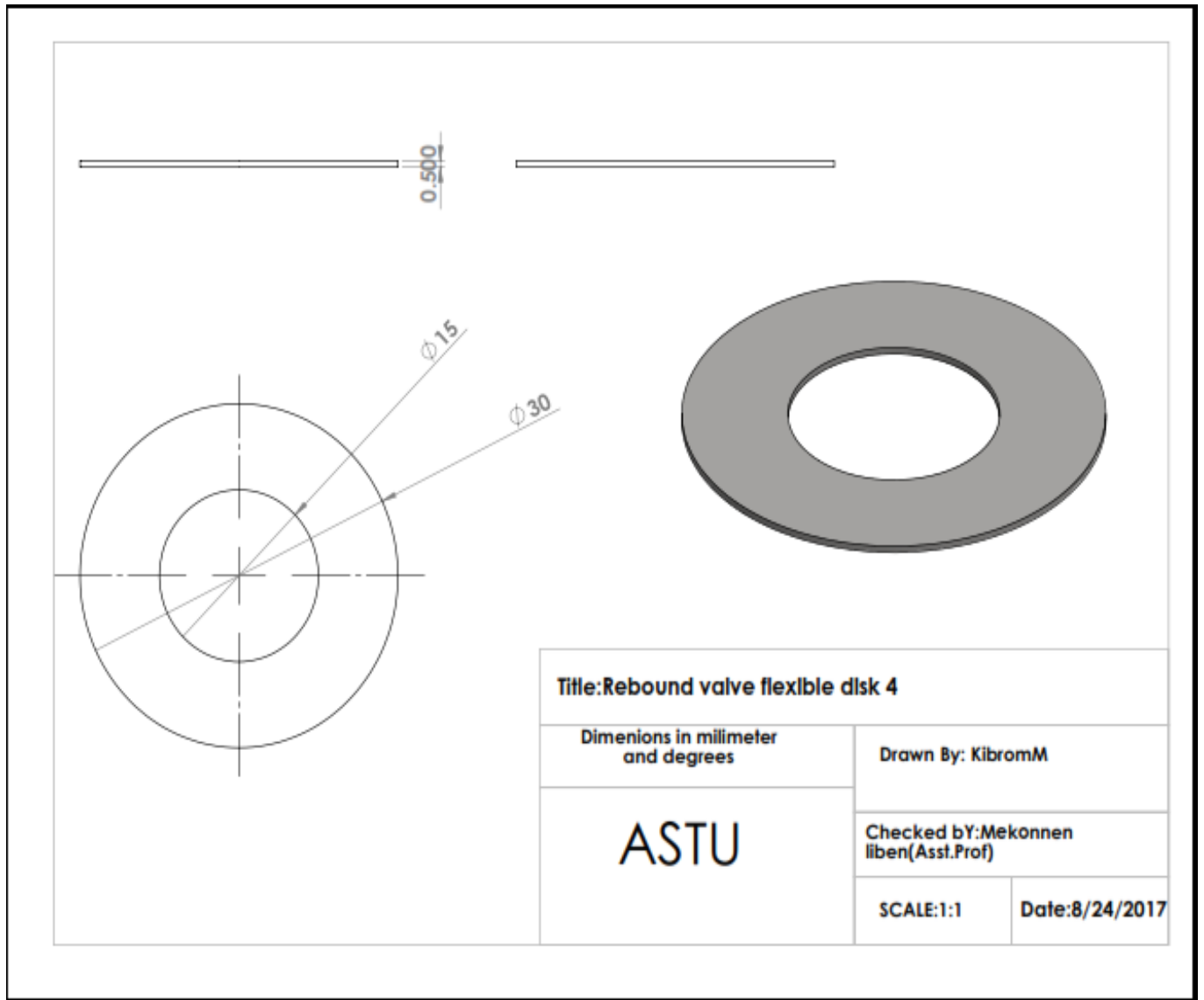
ASTU

**Checked by:Mokennen
liben(Asst.Prof)**

