DESIGN OPTIMIZATION OF DOUBLE WISHBONE SUSPENSION SYSTEM FOR MOTORCYCLE

Submitted in partial fulfilment of the requirements for the award of Bachelor of Engineering degree in mechanical engineering

By

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BONAFIDE CERTIFICATE

This is to certify that this Project Report is the bonafide work of **PARTHIV MANEPALLI (37150134)** and **NARENDRAN. R (37150127)** who carried out the project entitled **"DESIGN OPTIMIZATION OF DOUBLE WISHBONE SUSPENSION SYSTEM FOR MOTORCYCLE"** under my supervision from November 2020 to March 2021.

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We, **PARTHIV MANEPALLI, NARENDRAN. R,** Hereby declare that the Project report entitled "**DESIGN OPTIMIZATION OF DOUBLE WISHBONE SUSPENSION SYSTEM FOR MOTORCYCLE"** done by us under the guidance of **Dr . G. Arun Kumar, M.E, Ph. D** at Sathyabama Institute Of Science And Technology is submitted in partial fulfilment of the requirements for the award of Bachelor of Engineering degree in **MECHANICAL ENGINEERING**

1.

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ABSTRACT

Double wishbones are the most ideal suspension system. It is designed in such a way that it can handle wider variety of terrain. Generally this type of suspension is used in four wheelers. The double wishbone suspension system can also be used in two wheelers with a modified design which is optimized according to the performance of automobile. In this project we optimized the double wishbone suspension system by replacing existing suspension system. The cruiser bike is assembled with double wishbone suspension system and is integrated with a single spring damping unit which is linked with Upper and Bottom Yoke. This suspension system assembly is described brief in the further sections and its optimization process. This new system results in perfect caster and camber control, weight of the suspension system is optimized, wear and tear of the tires are controlled, wheels in contact with the road, better handling, vertical and lateral load can be tackled. These factors will be checked before the optimization of design and the results will be in decrease of weight. Autodesk fusion 360 is used for modelling and simulating the double wishbone suspension geometry.

Index terms – Suspension system, chamber, design, automobile, Shock absorber, optimized, wear, tear, modelling, geometry, weight, load, software, simulating, caster control

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LIST OF SYMBOLS AND ABBEREVIATIONS

Al	Aluminum
m^3	Cubic metre
%	Percentage
wt.%	Percentage by weight
MPa	Mega Pascal
Rpm	Revolutions Per Minute
CAD	Computer Aided Design
m/s	Metre per second
Kg	Kilogram
N	Newton
Si	Silicon
Mg	Magnesium
mm	Milli meter

CHAPTER 1
INTRODUCTION

Whenever a two wheeler is build, the structure decides the first look of the two wheeler. Naturally two wheeler structure act on appearance, handling and safety of the assembled machine. Suspension system of all country vehicle is one of the most censorious system is need to be designed for more endurance and convenience for the driver.

1.1 DOUBLE WISHBONE SUSPENSION SYSTEM

Double wishbone suspension is the classic suspension system. This double wishbone suspension system can be used in both front and rear wheels in car but in two wheeler only front wheel can be used. The double wishbone suspension system is used in century of 1934. The first double wishbone suspension system is used in French car maker Citroen. Double wishbone suspension system have a supercilious dynamic characteristics and this double wishbone suspension system is mostly found in the high range model two wheelers. The double wishbone suspension system can also called by another name is double A-shaped wishbone suspension system.

A Suspension system is an assembly of steering system and chassis. This helps the rider from tremble, jerks etc. Types of suspension system used in two wheelers are telescopic suspension system, Up-Side down suspension system and double wishbone. Double wishbone suspension system interacts with wheels and proceed separately from the other types. Control arms are connected with wheel and frame of the two wheeler at the ends. This suspension system helps during cornering by this the tire will be always in-contact to the ground which gives camber control. This double wishbone suspension system is assembled in cruiser two wheelers.

1.2 ADVANTAGES OF BOUBLE WISHBONE SUSPENSION SYSTEM

Advantages of this double wishbone suspension system are camber control, long suspension travel, flexible, compact. When compare to telescopic suspension system, double wishbone is better than telescopic suspension system because double wishbone suspension has degree of freedom for dampers are more, during cornering for cruiser two wheeler more force will be acting where in telescopic suspension system has less dampers so that during cornering for two wheeler has less contact

1.3 OPTIMIZATION TECHNIQUE

The optimization technique used in this project is generative design. This Generation design is a repetitive design process which is a program generates a selective number of outcomes according to the constrains given. Where designs can be selected according to the material, factor of safety, stress and strain, weight, volume, deformation and manufacturing method etc. In this project the mass optimization are performed for parts of double wishbone suspension system are suspension fork and wishbone. Suspension system contains shock absorbers, springs, and linkages which connect its brakes and a motor vehicle. Important parameters of the system are from shocks Reaching an important part from Consumer satisfaction 18, it will help to isolate Freight and passengers. During auto avoiding actions that's rolling it provides stability. The two tires have been stimulate by it and steering supports camber angles along with the steer.

Figure 1.1: Double Wishbone Suspension system CAD Model

1.4 TYPES OF DOUBLE WISHBONE SUSPENSION SYSTEM

Two double wishbone systems are reviewed in this research as illustrative of these two groups they are Girder suspension system and the Hossack system. These two types of designs encloses most of the existing double wishbone suspension systems. The Girder suspension system comprise of a pair of extended uprights where the front wheel is attached to it. These uprights are connected to the fork by an higher and a lower wishbones which conduct the suspension motion. Both fork rotate about the steering axis which is firm to the two wheeler chassis. A spring-damper unit is usually linked between the bottom wishbone and the upper fork on condition that the shock absorption system. Otherwise, the Hossack suspension system consists of a double wishbone structure directly linked to the chassis. The two wishbones rotate the both about axis perpendicular to the consistency plane of the motorcycle, on condition that the suspension motion. An upright is attached to the front tips of the wishbones by two ball joints, which permit it to turn left and right and also to move up and down. As a result, the steering axis explained by the process line passing through the geometric centre of the ball joints. The control over the steering angle is appeal by the passenger to the handle grip which is linked to the upright through the steering linkage. This is a structure of two levers, linked by

an axis, which can be compact or elongated in order to reach the length between the handle grip and the upright. The front wheel is connected to this upright, and the suspension reaction is supply by a spring-damper unit linked between the lower wishbone and the two wheeler chassis.

1.5 CHASSIS ASSEMBLY

A chassis comprise of an internal framework that supports a fabricated man-made object. It is comparable to an animal's skeleton. An example of a chassis is the under part of a motor cycle, containing of the frame with the wheels and machinery. In the matter of vehicles, the word chassis means the frame plus the conducting gear like engine, transmission, driveshaft, differential, and suspension. A body, which is usually unjustified for integrity of the structure, is construct on the chassis to complete the motor vehicle. While driving and transferring vertical and lateral loads, caused by accelerations, on the chassis through the suspension and the wheels and thus these automotive chassis is tasked with holding all the components. Hence, the chassis is considered as the influential element of the motor vehicle as it cover and holds all the parts and components jointly. Generally it is made of a steel frame, which carry the body and motor of an automotive motor vehicle. Chassis frame forms the backbone of a heavy goods vehicle, its fundamental function is to safely hold the maximum load for all designed operating conditions. It is necessary that the frame should not buckle on uneven road surfaces and that any distortions which may occur should not be transmitted to the body. The frame must therefore be torsion resistant.

A motor cycle has as distinguishing characteristic which are its high stiffness, high power-to-weight ratio, flexural resistance, inertia, low fuel consumption and its nimbleness, so why the lightness of motorcycle plays a crucial role and this is the important reason to make this thesis. Chassis is one of the most important body components of the motor vehicle; the two wheeler be made up of chassis must be powerful, strong and must support itself as well as the other components. It should be able to hold on static load such as holder engine and goes on. In this report, a two wheeler chassis is re-designed in order to make it stronger, safe, lighter and also that being easy to manufacture and maintain further by keeping the manufacturing cost good and small for the customers.

This type of spring has different spring rates. Spring rates also commonly known as spring constant is an important criteria to measure how much resistance is offered by a spring to force (weight) by undergoing deflection.

1.6 KINEMATICS AND DYNAMICS

As the vehicle move over the terrains the vehicle will rise and fall it may leads to damage the parts to overcome this they have divided into two separate areas, these are

1. Suspension dynamics
2. Suspension kinematics

Suspension dynamics depends on the road condition and amount of energy which are applied when motor vehicle passes and take results of it. So that we can chose the type of suspension can used and damper to optimize those values. Suspension kinematics as the vehicle rises and falls where the position changes and the amount of travelled, where should the suspension must be placed at what point must be attached can be seen By considering this we can conduct on different material and properties of it can be changed and assess how good this configuration will provide good control to vehicle, which will provide better contribution than other material and what is wanted from vehicle.

The two most common types of the independent suspension systems are the double Wish-bone and the MacPherson Strut. In a double wishbone suspension system the upper control and lower control arms are linked by a pivot to a motor vehicle chassis and respective outer portions are attached by a steering knuckle. A quarter-car model is commonly used to study the suspension dynamic habits. The two types of suspension geometries produce different responses in real systems due to their inherent geometrical and structural differences.

The All-Terrain Vehicle (ATV) was started in the 1960"s as a farm town motor vehicle in secluded, mountainous areas. During spring thaws and rainy seasons, steep mountainous roads constantly impassable with conventional motor vehicles. It soon became a recreational motor vehicle anyway, providing transportation to areas out of the way by other motorized transport. Royal Enfield CO built and put on sale a powered Quadra cycle in the year 1893 that worked in the likewise, and resembles, a modern quad-bike. ATVs were made in the United States a decade before 3 and

4 wheeled motor vehicles were established by Honda and other Japanese companies. During the 1960s, numerous manufacturers offered related small off-road motor vehicles that were designed to float and were able accomplished of traversing swamps, ponds and streams, as well as dry land.
Grand Prix Motorcycle Racing is a sophisticated motorcycle racing event held on racing circuits. Grand Prix two wheeler are purpose built machines that are basically inaccessible for legally riding on public roads. These big machines meet extensive loads under power-cornering manoeuvres, hence every component of these motorcycles are meticulously designed to improve safety of the vehicle and its performance. This research investigation is specifically focussed on swing arm component of the two wheeler.

Whenever a two wheeler is build, the structure decides the first look of the two wheeler. Naturally two wheeler structure act on appearance, handling and safety of the assembled machine. Certainly two wheeler frames have an impact on not only the appearance of the two wheeler but the handling and safety of the finished machine. Frames are the basic skeleton to which other components are linked. They hold the two wheeler tanks and engine and provide support to the whole motorcycle. Two wheeler frames are usually made from welded aluminium, steel or alloy, carbon-fiber is used in some costly or custom frames. The purpose of a two wheeler frame is to act as a base on which all the different components can be bolted. The engine basically sits inside the frame, the rear swing arm is connected by a pivot bolt and the front forks are linked to the front of the frame. The frame can also help to protect the more sensitive parts of a two wheeler in a crash. A two wheeler frame includes the head tube that holds the front fork and allows it to pivot. Some two wheeler include the engine as a load-bearing, stressed member. Traditionally frames have been steel, but titanium, aluminium, magnesium, and carbon-fibre, along with composites of these materials, have been used. Because of different two wheelers' varying needs of cost, complexity, weight distribution, stiffness, power output and speed, there is no single ideal frame design.

1.7 APPLICATIONS

MotoGP two wheeler are the most advanced racing motor cycle, and the solutions launched by the development teams for these motor cycle are carry out in public racing motor cycle within a few years. The uses of lightweight materials such as aluminium, titanium, magnesium and carbon fibre-reinforced plastics, as well as highly sophisticated suspension and brake units and extreme engine packages, make MotoGP two wheeler the most progressive and fastest track bikes in the world. Fabricating a high-performance MotoGP two wheeler prototype entails extensive engineering prior to skilled riders can evaluate the performance and handling qualities of the physical system. Unfortunately, the brief period between hectic MotoGP racing seasons damper the number of physical tests that can be performed during the design process. Therefore, virtual prototyping can plays a major role in this process, as mentioned in Cossalter and Lot in the year 2002, Evangelou et al in the year 2006, Frezza and Beghi in the year 2003, Hauser and Saccon in the year 2006, Sharp and Limebeer in the year 2001 and Sharp et al in the year 2004.

currently, most of the motor cycle are using a passive hydraulic suspension. One of the important features of hydraulic suspension is uses hydraulic oil as a damper. Whenever a vehicle experiences road unevenness, the stimulation force from road surface is absorbed by the damper. Inside the damper the absorbed energy is converted into heat. The motor vehicle body weight is supported by the mechanical spring linked with the damper. Anyway, there are some disadvantages of hydraulic system. According to Gysen, hydraulic damper contributes to environmental pollution in the problem of hose leaks and ruptures, as well as hydraulic fluids are toxic. So that, the hydraulic systems are considered inefficient due to the required continuously pressurized system. On the opposite side, magnetic suspension systems (MS) not need hydraulic fluid. It comprise two magnets and two springs. One magnet and one spring is linked to the chassis and another spring and magnet is attached to the road wheels. Essentially, faradays law of electromagnetic invention is more important supportive law for that, as we have heard about maglev train one of the best invention today's modern technology and it works on the magnetic levitation and easily support gap between track and railway. This same concept which is using for electromagnet suspension system due to it have a

comparably high force density that can control the motor vehicle body vibration same as in the hydraulic damper.

In this, Suspension system or shock absorber is a device which is drafted to smooth out or damp shock impulse, and scatter kinetic energy. The shock absorbers role is to absorb or dissipate energy. In a motor vehicle, it decreases the effect of traveling over rough ground, leading to better ride quality, and increase in comfort because of substantially will decreases the amplitude of disturbances. When a motor vehicle is traveling on an equal level road and the wheels hit a bump, the spring is compressed rapidly. The compressed spring will strive to return to its normal loaded length and will recover past to its normal height, causing the body to be lifted. The weight of the motor vehicle will press the spring down below its normal loaded height. This, in turn causes the spring to recover again. This bouncing process is sustained over and over, a small less each time, till the up-and-down movement finally stops. If bouncing is permit to go uncontrolled, it will not only cause an uncomfortable ride but will make handling of the motor vehicle very difficult. So, the design of spring is more important for the rider's safety.