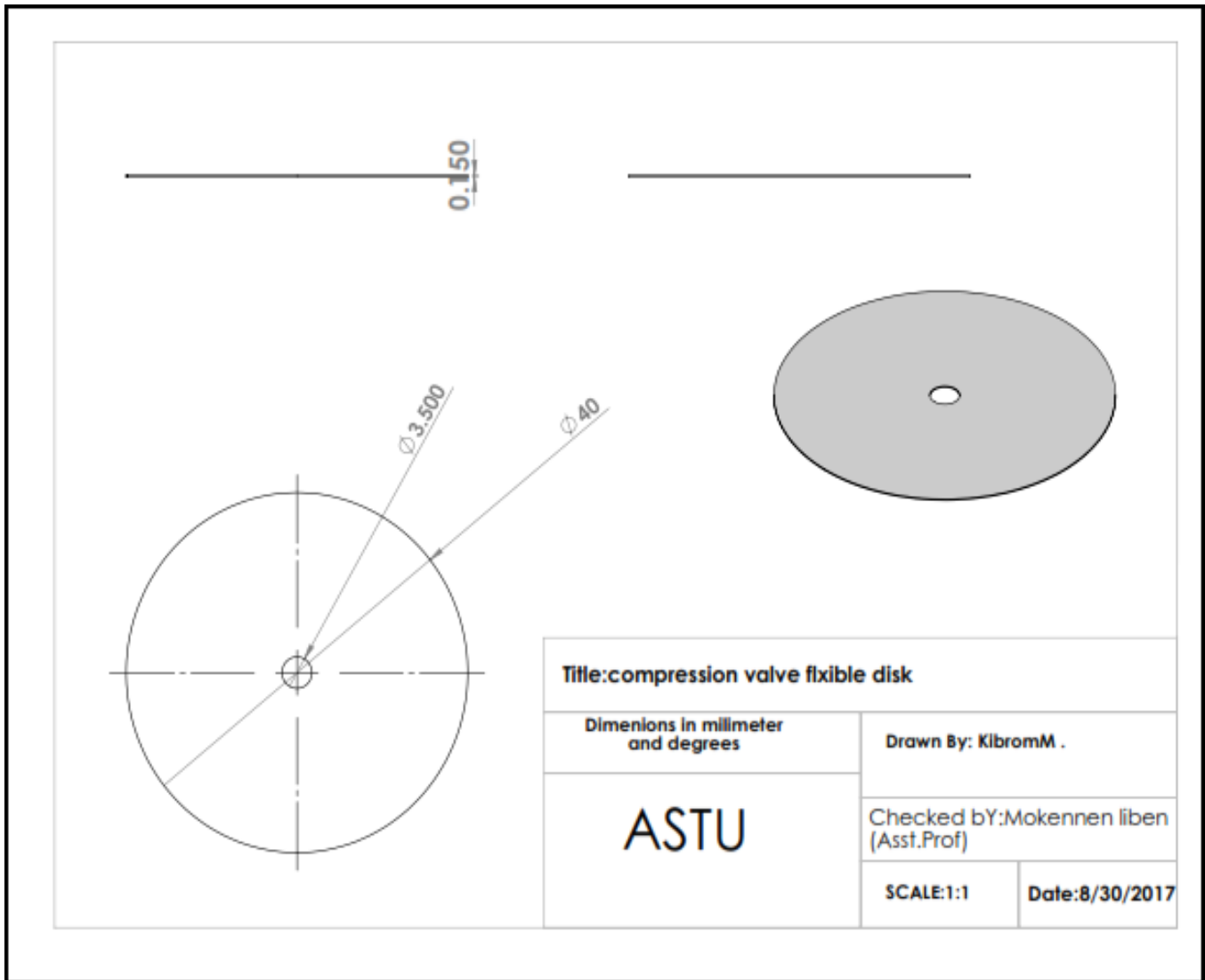
SCALE:1:1

Date:8/24/2017

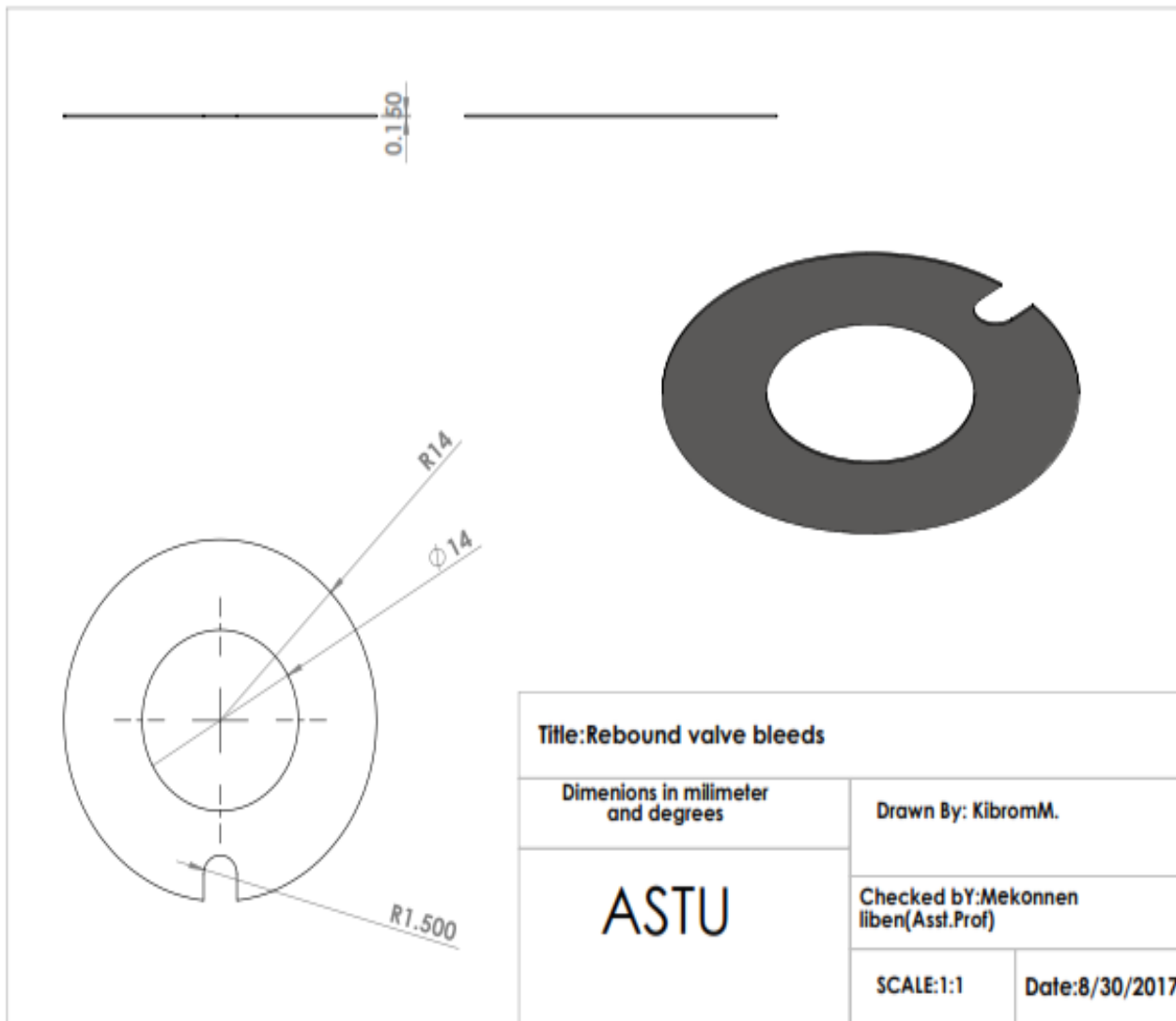
APPENDIX-D2