A plug-in hybrid electric motorcycle (PHEM) is provide with two or more energy source, basically a gasoline internal combustion engine is combined with an electric motor to propel the PHEM. The electric system in PHEM works at high efficiency, permit diversification of energy resources, zero local emission and work silently. Anyhow, PHEM need more components and complex drive-train mechanism as comprise to conventional internal combustion engine of two wheeler. These cause the design of PHEM complete system packaging is challenging as constraint by limited space in a two wheeler chassis.

CHAPTER 2

LITERATURE SURVEY

K. Sivaramakrishnan, Design And Analysis of Two Wheeler, explains that the chassis frame forms the backbone of a vehicle, its principle function is to safely carry the maximum load for all designed operating conditions.

Kiran Sokande, Optimization of Double Wishbone Suspension System by replacing its Upper Arm by Single Member in ATV, in this report explains on the ATV which refers to All Terrain Vehicle. It can be also called as quadricycle, quad and quad bike. Used by single operators. Wider variety of terrain is designed in such a way that it can handled.

Faisal o. mahroogi, Design and Analysis of Double Wishbone Suspension Systems for Automotive Applications, in this study the Suspension system is an important component in automotive systems. The system's part is to reduce vibrations and road disturbance because the motor vehicle moves on surfaces. This work report the methodology of double wishbone suspension method of analysis and design.

Ciro Moreno Ramírez, Dynamic Analysis of Double Wishbone Front Suspension Systems on Sport Motorcycles, In this paper two alternative front suspension systems and their influence on motorcycle non-linear dynamics are investigated.

Prakash Katdare, Design Optimization of Two Wheeler (Bike) Chassis, in this report explains the chassis frame forms the backbone of a motor vehicle; its basis function is to safely carry the maximum load for all designed in operating conditions.

Vignesh B S, Double Wishbone Suspension System; A Research, in this study the advancements in science and technology, effective designs and newly advanced ways of manufacturing for the need to satisfy the customer expectations and to provide them preferable has let to these developments.

Akshay Bhoraskar, Analysis of the Double Wishbone Front Suspension System, in this report explains the Suspension system in a car connects the chassis to its wheels and it comprises of a system of springs, dampers and linkages.

Syed Hassaan Abdullah, Design of Racing Motorcycle Swingarm with Shape Optimisation, it studies the Grand Prix Motorcycle Racing (MotoGP) refers to the premier class of motorcycle racing events held on racing circuits sanctioned by FIM.

Shijil P, Design and Analysis of Suspension System for an all-terrain Vehicle, In this report it studies the static and dynamic parameter of the suspension system of an ATV by determining and analysing the dynamics of the motor vehicle when move on and off road racetrack.

Shubham Kadu, Study and Design of Advance Suspension System for Two Wheeler, Now a day's various types of suspension are available in market but for optimizing the better and efficient cushioning effect.

Manmohan Singh Thakur, Investigation of Design Parameter of Two Wheeler Frame through Comparative Analysis, The work done by the team was comparison in mechanical differences between square and sectional frame and to take conclusions. They researched about the two wheeler frame using two different materials they are aluminium and steel.

Gaurav Vasantrao Bhunte and Dr. Tushar R. Deshmukh, A Review on Design And Analysis of Two Wheeler Chassis, The authors made an investigation on different analysis techniques of automobile frames. And they determined different analysis in various conditions and developed analysis for those frames and obtained results of those analysis and studied on those frames.

CHAPTER 3

AIM & SCOPE

3.1 AIM

The project aims to optimize the design for existing double wishbone suspension system weight according to the performance of the Moto GP bike. The main idea behind this project is to reduce the weight using the new type of optimization technique called generative design. This new system results in perfect caster and camber control, weight of the suspension system is optimized, wear and tear of the tires are controlled, wheels in contact with the road, better handling, vertical and lateral load can be tackled. In the result there must be improvement in the suspension system performance. Optimizing the existing double wishbone suspension used in the tourist cruiser bikes according to the racing bikes performance.

3.2 SCOPE OF THE PROJECT

- To check the structure, parts of the existing double wishbone suspension used in the tourist cruiser bike.
- To calculate the parameters for the race bike. (Swing arm reaction force, caster angle, shock absorber angle, Fork angle, vibration, wheelbase, steering axis).
- To create the design for suspension parts and create the 3D model.
- To analyze and optimize the new design of double wishbone suspension.
- To simulate the new optimized double wishbone suspension design.

CHAPTER 4

MATERIAL SELECTION AND METHODOLOGY WITH DESIGN

4.1 MATERIAL SELECTION

4.1.1 Aluminum AlSi10Mg:

TABLE 4.1: Aluminum AlSi10Mg Mechanical Properties

Mechanical Properties	Aluminium AlSi10Mg
Tensile Strength, MPa	460 ± 20
Yield Strength, MPa	260 ± 20
Modulus of Elasticity, GPa	75 ± 10

AlSi10Mg is a material where it is used in components manufactured using additive manufacturing. It is a typical cast alloy and it has good casting properties, this material is generally for complicated geometry model. Where its properties are good in strength, hardness and dynamic properties. Thus it is used for components subjected to high impacts or loads. Aluminium AlSi10Mg have different applications because it has good thermal properties and low weight. This material has many more special qualities like machining, welding, spark-eroding, polishing and coating. Traditional cast parts using this type of aluminium alloy are regularly heat treated for improvement in mechanical properties. The laser – sintering process is used for manufacturing the components using AlSi10Mg because this material has the properties like rapid melting and re- solidification. These properties help in metallurgy, mechanical properties and heat treatment for casted parts. The relieving cycle for this materials is 2 hours at 300 °C (572 °F). Layer wise additive manufacturing method is used for manufacturing the parts using this material.

4.1.2 Stainless steel:

TABLE 4.2: Stainless steel Mechanical Properties

Mechanical Properties	Type 316L Stainless Steel
Yield Point, MPa	332
Tensile strength, MPa	673
Modulus of Elasticity, GPa	165
Strength at break, MPa	586
Elongation at break, mm	35.5

Stainless steel has a composition of iron and chromium which has greater than or equal to 10.5% and carbon which is lesser than or equal to 1.2% and some other cases nickel and other metals are used. The manufacturing of stainless steel has numbers of processes firstly the raw material have been melted and processed to excess carbon removal, tuning and stirring, forming the metal, heat treatment or annealing, cutting and shaping of material and final application of surface finishes.

The types of stainless steel are austenitic, ferritic, duplex and martensitic. Stainless steel possesses good corrosion resistance and 100% recyclable so the material wastage and the environmental pollution can be reduced significantly. It is widely used in many fields, it is used to produce household products like kitchen sinks, cutlery, cookware also used in medical application and equipment like surgical implants, haemostats temporary crowns. It is also used in architecture like bridges, sculpture, moments and airport roofs. It is also used to produce automobile parts, aircraft applications and rail cars.

4.2 METHODOLOGY:

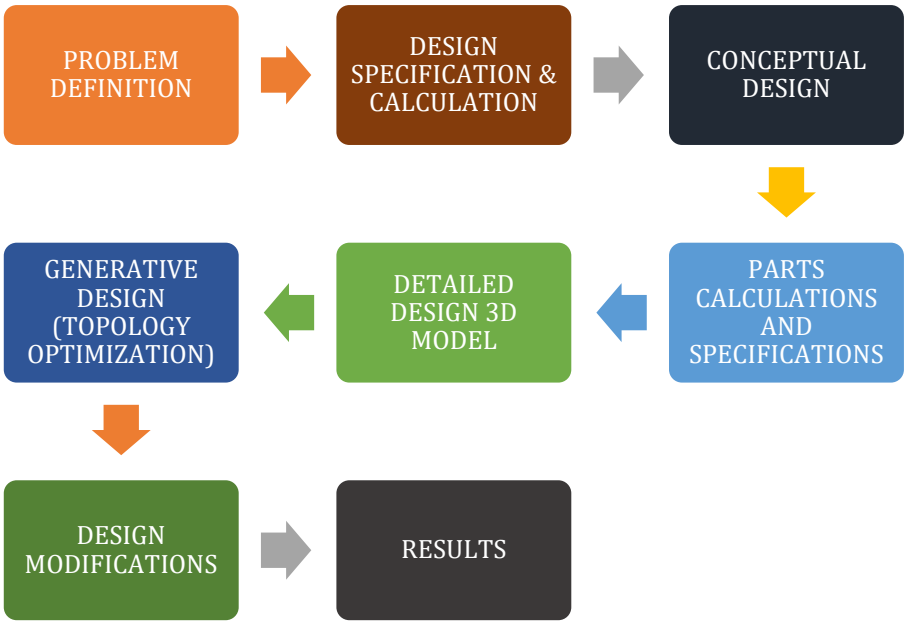

Figure. 1 Methodology: summarizes the sequence of steps followed in the form of a flow chart.

4.2.1 DESIGN SPECIFICATION & DESIGN CALCULATION:

4.2.1.1 Basic Consideration for Suspension System (Parameters):

TABLE 4.3: Basic Consideration for Suspension System (Parameters) Boundary Conditions of Generative Design for Fork

Topics	Values
Swing Arm Reaction Force(Mean Value)	1,207.0666666667 N
Caster Angle	25 Deg
Shock Absorber Angle	63.7 Deg
Fork Angle	24 Deg
Vibration	4.51 Hz
Wheelbase	1450 Mm
Steering Axis	45 Deg
Cg Hight	650 Mm
Tyre Radius	300 Mm
Slant Angle	1 Deg
Kerb Weight	157 Kg
Existing Suspension System Weight	25 Kg
Fork Compression	50 MM

4.2.1.2 Multiaxial Load Conditions:

Weight distributed on rear axle is about 60%

MAXIMUM acceleration of bike can be 7m/s

Weight of motorcycle = 190kg

Weight of passenger = 80+80 = 160

Longitudinal force = m x a = (190+160) x 7=1750N

I. **ON CORNERING**

The load variation can depend on magnitude lean angle and vehicle speed during cornering.

Let us take the maximum inclination angle about **40 degree at 50km/hr.**

At stationary or straight-line riding condition the swing arm experiences 50/50 on its arms. While cornering we assume that about 20% of its weight shifts towards the inner side

So totally we acquire **70% weight on the inner side** and **30% on the outer side**

Maximum force = (**70**/100)*m/g=0.7*350*9.81 = 2,403.45N

Minimum force = (**30**/100)*m/g 0.3*350*9.81 = 1,030.05N

II. **Horizontal components (lateral imbalance force)**

Force on inner horizontal = 2403.45*cos40 = 1841.149N

Force on outer horizontal = 1030.05*cos40 = 789.06N

III. **Vertical component (lateral torsion force)**

Force on inner vertical = 2403.45*sin40 = 1544.90N

Force on outer vertical = 1030.05*sin40 = 662.10N

IV. **Braking Condition**

Maximum deceleration of motorcycle is **-7m/s** (at 50 to 0 in 2sec)

Maximum force = m*a = 350*7 = 2450N

4.2.1.3 Design calculations:

- Kerb weight = 157 kg
- Maximum force = 2403.45 N
- Minimum force = 1030.05 N
- Design load (w) = Total load x Factor of safety
  = 1030.05 x 3 = 3090.15 N
- Allowable shear stress (s) = 470 MPa
- Module of rigidity (G) = 7.2 x 10^4 MPa
- Spring index (c) = 7
- Deflection (s) = 90 mm
- Spring constant k = w/δ = 3090.15/90 = 34.335

Kw = (4c – 1)/(4c – 4) + 0.615/c

Kw = 1.2128

S = Kw*(8 w*c/ 3.14 x d)

470 = 1.2128 * (8 x 3090.15 x 7/3.14 x d)

d = 13.7 mm

D = c x d

= 95.9 mm

Active number of turns

K = (Gd^4)/(8xD^3xN)x17.5

34.335 = (72000 x 13.7^4)/(8 x 95.9^3 x N) x 17.5

N = 14

Total number of turns

N = 12 + 2 = 14

Solid length,

Ls = N x d = 14 x 13.7

Ls = 191.8

Free length of spring,

Lf = solid length + compression + clearance

Lf = 191.8 + 90 + (90 x 0.11) = 291.7 mm

Pitch of spring,

P = 4/N-1 = 22.44 mm

Outer diagram of spring

D0 = D + d = 109.6 mm

Inner diagram of spring

Di = D – d = 82.2 mm

4.2.2 3D CAD MODEL:

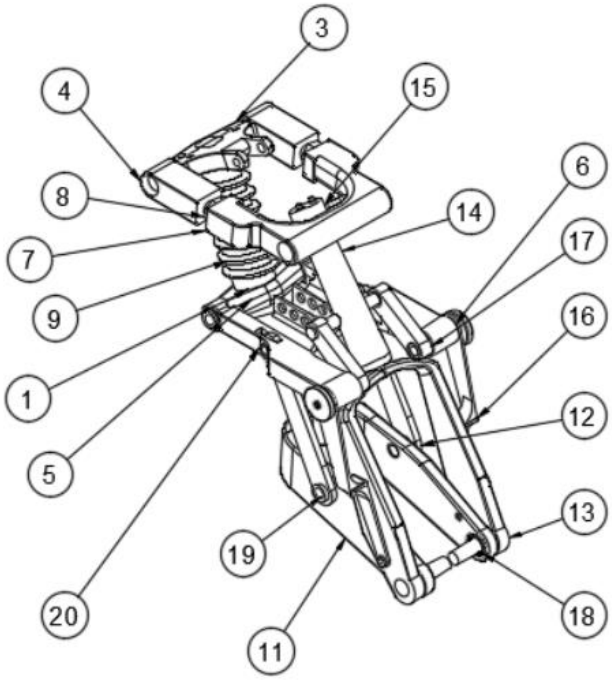

Figure 4.1: Double Wishbone Suspension system assembly

1. SHAFT
2. HEAD BEARINGS
3. UPPER YOKE
4. UPPER LINK
5. BOTTOM YOKE
6. LOWER LINK
7. UPPER LINK
8. ADJUSTMENT ROD (1)
9. SPRING
10. ADJUSTMENT ROD (2)
11. WISHBONE

12. FORK ARM
13. FORK
14. FORK ROD
15. CAP
16. FORK LINK
17. CONNECTOR
18. BEARING
19. WASHER
20. SPACERS

The springs and dampers coefficients has been calculated according to the real time bike suspension system. The fork assembly in the design has a complex geometry and consists of various geometries for assembly of springs, bearings, dampers etc. The cruiser bike is assembled with double wishbone suspension system and is integrated with a single spring damping unit which is linked with Upper and Bottom Yoke. Since it has two links, two yokes which are assembled with fork and spring so called double wishbone suspension system. This suspension system assembly is described brief in the further sections and its optimization process.

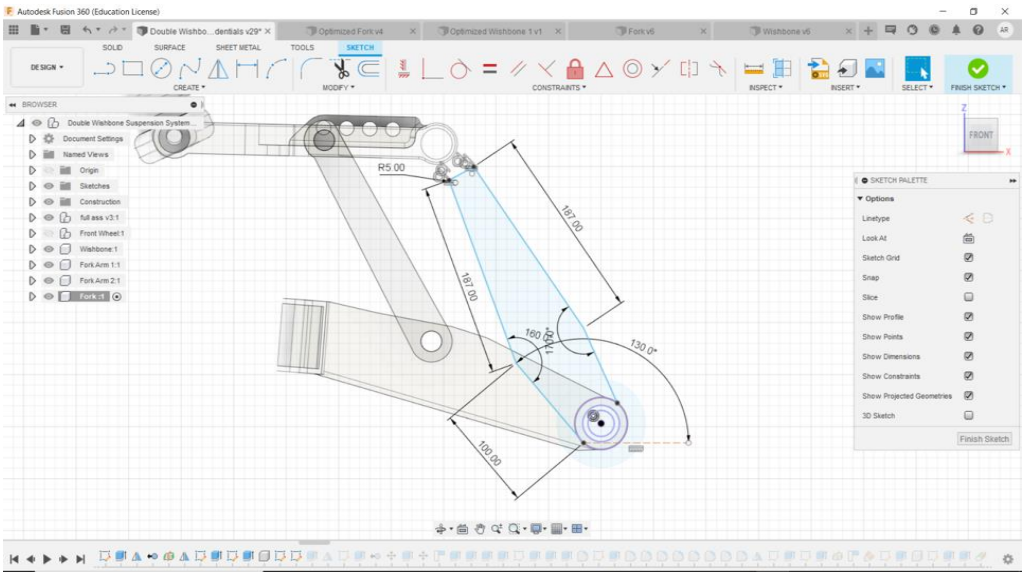

Figure 4.2: Fork 2D Sketch

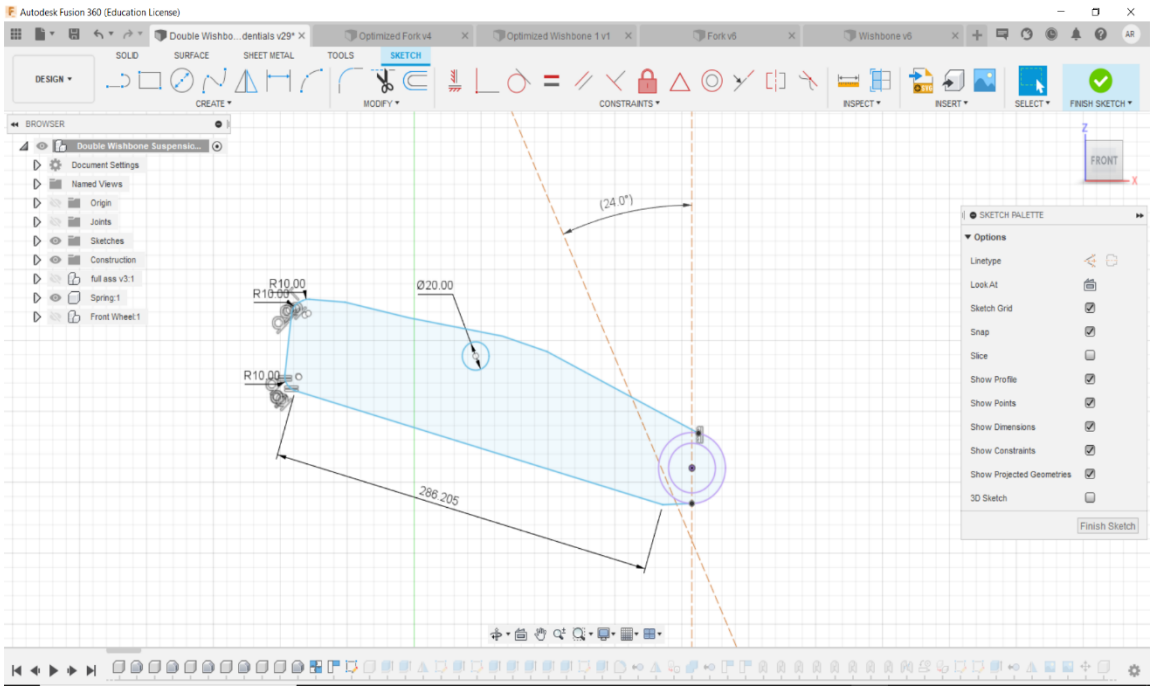

Figure 4.3: Wishbone 2D Sketch

The double wishbone suspension has a type of fork where the internal tube is placed in the lower position and assembled to the wheel. This type of fork is used in the cruiser bikes in recent times.

This suspension system unit is simulated according to the racing scenario, all properties of the suspension fork had identified and integrated into the simulation model. The spring inside the fork has a 3090.15 N force acting on it and a deflection of 90 mm. The deflection does not change spring constant but during simulation the effect of the spring is shown in the Figure 4.5.

Figure 4.4: Load on Suspension fork

Figure 4.5: Load acting on mountings of suspension system

The deflection is used to adjust the travel result from the steering or control unit during certain load. The load acting on the spring opposite to the extension of the fork which has load of 2403.45 N and a travel deflection of 40 mm. This spring has a deflection of first 40 mm of the stroke which is determined corresponding to the full length of the spring. The maximum travel length of the fork spring is 150mm. The

function of spring here is used for free joint motion between the upper and lower arm links.

In compression phase, the fork housing contains volume of air which helps as a dual function.

- Here air acts as a spring that is distinguished as a high progressive rate.
- Some amount of pressure is created inside the fork, which doesn't allow any cavitation problems and any formation of foam from the oil to air contact.

The progressive air spring effect is integrated into the model using a function that was incorporated to brief the behaviour of the spring in the fork housing. This operation included the gas properties of pressure inside the fork housing, which is depended on the front wheel. This pressurized gas creates a force that acts to extend the fork spring length. The force becomes zero when it reaches full extension. This force formed by the gas pressure is exhibited as a function of the stroke length. Cavitation match to create gas bubbles in the oil in the damper piston at a high damping velocity. Cavitation is formed by a large pressure on the fork's orifice. Those effects are negligible in this model because a suitable configured fork ensure that cavitation occurs rarely. This advanced dampers maintain the oil under required high pressure where gas doesn't form.

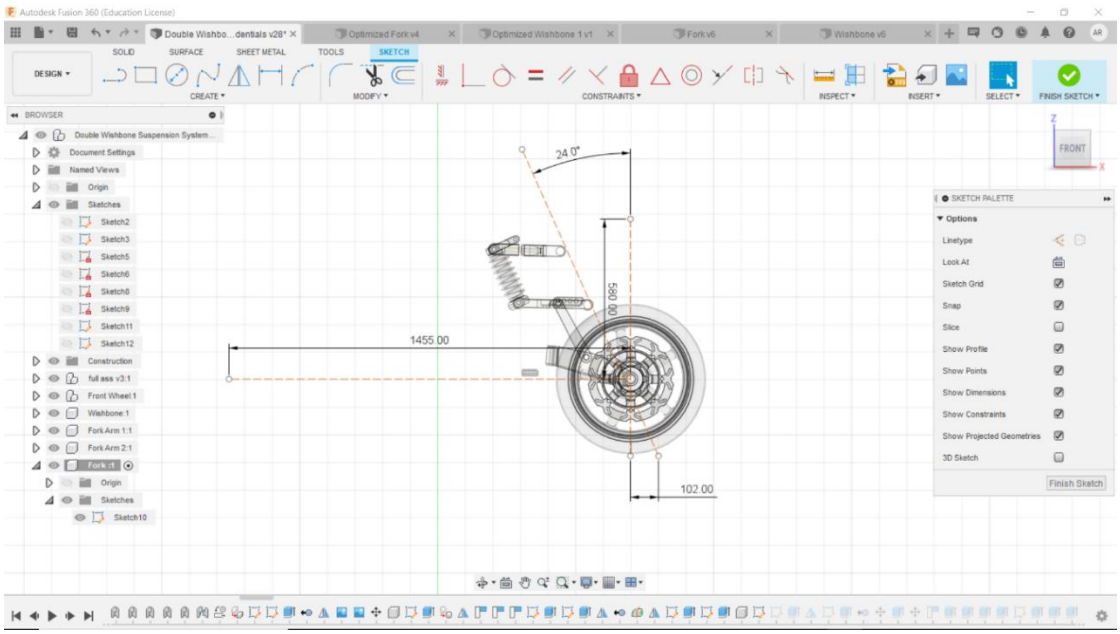

Figure 4.6: Rack angle and Wheel Base Geometry placement

The load acting on the model is static on the front wheel is 2403.45 N for a fork angle of 24°, which is calculated to a load of 1030.05 N in the direction of the fork. This load resulted in an initial fork compression is about 60 mm, which is measured relative to the fully extended length of the fork. The suspension characteristics of fork leg and wishbone modelled by single spring and a joint that acts as a spring and a damper. The spring in the fork model and pressure simulated inside the fork.

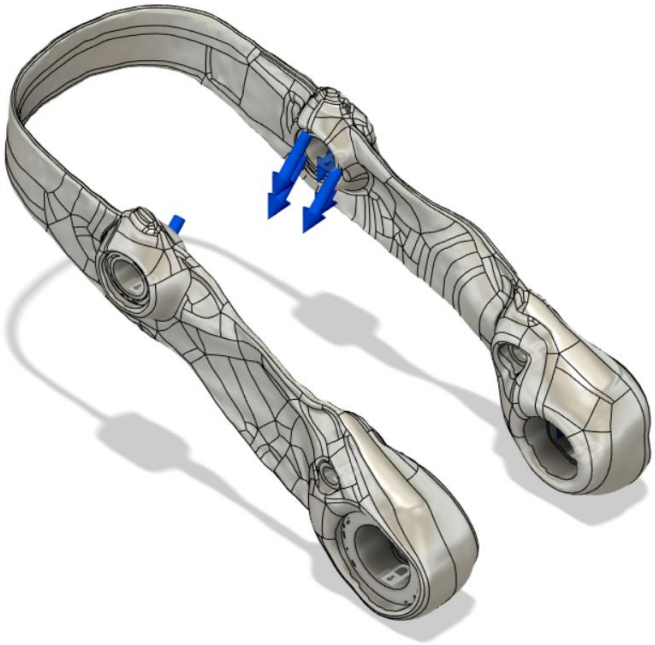

Figure 4.7: Loads acting on wishbone

This spring works during extension of fork and this assembly has inbuilt with stiff spring assembly that acts as dampers to provide the wheel stroke to limit up-to 150 mm. The fork has a extension from 50 mm and compressed to 90 mm till its original position. The spring were assembled perpendicular to the each fork leg housing and connected with wishbone.

4.2.3 GENERATIVE DESIGN:

Generation design is a repetitive design process which is a program generates a selective number of outcomes according to the constrains given. Where designs can be selected according to the material, factor of safety, stress and strain, weight, volume, deformation and manufacturing method etc. In this project the mass optimization are performed for parts of double wishbone suspension system are suspension fork and wishbone.

Setting up the Part:

In generative design process many factors are depended according design part to run this process. First the part setting must be with a predefined model. Then place the mountings in the model so that the loads are been placed. The existing geometry features helps to define the load points.

The generative design process consists of 3 steps:

- Part modelling
- To create obstacle and preserve geometry model
- Model configuration

- **Part modelling**

  The starting step in the generative design process is predefine a model or to create a new model with basic shapes like holes, load region, mountings etc. These models can be done using basic 3D model techniques and keep them as a single body and assembly model can make many complications to solve the design and any kind of peripheral parts can be removed.

- **To create obstacle and preserve geometry model**

  After completion of modelling, new bodies has to be created for representing of two sets of data called obstacles and preserve regions and these help to define the generative design areas. The areas where generative design must not be allowed to enter is called obstacles. And preserve regions are the regions where the shape and size has be to conserve which is defined from the model.

- **Configure the Model**

  Before starting the generative design process two more important parameters has to be taken to consideration. First is to setting the material type to the body and set the units. The desired material will be given as a input before that a desired weight reduction has to be given for the exact weight has to be also noted. Different bodies is set with colors in the next step. There are three colors used here for different areas in the generative design process, red is set for obstacle geometry, yellow is set for starting shape and green for preserve geometry.

- **Analysis Steps**

For running an analysis there are five basic steps as following:

a. **To identify the problem and collect data**

The first step requires a brainstorm to solve the problem according to the assembly clearance and kind of manufacturing and other real time parameters are considered. The dynamic parts like load variation, moving parts etc. are also considered. Finally mass reduction, performance, aesthetics and other design changes also to be considered.

b. **Create Preserve Geometry**

Start the process with existing 3D model whether it can be single component or assembly. Then interface geometry has to be created like bolts, flanges etc. for the generative design process as new bodies. This region always depends according to the usage of the part or assembly.

c. **Create Obstacle Geometry**

Now again the base model has to be edited again because an obstacle geometry has to be created. These geometry includes bolts, screws, nuts, pins etc. and other impenetrable geometry bodies are considered as an obstacle geometry. But thing is clearances has to be noted. And finally the obstacles has to cover the problem statement.

d. **Setting up Boundary Conditions**

The parameter or constraints like loads, manufacturing method, and material, factor of safety, stiffness, mass target and buckling frequency are specified. Proper or calculated load cases are set on the preserves according to the motions that are involved. In this case the fork and wishbone are experienced with braking, cornering, horizontal and vertical forces.

e. **Run the Study**

Final step for this analysis study is to run the process. The early results has to be ensured, to get the expected result in different iterations. During this process if any parameter input is missed can be identified.