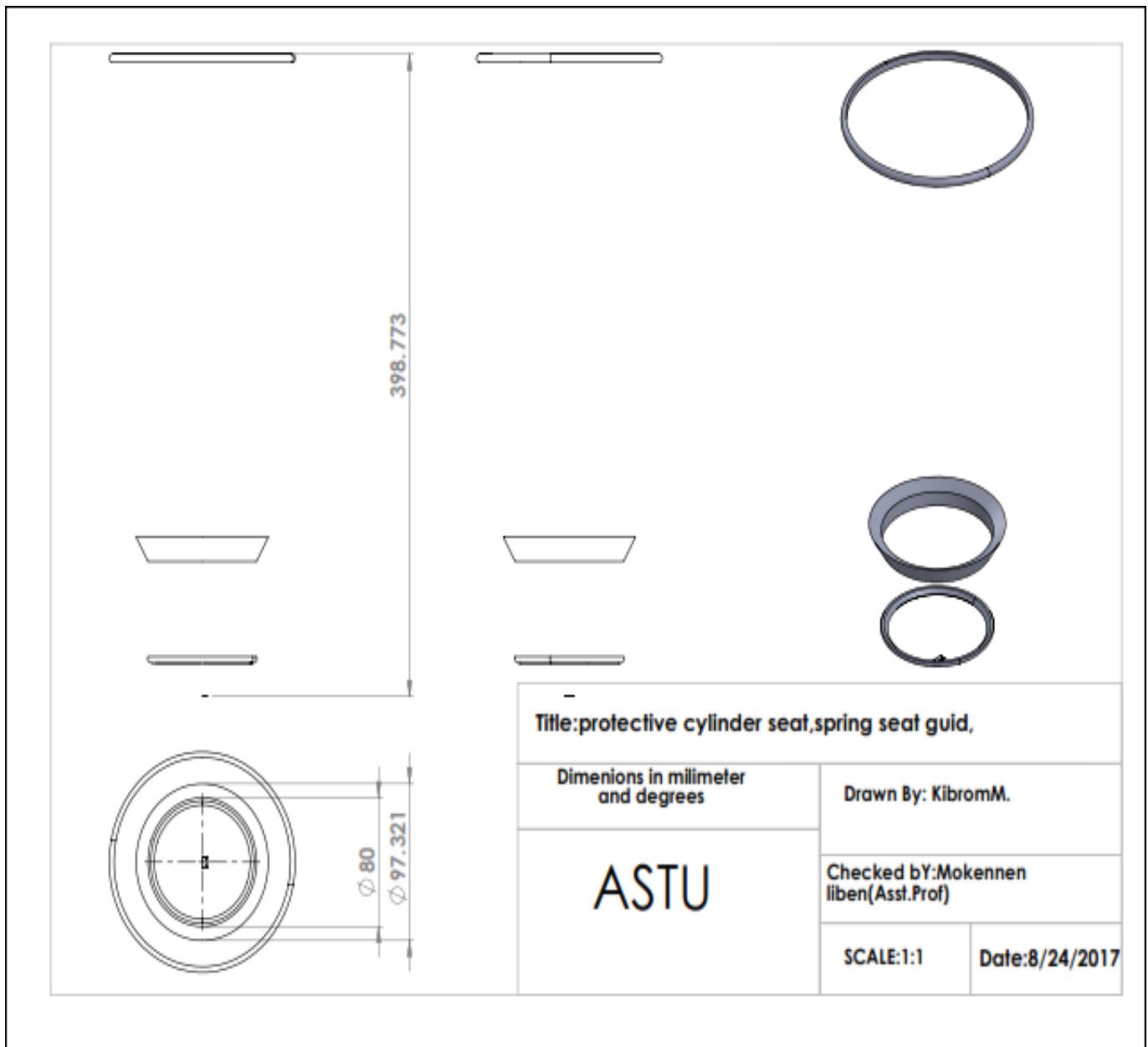
APPENDIX-E2



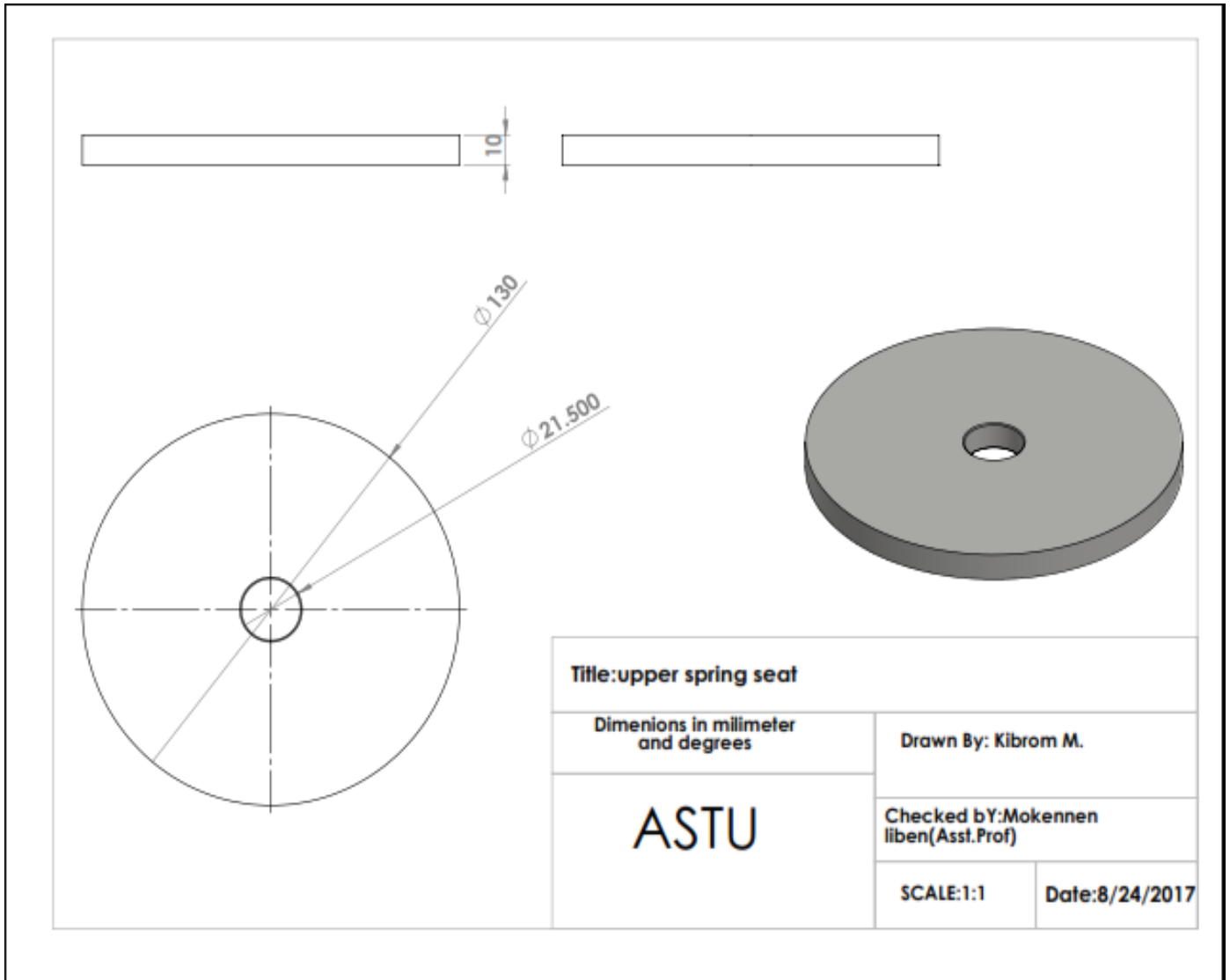
APPENDIX-F2