The analysis is reviewed to check the data given by the software is accurate and correct which is defined to get the generative design. The results are previewed

with various options according to the number of materials given as input and other parameters given to the system.

As the results are completed, the tools are used for determining the solutions which are worthful for the problem statement. The output result sorted using the scatter plots, comparing the results according to the model stress analysis by loads given during to pre-processing of the generative design. The evaluation of results depends according to the following:

- **Compare Results with Initial Expectations**

Once the results are generated has to be validated according to the desired shape or design. The results are refined in various iterations by the starting shape. The manoeuvre result make an appreciable starting shape for the secondary study. For few results validated by using additional FEA package.

- **Design modification**

  once the satisfied result is generated, the design is exported as a B-rep/T-rep and it is placed back as the base model. Again brainstorm about the steps must take next according to the CAM, CAE simulated validation and additive manufacturing. Required features must be added like T- spline modifications are considered to get desired design requirements. The final model is taken to CAM, 3D printing software as STL file and CAE then assemble the part in the original design. This process is iteratively continuous work to refine the designs in further. This generative design process is a complicated set of tools which help to develop the parts to give their performance at its best, it is as good as the more information given to solve the model easily and to get accurate results.

4.2.3.1 GENERATIVE DESIGN FOR FORK

Suspension fork in double wishbone suspension is assembled with hydraulic shock absorbers that is mounted to the frame to front wheel. Those shock absorbers help wheel to travel with fork. This active movement absorbs the jerks and other impacts of tough terrain, by this a comfortable ride will be given to the rider.

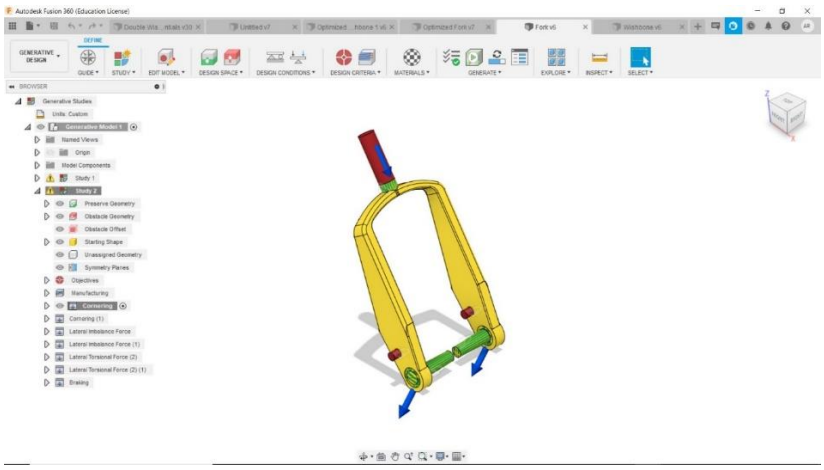

Figure 4.8: Generative design setup for Suspension Fork

- **To identify the problem and collect data**

  The first step requires a brainstorm to solve this problem according to the assembly clearance and kind of manufacturing and other real time parameters are considered. The dynamic parts like load variation, moving parts etc. are also considered. Finally mass reduction, performance, aesthetics and other design changes also to be considered.

- **Create Preserve Geometry**

  Start the process with existing 3D model whether it can be single component or assembly. Then interface geometry has to be created like bolts, flanges etc. for the generative design process as new bodies. This region always depends according to the usage of the part or assembly.

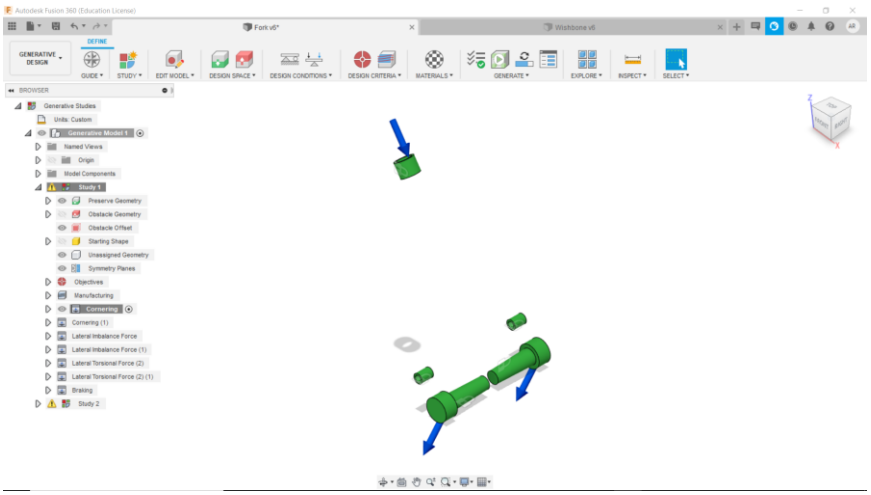

Figure 4.9: Preserves bodies for Suspension Fork

- **Create Obstacle Geometry**

  Now again the base model has to be edited again because an obstacle geometry has to be created. These geometry includes bolts, screws, nuts, pins etc. and other impenetrable geometry bodies are considered as an obstacle geometry. But thing is clearances has to be noted. And finally the obstacles has to cover the problem statement.

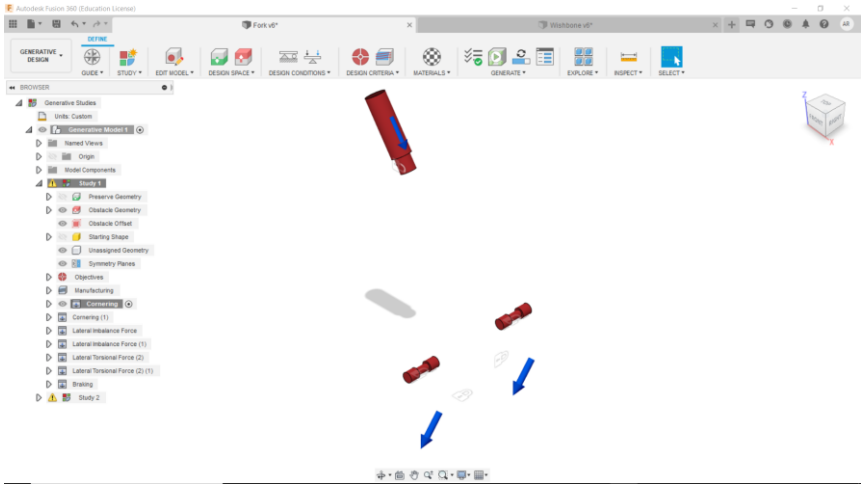

Figure 4.10: Obstacle bodies for Suspension Fork

- **Setting up Boundary Conditions**

  The parameter or constraints like loads, manufacturing method, and material, factor of safety, stiffness, mass target and buckling frequency are specified. Proper or calculated load cases are set on the preserves according to the motions that are involved. In this case the fork and wishbone are experienced with braking, cornering, horizontal and vertical forces.

Table 4.4: Boundary Conditions of Generative Design for Fork

BOUNDARY CONDITIONS	VALUES
MATERIALS	ALUMNIUM AlSi10Mg
FACTOR OF SAFETY	3
MANUFACTURING METHOD	ADDITIVE MANUFACTURING
FORCE (N) These forces act at different scenarios and has different load cases on the fork	3434 N 2403 N 1030 N 1841 N 789.1 N 1545 N 662.1 N
SHOCK LOAD	4120N

- **Run the Study**

  Final step for this analysis study is to run the process. The early results has to be ensured, to get the expected result in different iterations. During this process if any parameter input is missed can be identified. The analysis is reviewed to check the data given by the software is accurate and correct which is defined to get the generative design. The results are previewed with various options according to the number of materials given as input and other parameters given to the system.

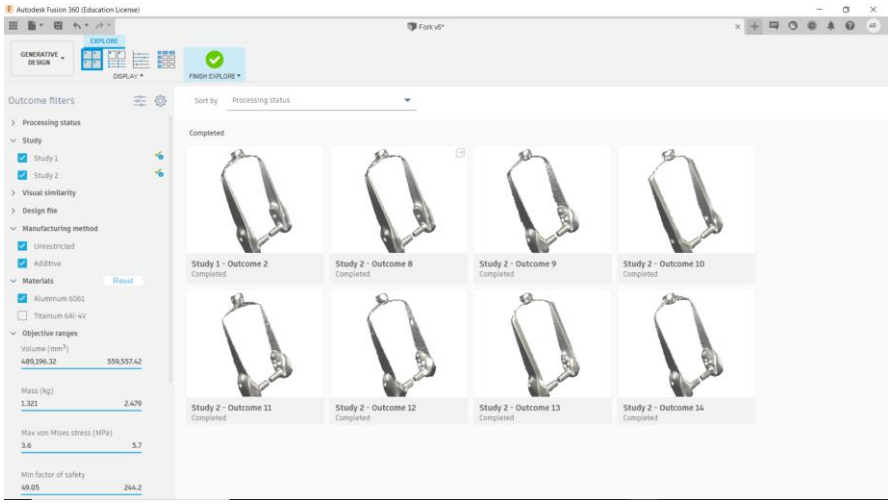

Figure 4.11: Selection of design outcome according to the shape, material, weight, cost of manufacturing, FOS, Stress etc

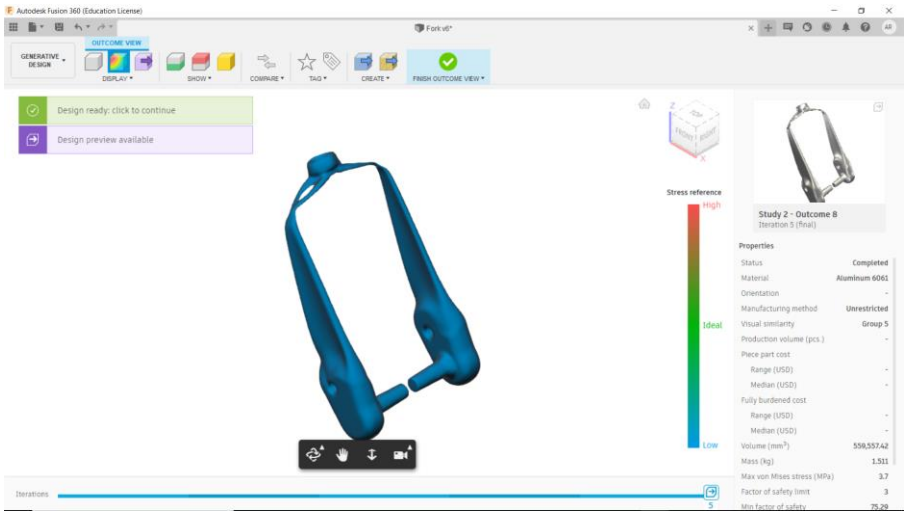

Figure 4.12: Selection of design outcome after solving various iterations and according to the shape and stresses shown in the outcome view

4.2.3.2 Generative design for Wishbone

The wishbone is an important part for this suspension system. Its function is to control wheel travel and transmits the load on the wheel. When the bike is in travel, this wishbone is undergoes to complicated loads according to time and other various boundary condition. The functions of wishbone are to absorb the road shocks and provide the rider a comfortable riding experience. It helps bike to

be in stable conditions during many scenarios like braking, turning, rolling and pitching. Also minimizes the effects of stresses from the shock loads on the bike mechanical parts and gives a cushioning effect. It also helps the bike place in proper height and structure and also maintains the rack angle and doesn't allow any torque, braking, force, swing arm reaction force, so that during the rider travel in uneven road the body is perfectly balanced and sTABLE

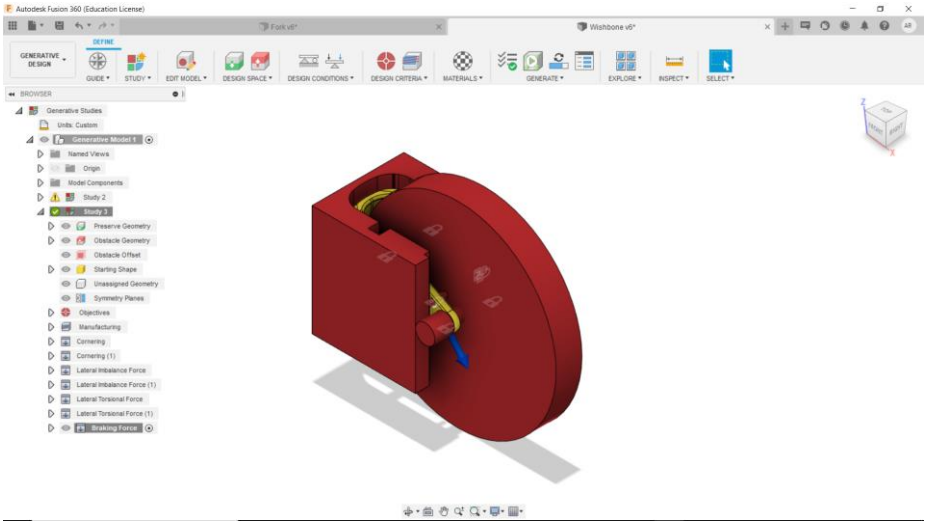

Figure 4.13: Generative Design Setup for Wishbone

- **To identify the problem and collect data**

  The first step requires a brainstorm to solve this problem according to the assembly clearance and kind of manufacturing and other real time parameters are considered. The dynamic parts like load variation, moving parts etc. are also considered. Finally mass reduction, performance, aesthetics and other design changes also to be considered.

- **Create Preserve Geometry**

  Start the process with existing 3D model whether it can be single component or assembly. Then interface geometry has to be created like bolts, flanges etc. for the generative design process as new bodies. This region always depends according to the usage of the part or assembly.

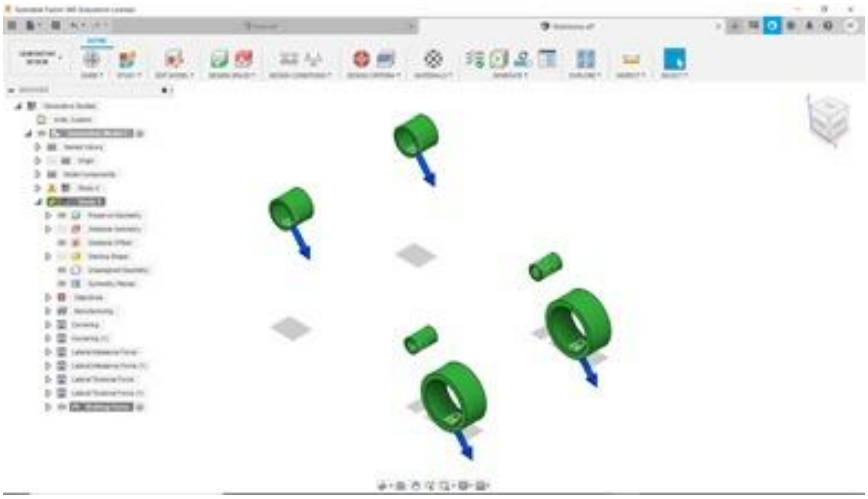

Figure 4.14: Preserve bodies for Wishbone

- **Create Obstacle Geometry**

  Now again the base model has to be edited again because an obstacle geometry has to be created. These geometry includes bolts, screws, nuts etc. and other impenetrable geometry bodies are considered as an obstacle geometry. And finally the obstacles has to cover the problem statement.

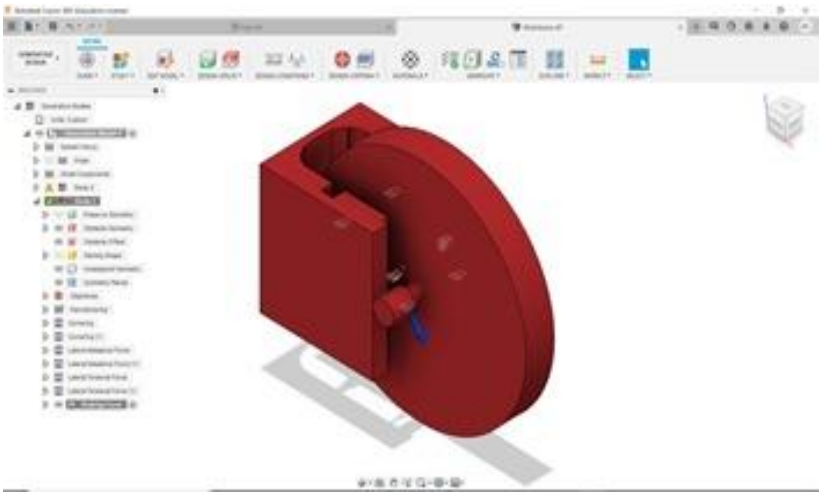

Figure 4.15: Obstacle bodies for wishbone

- **Setting up Boundary Conditions**

  The parameter or constraints like loads, manufacturing method, and material, factor of safety, stiffness, mass target and buckling frequency are specified. Proper or calculated load cases are set on the preserves according to the motions that are involved. In this case the fork and wishbone are experienced with braking, cornering, horizontal and vertical forces.

TABLE 4.5: Boundary Conditions of Generative Design for Wishbone

BOUNDARY CONDITIONS	VALUES
MATERIALS	ALUMNIUM AlSi10Mg
FACTOR OF SAFETY	3
MANUFACTURING METHOD	ADDITIVE MANUFACTURING
FORCE (N) These forces act at different scenarios and has different load cases on the wishbone	3434 N 2403 N 1030 N 1841 N 789.1 N 1545 N 662.1 N
SHOCK LOAD	4120N

- **Run the Study**

  Final step for this analysis study is to run the process. The early results has to be ensured, to get the expected result in different iterations. During this process if any parameter input is missed can be identified. The analysis is reviewed to check the data given by the software is accurate and correct which is defined to get the generative design. The results are previewed with various options according to the number of materials given as input and other parameters given to the system.

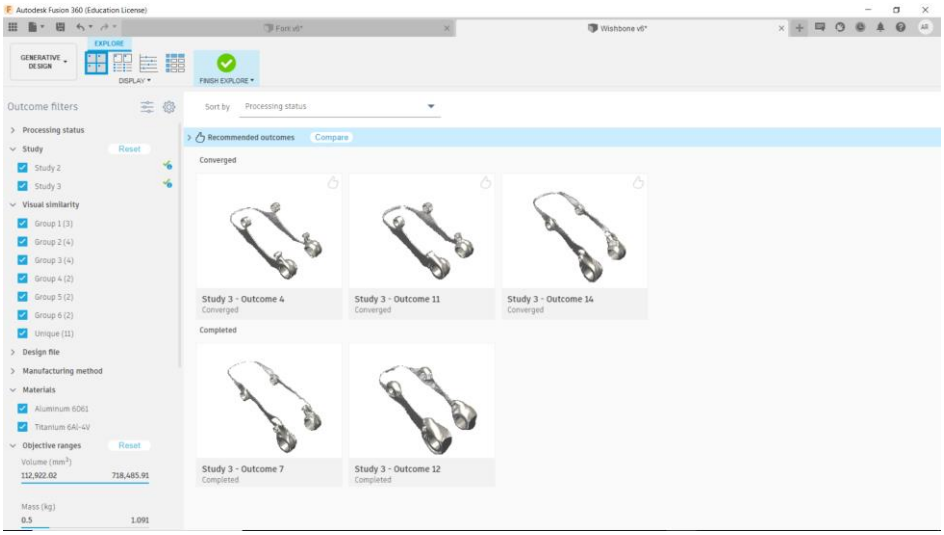

Figure 4.16: Selection of design outcome according to the shape, material, weight, cost of manufacturing, FOS, Stress etc.

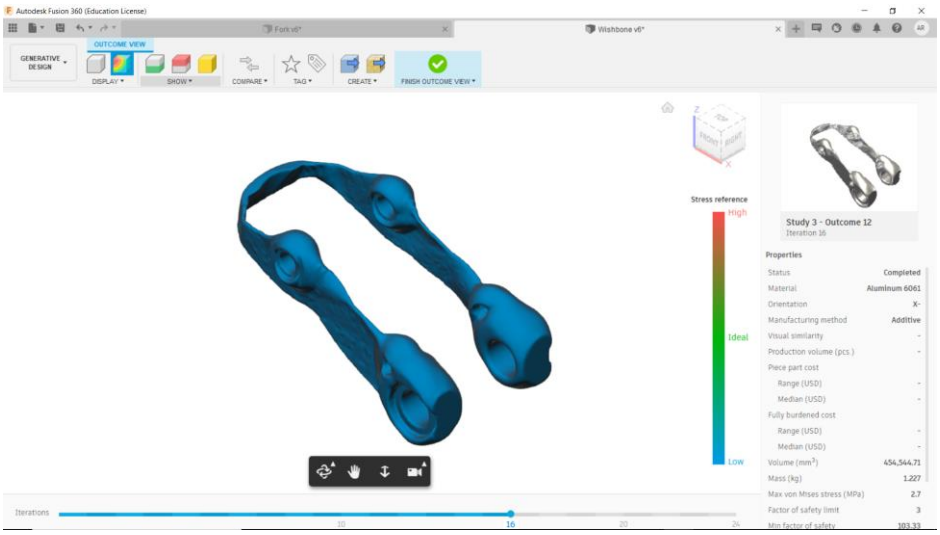

Figure 4.17: Selection of design outcome after solving various iterations and according to the shape and stresses shown in the outcome view

CHAPTER 5

RESULT AND DISCUSSIONS

5.1 FINITE ELEMENT ANALYSIS

FEA or finite element analysis is a commonly used method for physical problems using numerical techniques, since suspension system is a complex mechanism it is difficult to solve in a theoretical manner. It is used to solve very difficult real-time propagation or fluid flow. Any continuous object in space has multiple degrees of freedom which makes it difficult to solve in conventional ways, the outcome of the solution is very far from the real-time study.

In FEA, the object is needed to be solved can be splatted the surface body into a discrete number of elements for which the solution can be calculated and solved. The element can be spread its boundary condition throughout the whole domine. FEA adds substitutional value to the product design process.it provides significant insight and design that helps to create better products. Some of the specific benefits and outcome of using FEA include the following:

- It can predict the performance of the plan to be used
- It can predict the failure accusation for the case study
- It is used to evaluate and correct observed failures
- It is used to evaluate the design and improve the performance and safety
- Improvises the cost for manufacturing
- Improvises the weight of the known design with optimal performance
- To develop new innovative concepts with high savings on development
- Gains insight into the design concepts or directions
- Modeling decisions a be driven through the goals

➢ **Meshing**

It is an important process to obtain results accurately in finite element analysis. The smaller the mesh size the more resolution of the design solutions as samples across the physical domains for higher accuracy the mesh size to be very fine result in an increase in time taken for processing.

A finite element mesh includes both nodes and elements

- Nodes are points in 3d space
- Elements are area or volume defined by nodes

The geometry provided by the element poses the template for meshing. Based on the area defined in a domain the element types are wildly classified into four types they are

- Shell elements
- Solid elements
- Line elements

The shell elements are usually used in modal structures in which one dimension, the thickness, is significantly smaller than the other dimensions. Conventional shell elements use the condition to discretize the body of 3d model by defining the geometry at a reference surface of the model. Conventional shell elements have both displacement and rotational degrees of freedom but in the continuum, shell element discretizes an entire three-dimensional body only. The thickness is decided from the element nodal geometry. while the Continuum shell elements have transposition degrees of freedom only in the thickness property of the material. From a modelling point of view, continuum shell elements appear as if 3d continuous solid, but their kinetic and constitutive behaviour is analogous to standard shell elements.

In Solid elements, the entire geometry is been described by the cad model geometry. The tetrahedrons are the element type for three-dimensional, solid meshes. There are two types of tetrahedrons;

- Linear tetrahedrons
- Parabolic tetrahedrons

In linear tetrahedral element is defined by four corner nodes connected by straight edges. In parabolic tetrahedral element is described by four corner nodes, six mid-side nodes, four corner nodes, and six edges. For the same mesh density for a number of elements, parabolic tetrahedral elements possess improved results than linear elements because they denote curved boundaries more accurately and they make better mathematical approximations. The

parabolic tetrahedral elements require more computational resources than linear elements.

5.1.1 FEA ANALYSIS BEFORE OPTIMIZATION FOR FORK:

Mesh Details:

TABLE 5.1: Mesh Details of FEA Analysis before Optimization for Fork

Average Element Size (% of model size)		
Average Element Size (absolute value)		10 mm
Element Order		Parabolic
Max. Turn Angle on Curves (Deg.)		60
Max. Adjacent Mesh Size Ratio		1.5
Max. Aspect Ratio		10
Minimum Element Size (% of average size)		20
Type	**Nodes**	**Elements**
Solids	49368	29614

5.1.1.1 STATIC STESS STUDY:

Scenario: Cornering Towards Right

Results Summary

TABLE 5.2: Results for FEA Analysis before Optimization for Fork Static Stress Study Cornering Towards Right

Name	**Minimum**	**Maximum**
Safety Factor		
Safety Factor (Per Body)	2.252	15
Stress		
Von Mises	4.182E-05 MPa	122.1 MPa
Displacement		
Total	0 mm	0.003199 mm
Reaction Force		
Total	0 N	123.4 N
Strain		
Equivalent	6.621E-10	0.001929
Contact Pressure		
Total	0 MPa	6.227 MPa

Stress

Figure 5.1: Stress for before optimization of fork during cornering towards right

Displacement

Figure 5.2: Displacement for before optimization of fork during cornering towards right

Scenario: Cornering Towards Left

Results Summary

TABLE 5.3: Results for FEA Analysis before Optimization for Fork Static Stress Study Cornering Towards Left

Name	**Minimum**	**Maximum**
Safety Factor		
Safety Factor (Per Body)	2.252	15
Stress		
Von Mises	4.182E-05 MPa	122.1 MPa
Displacement		
Total	0 mm	0.003199 mm
Reaction Force		
Total	0 N	123.4 N
Strain		
Equivalent	6.621E-10	0.001929

Stress

Figure 5.3: Stress for before optimization of fork during cornering towards left

Displacement

Figure 5.4: Displacement for before optimization of fork during cornering towards left

Scenario: Lateral Imbalance towards Right

Result Summary

TABLE 5.4: Results for FEA Analysis before Optimization for Fork Static Stress Study Lateral Imbalance towards Right

Name	**Minimum**	**Maximum**
Safety Factor		
Safety Factor (Per Body)	2.252	15
Stress		
Von Mises	4.182E-05 MPa	122.1 MPa
Displacement		
Total	0 mm	0.003199 mm
Reaction Force		
Total	0 N	123.4 N

Stress

Figure 5.5: Stress for before optimization of fork during Lateral Imbalance towards Right

Displacement

Figure 5.6: Displacement for before optimization of fork during Lateral Imbalance towards Right

Scenario: Lateral torsional Towards Right

Result Summary

TABLE 5.5: Results for FEA Analysis before Optimization for Fork Static Stress Study Lateral torsional Towards Right

Name	Minimum	Maximum
Safety Factor		
Safety Factor (Per Body)	2.252	15
Stress		
Von Mises	4.182E-05 MPa	122.1 MPa
Displacement		
Total	0 mm	0.003199 mm
Reaction Force		
Total	0 N	123.4 N
Strain		
Equivalent	6.621E-10	0.001929
Contact Pressure		
Total	0 MPa	6.227 MPa

Stress

Figure.5.7: Stress for before optimization of fork during Lateral torsional towards Right

Displacement

Figure 5.8: Displacement for before optimization of fork during Lateral torsional Towards Right

Scenario: Lateral Imbalance towards Left

Result Summary

Table 5.6: Results for FEA Analysis before Optimization for Fork Static Stress Study Lateral Imbalance towards Left