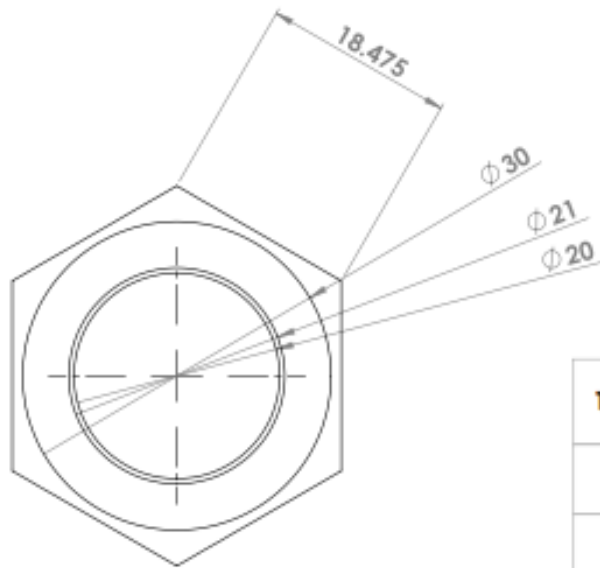
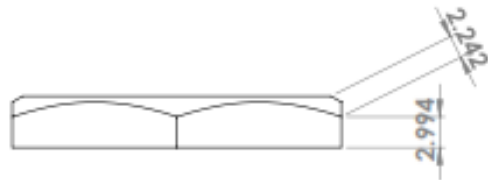
APPENDIX-G2



APPENDIX-H2



APPENDIX-I2



Title:upper spring seat nut

Dimenions in millimeter
and degrees

Drawn By: KibromM.