Name	**Minimum**	**Maximum**
Safety Factor		
Safety Factor (Per Body)	2.252	15
Stress		
Von Mises	4.182E-05 MPa	122.1 MPa
Displacement		
Total	0 mm	0.003199 mm
Reaction Force		
Total	0 N	123.4 N
Strain		
Equivalent	6.621E-10	0.001929

Stress

Figure 5.9: Stress for before optimization of fork during Lateral Imbalance towards Left

Displacement

Figure 5.10: Displacement for before optimization of fork during Lateral Imbalance towards Left

Scenario: Lateral torsional Towards Left

Result Summary

TABLE 5.7: Results for FEA Analysis before Optimization for Fork Static Stress Study Lateral torsional towards Left

Name	Minimum	Maximum
Safety Factor		
Safety Factor (Per Body)	2.252	15
Stress		
Von Mises	4.182E-05 MPa	122.1 MPa
Displacement		
Total	0 mm	0.003199 mm
Reaction Force		

Total	0 N	123.4 N
Strain		
Equivalent	6.621E-10	0.001929
Contact Pressure		
Total	0 MPa	6.227 MPa

Stress

Figure 5.11: Stress for before optimization of fork during Lateral torsional towards Left

Displacement

Figure 5.12: Displacement for before optimization of fork during Lateral torsional Towards Left

Scenario: Braking

Result Summary

TABLE 5.8: Results for FEA Analysis before Optimization for Fork Static Stress Study Braking

Name	**Minimum**	**Maximum**
Safety Factor		
Safety Factor (Per Body)	2.252	15
Stress		

Von Mises	4.182E-05 MPa	122.1 MPa
Displacement		
Total	0 mm	0.003199 mm
Reaction Force		
Total	0 N	123.4 N
Strain		
Equivalent	6.621E-10	0.001929
Contact Pressure		
Total	0 MPa	6.227 MPa

Stress

Figure 5.13: Stress for before optimization of fork during Braking

Displacement

Figure 5.14: Displacement for before optimization of fork during Braking

5.1.1.2 Modal Frequencies

Result Summary

TABLE 5.9: Results for FEA Analysis before Optimization for Fork Modal Frequencies

Frequency
Mode 1: 1273 Hz
Mode 2: 1353 Hz
Mode 3: 1513 Hz
Mode 4: 1515 Hz
Mode 5: 3339 Hz
Mode 6: 3346 Hz
Mode 7: 4138 Hz
Mode 8: 4292 Hz

Total Modal Displacement

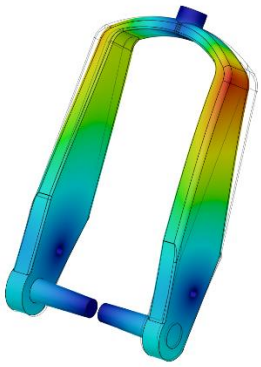

Figure 5.15: Mode 1: 1273 Hz Total Modal Displacement

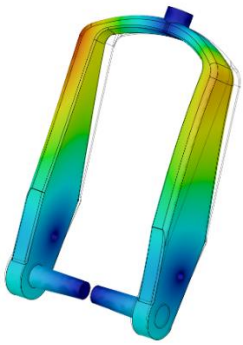

Figure 5.16: Mode 2: 1353 Hz Total Modal Displacement

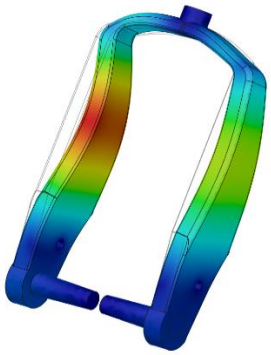

Figure 5.17: Mode 3: 1513 Hz Total Modal Displacement

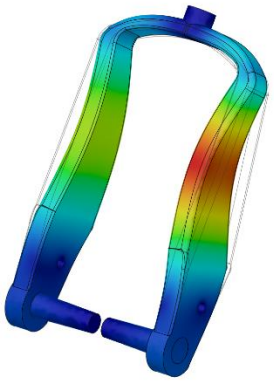

Figure 5.18: Mode 4: 1515 Hz Total Modal Displacement

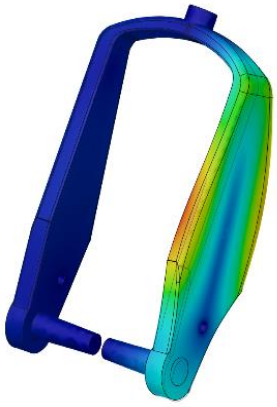

Figure 5.19: Mode 5: 3339 Hz Total Modal Displacement

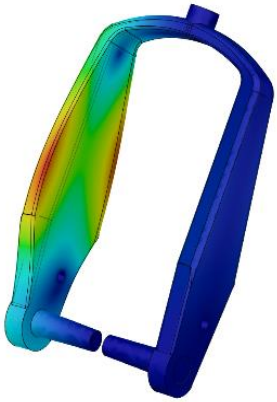

Figure 5.20: Mode 6: 3346 Hz Total Modal Displacement

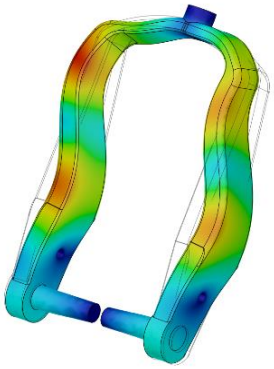

Figure 5.21: Mode 7: 4138 Hz Total Modal Displacement

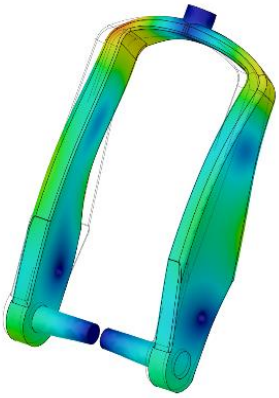

Figure 5.22: Mode 8: 4292 Hz Total Modal Displacement

5.2 FEA ANALYSIS AFTER OPTIMIZATION FOR FORK:

Mesh Details:

TABLE 5.10: Mesh Details of FEA Analysis after Optimization for Fork

Average Element Size (% of model size)		
Average Element Size (absolute value)	5 mm	
Element Order	Parabolic	
Max. Turn Angle on Curves (Deg.)	60	
Max. Adjacent Mesh Size Ratio	1.5	
Max. Aspect Ratio	10	
Minimum Element Size (% of average size)	20	
Type	**Nodes**	**Elements**
Solids	76626	46974

5.2.1 STATIC STESS STUDY:

Scenario: Cornering Towards Right

Result Summary

TABLE 5.11: Results for FEA Analysis after Optimization for Fork Static Stress Study Cornering Towards Right

Name	**Minimum**	**Maximum**
Safety Factor		
Safety Factor (Per Body)	15	15
Stress		
Von Mises	4.611E-06 MPa	17.83 MPa
Displacement		
Total	0 mm	4.144E-04 mm
Reaction Force		
Total	0 N	46.54 N
Equivalent	1.065E-10	4.306E-04

Stress

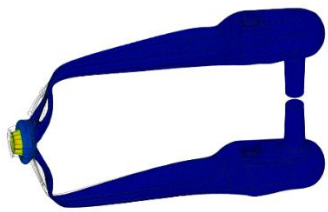

Figure 5.23: Stress for after optimization of fork during cornering towards right

Displacement

Figure 5.24: Displacement for after optimization of fork during cornering towards right

Scenario: Cornering Towards Left

Result Summary

TABLE 5.12: Results for FEA Analysis after Optimization for Fork Static Stress Study Cornering Towards Left

Name	Minimum	Maximum
Safety Factor		
Safety Factor (Per Body)	15	15
Stress		
Von Mises	4.611E-06 MPa	17.83 MPa
Displacement		
Total	0 mm	4.144E-04 mm
Reaction Force		
Total	0 N	46.54 N
Strain		
Equivalent	1.065E-10	4.306E-04

Stress

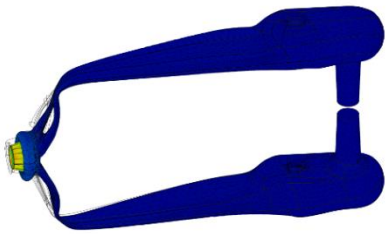

Figure 5.25: Stress for after optimization of fork during cornering towards left

Displacement

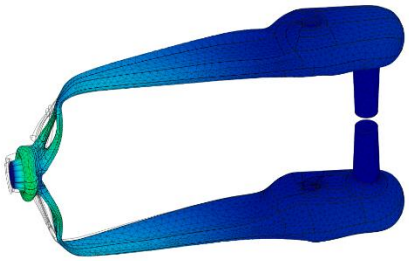

Figure 5.26: Displacement for after optimization of fork during cornering towards left

Scenario: Lateral Imbalance Force towards Right

Result Summary

Table 5.13: Results for FEA Analysis after Optimization for Fork Static Stress Study Lateral Imbalance Force towards Right

Name	Minimum	Maximum
Safety Factor		
Safety Factor (Per Body)	15	15
Stress		
Von Mises	4.611E-06 MPa	17.83 MPa
Displacement		
Total	0 mm	4.144E-04 mm
Reaction Force		
Total	0 N	46.54 N
Strain		
Equivalent	1.065E-10	4.306E-04

Stress

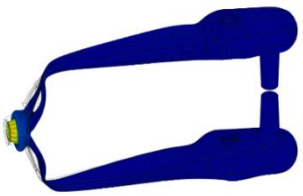

Figure 5.27: Stress for after optimization of fork during Lateral Imbalance Force towards Right

Displacement

Figure 5.28: Displacement for after optimization of fork during Lateral Imbalance Force towards Right

Scenario: Lateral Imbalance towards Left

Result Summary

Table 5.14: Results for FEA Analysis after Optimization for Fork Static Stress Study Lateral Imbalance Force towards Left

Name	Minimum	Maximum
Safety Factor		
Safety Factor (Per Body)	15	15
Stress		
Von Mises	4.611E-06 MPa	17.83 MPa
Displacement		
Total	0 mm	4.144E-04 mm
Reaction Force		
Total	0 N	46.54 N
Strain		
Equivalent	1.065E-10	4.306E-04

Stress

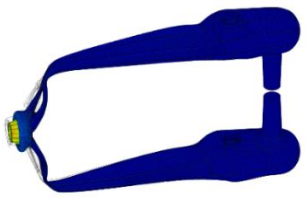

Figure 5.29: Stress for after optimization of fork during lateral Imbalance towards Left

Displacement

Figure 5.30: Displacement for after optimization of fork during lateral Imbalance towards Left

Scenario: Lateral Torsional Force towards Right

Result Summary

TABLE 5.15: Results for FEA Analysis after Optimization for Fork Static Stress Study Lateral torsional Force towards Right

Name	**Minimum**	**Maximum**
Safety Factor		
Safety Factor (Per Body)	15	15
Stress		
Von Mises	4.611E-06 MPa	17.83 MPa
Displacement		
Total	0 mm	4.144E-04 mm
Reaction Force		
Total	0 N	46.54 N
Strain		
Equivalent	1.065E-10	4.306E-04

Stress

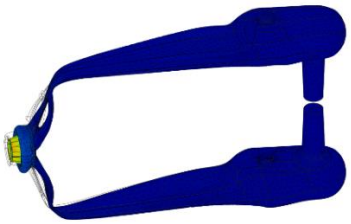

Figure 5.31: Stress for after optimization of fork during lateral torsional force towards Right

Displacement

Figure 5.32: Displacement for after optimization of fork during Lateral torsional Force towards Right

Scenario: Lateral Torsional Force towards Left

Result Summary

TABLE 5.16: Results for FEA Analysis after Optimization for Fork Static Stress Study Lateral torsional Force towards Left

Name	Minimum	Maximum
Safety Factor		
Safety Factor (Per Body)	15	15
Stress		
Von Mises	4.611E-06 MPa	17.83 MPa
Displacement		
Total	0 mm	4.144E-04 mm
Reaction Force		
Total	0 N	46.54 N
Strain		
Equivalent	1.065E-10	4.306E-04

Stress

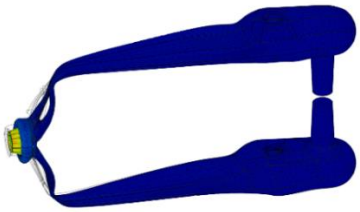

Figure 5.33: Stress for after optimization of fork during Lateral torsional towards Left

Displacement

Figure 5.34: Displacement for after optimization of fork during Lateral torsional towards Left

Scenario: Braking

Result Summary

TABLE 5.17: Results for FEA Analysis after Optimization for Fork Static Stress Study Braking

Name	Minimum	Maximum
Safety Factor		
Safety Factor (Per Body)	15	15
Stress		
Von Mises	4.611E-06 MPa	17.83 MPa
Displacement		
Total	0 mm	4.144E-04 mm
Reaction Force		
Total	0 N	46.54 N
Strain		
Equivalent	1.065E-10	4.306E-04

Stress

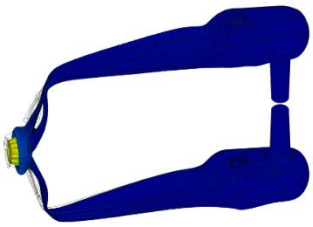

Figure 5.35: Stress for after optimization of fork during Braking

Displacement

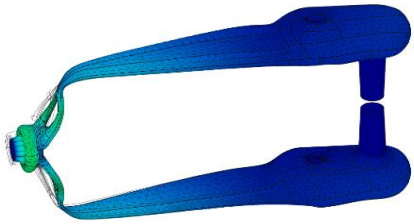

Figure 5.36: Displacement for after optimization of fork during Braking

5.2.2: Modal Frequencies

Result Summary

TABLE 5.18: Results for FEA Analysis after Optimization for Fork Modal Frequencies

Frequency:
Mode 1: 335.1 Hz
Mode 2: 335.2 Hz
Mode 3: 409.2 Hz
Mode 4: 409.3 Hz
Mode 5: 865.7 Hz
Mode 6: 865.9 Hz
Mode 7: 903.2 Hz
Mode 8: 903.5 Hz
Mode 9: 1632 Hz
Mode 10: 1632 Hz

Total Modal Displacement

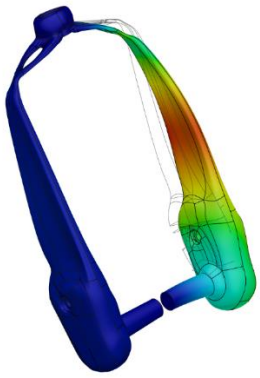

Figure 5.37: Mode 1: 335.1 Hz Total Modal Displacement

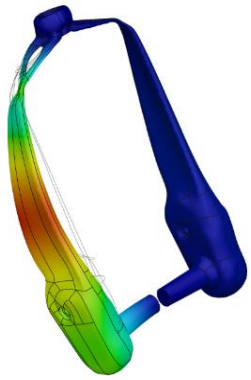

Figure 5.38: Mode 2: 335.2 Hz Total Modal Displacement

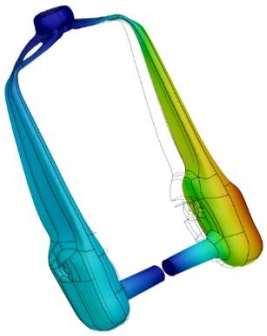

Figure 5.39: Mode 3: 409.2 Hz Total Modal Displacement

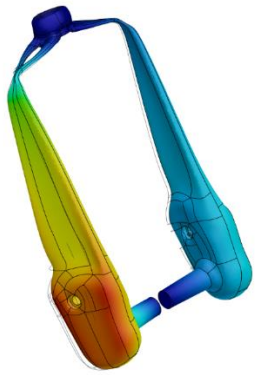

Figure 5.40: Mode 4: 409.3 Hz Total Modal Displacement

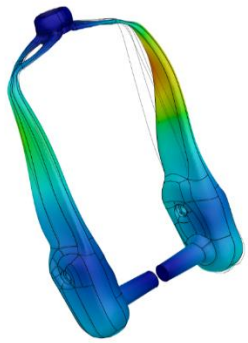

Figure 5.41: Mode 5: 865.7 Hz Total Modal Displacement

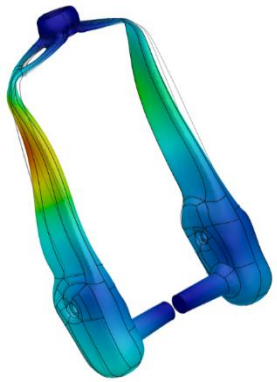

Figure 5.42: Mode 6: 865.9 Hz Total Modal Displacement

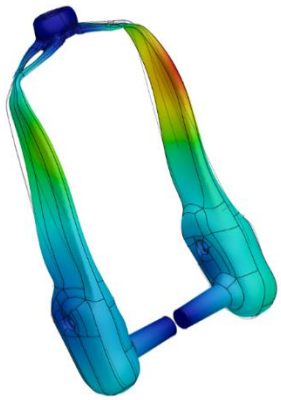

Figure 5.43: Mode 7: 903.2 Hz Total Modal Displacement

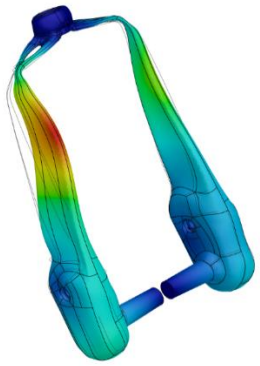

Figure 5.44: Mode 8: 903.5 Hz Total Modal Displacement

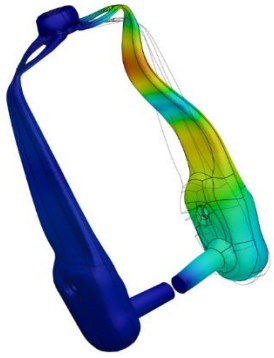

Figure 5.45: Mode 9: 1632 Hz Total Modal Displacement

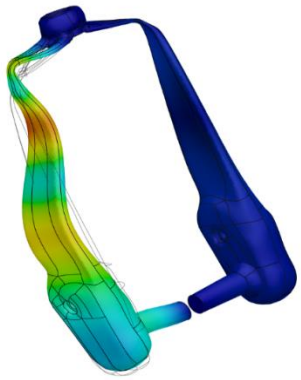

Figure 5.46: Mode 10: 1632 Hz Total Modal Displacement

5.3 FEA ANALYSIS BEFORE OPTIMIZATION FOR WISHBONE:

Mesh Details:

TABLE 5.19: Mesh Details of FEA Analysis before Optimization for Wishbone

Average Element Size (% of model size)		
Average Element Size (absolute value)		10 mm
Element Order		Parabolic
Max. Turn Angle on Curves (Deg.)		60
Max. Adjacent Mesh Size Ratio		1.5
Max. Aspect Ratio		10
Minimum Element Size (% of average size)		20
Type	**Nodes**	**Elements**
Solids	135830	84575

5.3.1 Static Stress

Scenario: Cornering Towards Right

TABLE 5.20: Results for FEA Analysis before Optimization for Wishbone – Static Stress Study – Cornering Towards Right

Name	**Minimum**	**Maximum**
Safety Factor		
Safety Factor (Per Body)	5.675	15
Stress		
Von Mises	7.229E-05 MPa	48.46 MPa
Displacement		
Total	0 mm	0.06414 mm
Reaction Force		
Total	0 N	275.6 N
Strain		
Equivalent	1.095E-09	0.001165

Stress:

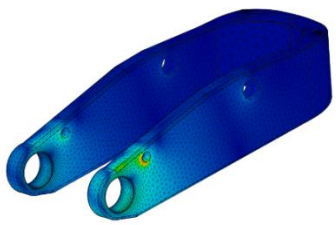

Figure 5.47: Stress for before optimization of Wishbone during cornering towards right

Displacement:

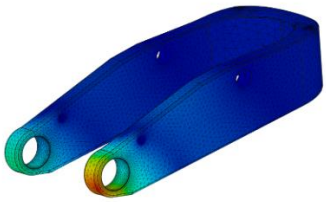

Figure 5.48: Displacement for before optimization of Wishbone during cornering towards right

Scenario: Cornering Towards Left

Result Summary

TABLE 5.21: Results for FEA Analysis before Optimization for Wishbone – Static Stress Study – Cornering Towards Left

Name	Minimum	Maximum
Safety Factor		
Safety Factor (Per Body)	5.675	15
Stress		
Von Mises	7.229E-05 MPa	48.46 MPa
Displacement		
Total	0 mm	0.06414 mm
Reaction Force		
Total	0 N	275.6 N
Strain		
Equivalent	1.095E-09	0.001165

Stress

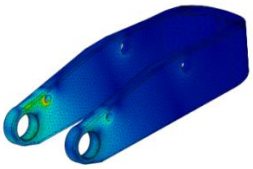

Figure 5.49: Stress for before optimization of Wishbone during cornering towards left

Displacement:

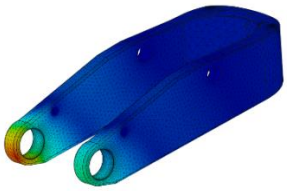

Figure 5.50: Displacement for before optimization of Wishbone during cornering towards left

Scenario: Lateral Imbalance towards Right

Result Summary

TABLE 5.22: Results for FEA Analysis before Optimization for Wishbone – Static Stress Study – Lateral Imbalance Towards Right

Name	**Minimum**	**Maximum**
Safety Factor		
Safety Factor (Per Body)	5.675	15
Stress		
Von Mises	7.229E-05 MPa	48.46 MPa
Displacement		
Total	0 mm	0.06414 mm
Reaction Force		
Total	0 N	275.6 N
Strain		
Equivalent	1.095E-09	0.001165

Stress

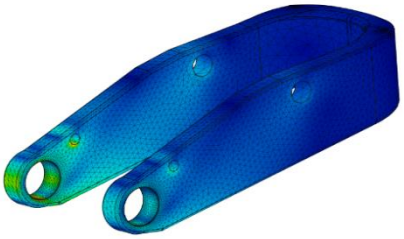

Figure 5.51: Stress for before optimization of Wishbone during Lateral Imbalance Force towards Right

Displacement

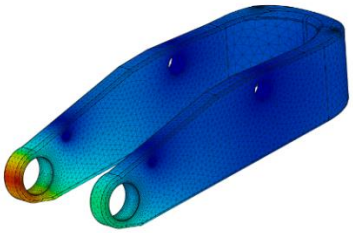

Figure 5.52: Displacement for before optimization of Wishbone during Lateral Imbalance Force towards Right

Scenario: Lateral Imbalance towards Left

Result Summary

TABLE 5.23: Results for FEA Analysis before Optimization for Wishbone – Static Stress Study – Lateral Imbalance towards Left

Name	Minimum	Maximum
Safety Factor		
Safety Factor (Per Body)	5.675	15
Stress		
Von Mises	7.229E-05 MPa	48.46 MPa
Displacement		
Total	0 mm	0.06414 mm
Reaction Force		

Total	0 N	275.6 N
Strain		
Equivalent	1.095E-09	0.001165

Stress

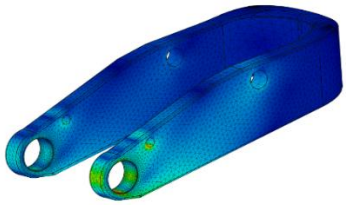

Figure 5.53: Stress for before optimization of Wishbone during Lateral Imbalance towards Left

Displacement

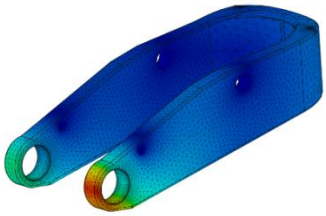

Figure 5.54: Displacement for before optimization of Wishbone during Lateral Imbalance towards Left

Scenario: Lateral Torsional Towards Right

Result Summary

TABLE 5.24: Results for FEA Analysis before Optimization for Wishbone – Static Stress Study – Lateral Torsional towards Right

Name	Minimum	Maximum
Safety Factor		
Safety Factor (Per Body)	5.675	15
Stress		
Von Mises	7.229E-05 MPa	48.46 MPa
Displacement		

Total	0 mm	0.06414 mm
Reaction Force		
Total	0 N	275.6 N
Strain		
Equivalent	1.095E-09	0.001165

Stress

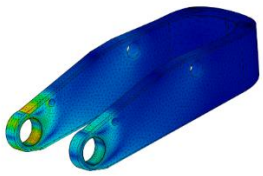

Figure 5.55: Stress for before optimization of Wishbone during Lateral torsional Force towards Right

Displacement

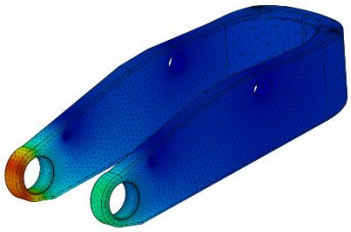

Figure 5.56: Displacement for before optimization of Wishbone during Lateral torsional Force towards Right

Scenario: Lateral Torsional Towards Left

TABLE 5.25: Results for FEA Analysis before Optimization for Wishbone Static Stress Study Lateral torsional towards Left

Name	**Minimum**	**Maximum**
Safety Factor		
Safety Factor (Per Body)	5.675	15
Stress		
Von Mises	7.229E-05 MPa	48.46 MPa
Displacement		

Total	0 mm	0.06414 mm
Reaction Force		
Total	0 N	275.6 N
Strain		
Equivalent	1.095E-09	0.001165

Stress

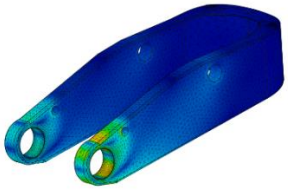

Figure 5.57: Stress for before optimization of Wishbone during Lateral torsional towards Left

Displacement

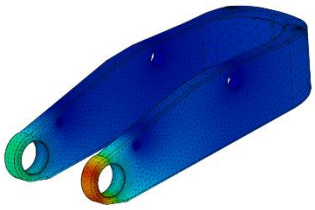

Figure 5.58: Displacement for before optimization of Wishbone during Lateral torsional towards Left

Scenario: Braking

Result Summary

TABLE 5.26: Results for FEA Analysis before Optimization for Wishbone

Static Stress Study Braking

Name	**Minimum**	**Maximum**
Safety Factor		
Safety Factor (Per Body)	5.675	15
Stress		
Von Mises	7.229E-05 MPa	48.46 MPa

Displacement		
Total	0 mm	0.06414 mm
Reaction Force		
Total	0 N	275.6 N
Strain		
Equivalent	1.095E-09	0.001165

Stress

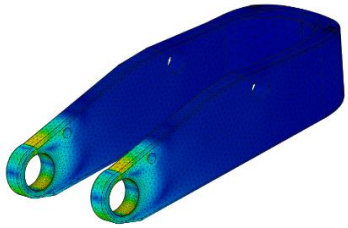

Figure 5.59: Stress for before optimization of Wishbone during Braking

Displacement

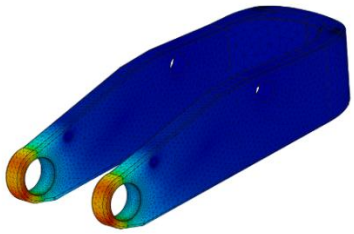

Figure 5.60: Displacement for before optimization of Wishbone during Braking

5.3.2 Modal Frequencies

Result Summary

TABLE 5.27: Results for FEA Analysis before Optimization for Wishbone

Modal Frequencies

Frequency
Mode 1: 1557 Hz
Mode 2: 2048 Hz
Mode 3: 3081 Hz
Mode 4: 3257 Hz
Mode 5: 3352 Hz
Mode 6: 3743 Hz
Mode 7: 3744 Hz
Mode 8: 4023 Hz