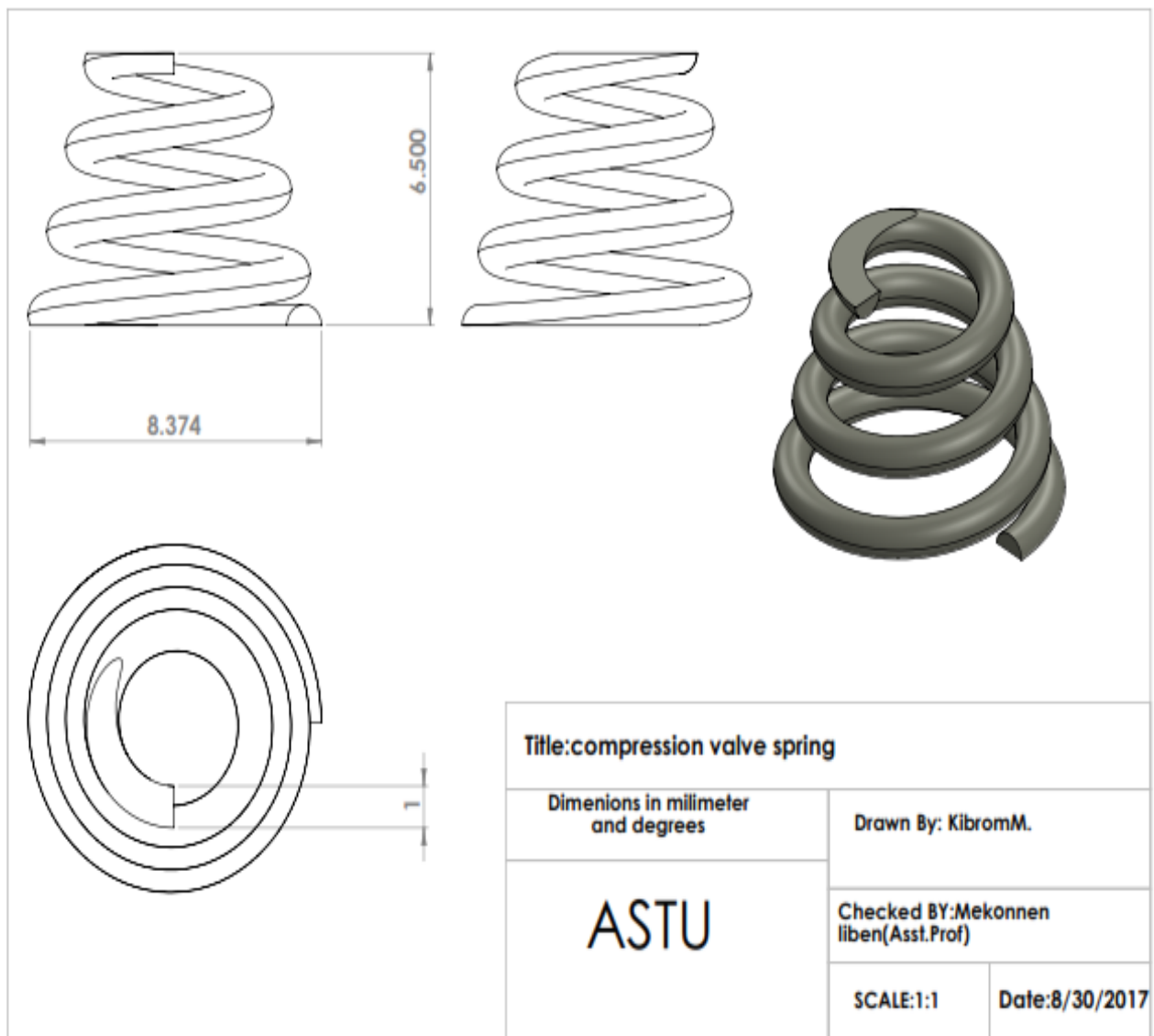
ASTU

Checked bY:Mokennen
liben(Asst.Prof)

SCALE:1:1

Date:8/24/2017

APPENDIX-J2



APPENDIX-K2