Total Modal Displacement

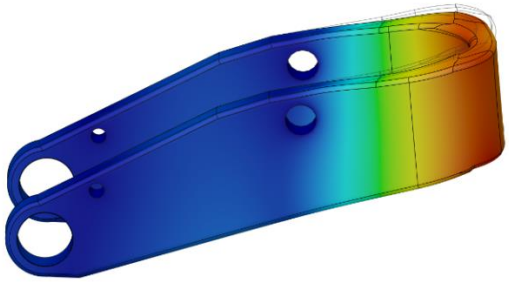

Figure 5.61: Mode 1: 1557 Hz Total Modal Displacement

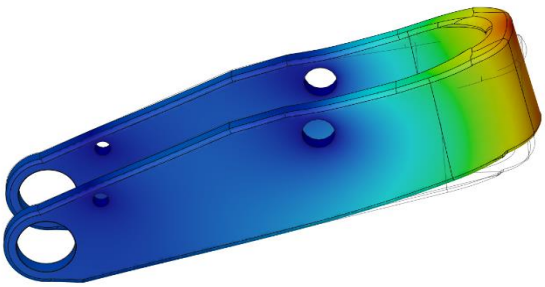

Figure 5.62: Mode 2: 2048 Hz Total Modal Displacement

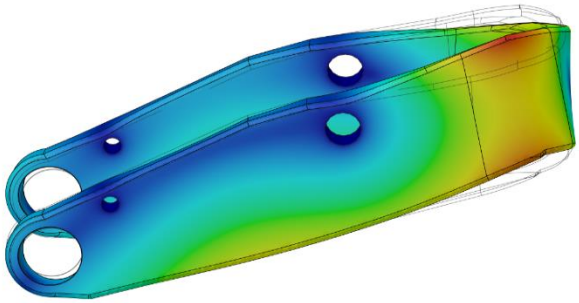

Figure 5.63: Mode 3: 3081 Hz Total Modal Displacement

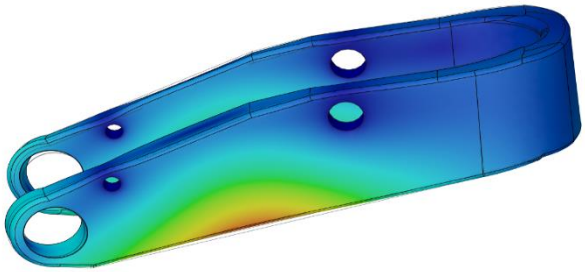

Figure 5.64: Mode 4: 3257 Hz Total Modal Displacement

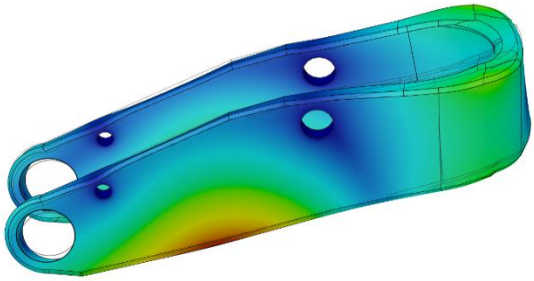

Figure 5.65: Mode 5: 3352 Hz Total Modal Displacement

Figure 5.66: Mode 6: 3743 Hz Total Modal Displacement

Figure 5.67: Mode 7: 3744 Hz Total Modal Displacement

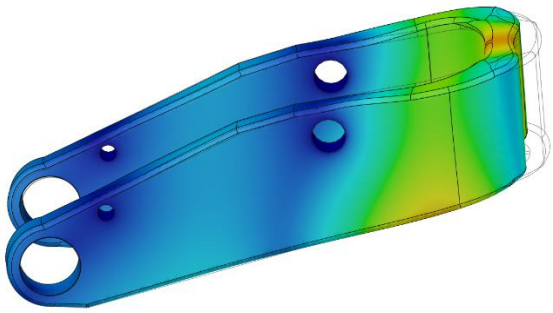

Figure 5.68: Mode 8: 4023 Hz Total Modal Displacement

5.4 FEA ANALYSIS AFTER OPTIMIZATION FOR WISHBONE:

5.4.1 Study: Static Stress

Mesh

TABLE 5.28: Mesh Details of FEA Analysis after Optimization for Wishbone

Average Element Size (% of model size)	
Average Element Size (absolute value)	10 mm
Element Order	Parabolic
Max. Turn Angle on Curves (Deg.)	60
Max. Adjacent Mesh Size Ratio	1.5
Max. Aspect Ratio	10
Minimum Element Size (% of average size)	20

Scenario: Cornering Towards Right

Result Summary

TABLE 5.29: Results for FEA Analysis after Optimization for Wishbone Static Stress Study Cornering Towards Right

Name	Minimum	Maximum
Safety Factor		
Safety Factor (Per Body)	0.0701	15
Stress		
Von Mises	1.226E-05 MPa	3923 MPa
Displacement		
Total	0 mm	0.02211 mm
Reaction Force		
Total	0 N	221.3 N
Strain		
Equivalent	2.469E-10	0.093

Stress

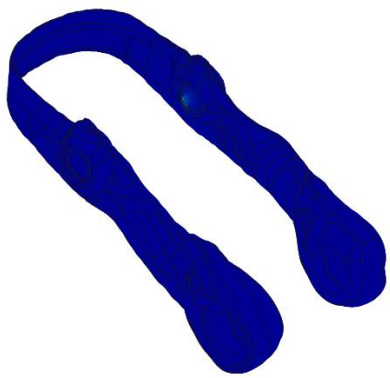

Figure 5.69: Stress for after optimization of Wishbone during cornering towards right

Displacement

Figure 5.70: Displacement for after optimization of Wishbone during cornering towards right

Scenario: Cornering Towards Left

Result Summary

TABLE 5.30: Results for FEA Analysis after Optimization for Wishbone – Static Stress Study – Cornering towards Left

Name	**Minimum**	**Maximum**
Safety Factor		
Safety Factor (Per Body)	0.0701	15
Stress		
Von Mises	1.226E-05 MPa	3923 MPa
Displacement		
Total	0 mm	0.02211 mm
Reaction Force		
Total	0 N	221.3 N
Strain		
Equivalent	2.469E-10	0.093

Stress

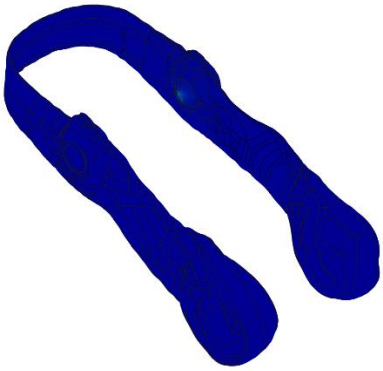

Figure 5.71: Stress for after optimization of Wishbone during cornering towards left

Displacement

Figure 5.72: Displacement for after optimization of Wishbone during cornering towards left

Scenario: Lateral Imbalance Force towards Right

Result Summary

TABLE 5.31: Results for FEA Analysis after Optimization for Wishbone – Static Stress Study – Lateral Imbalance Force towards Right

Name	Minimum	Maximum
Safety Factor		
Safety Factor (Per Body)	0.0701	15
Stress		
Von Mises	1.226E-05 MPa	3923 MPa
Displacement		
Total	0 mm	0.02211 mm

Reaction Force		
Total	0 N	221.3 N
Strain		
Equivalent	2.469E-10	0.093

Stress

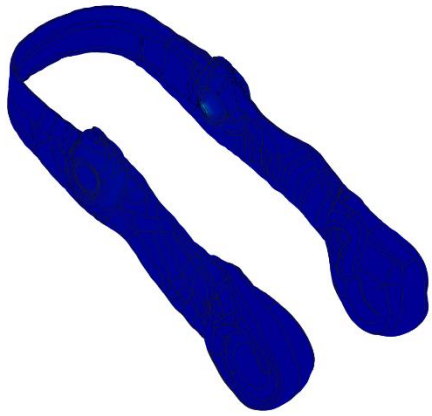

Figure 5.73: Stress for after optimization of Wishbone during Lateral Imbalance Force towards Right

Displacement

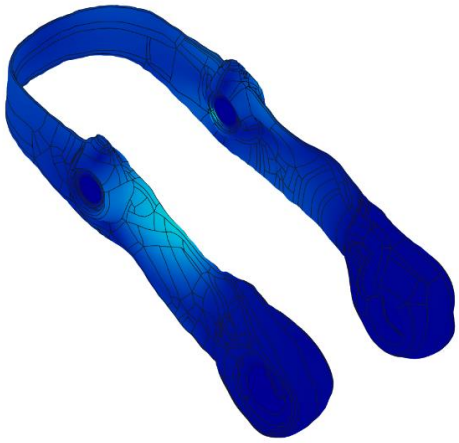

Figure 5.74: Displacement for after optimization of Wishbone during Lateral imbalance Force towards Right

Scenario: Lateral Imbalance Force Left

Result Summary

TABLE 5.32: Results for FEA Analysis after Optimization for Wishbone – Static Stress Study – Lateral Imbalance Force towards Left

Name	Minimum	Maximum
Safety Factor		
Safety Factor (Per Body)	0.0701	15
Stress		
Von Mises	1.226E-05 MPa	3923 MPa
Displacement		
Total	0 mm	0.02211 mm
Reaction Force		
Total	0 N	221.3 N
Strain		
Equivalent	2.469E-10	0.093

Stress

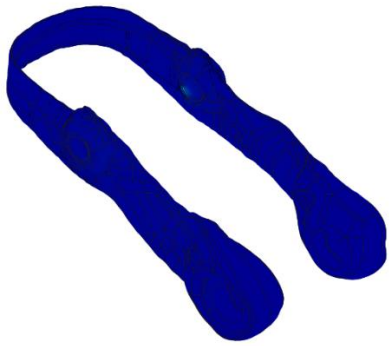

Figure 5.75: Stress for after optimization of Wishbone during Lateral Imbalance towards Left

Displacement

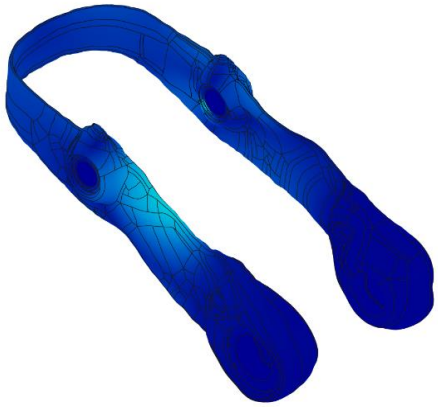

Figure 5.76: Displacement for after optimization of Wishbone during Lateral Imbalance towards Left

Scenario: Lateral Torsional Force towards Right

TABLE 5.33: Results for FEA Analysis after Optimization for Wishbone – Static Stress Study – Lateral torsional Force towards Right

Name	Minimum	Maximum
Safety Factor		
Safety Factor (Per Body)	0.0701	15
Stress		
Von Mises	1.226E-05 MPa	3923 MPa
Displacement		
Total	0 mm	0.02211 mm
Reaction Force		
Total	0 N	221.3 N
Strain		
Equivalent	2.469E-10	0.093

Stress

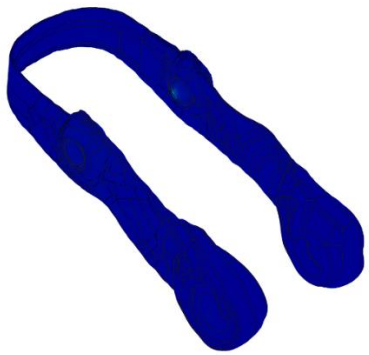

Figure 5.77: Stress for after optimization of Wishbone during Lateral torsional Force towards Right

Displacement

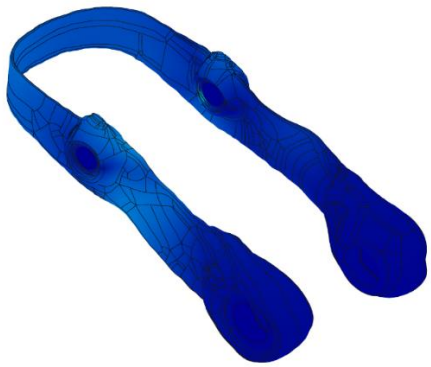

Figure 5.78: Displacement for after optimization of Wishbone during Lateral torsional Force towards Right

Lateral Torsional Force towards Left

Result Summary

TABLE 5.34: Results for FEA Analysis after Optimization for Wishbone – Static Stress Study – Lateral torsional Force towards Left

Name	Minimum	Maximum
Safety Factor		
Safety Factor (Per Body)	0.0701	15
Stress		
Von Mises	1.226E-05 MPa	3923 MPa
Displacement		
Total	0 mm	0.02211 mm
Reaction Force		
Total	0 N	221.3 N
Strain		
Equivalent	2.469E-10	0.093

Stress

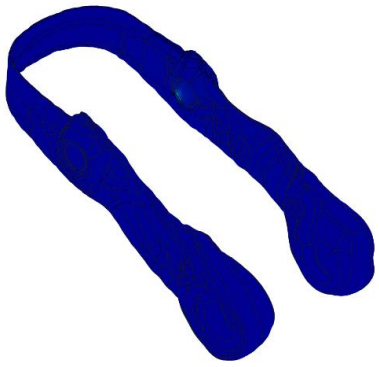

Figure 5.79: Stress for after optimization of Wishbone during Lateral torsional towards Left

Displacement

Figure 5.80: Displacement for after optimization of Wishbone during Lateral torsional towards Left

Braking Force

Result Summary

TABLE 5.35: Results for FEA Analysis After Optimization for Wishbone – Static Stress Study – Braking

Name	Minimum	Maximum
Safety Factor		
Safety Factor (Per Body)	0.0701	15
Stress		
Von Mises	1.226E-05 MPa	3923 MPa
Displacement		
Total	0 mm	0.02211 mm
Reaction Force		
Total	0 N	221.3 N
Strain		
Equivalent	2.469E-10	0.093

Stress

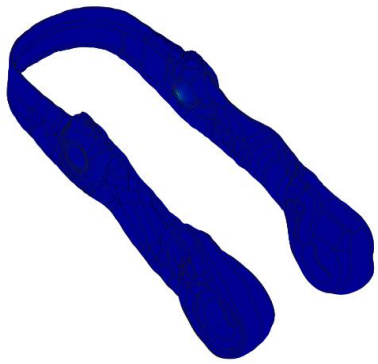

Figure 5.81: Stress for after optimization of Wishbone during Braking

Displacement

Figure 5.82: Displacement for after optimization of Wishbone during Braking

5.4.2 Modal Frequencies

Result Summary

TABLE 5.36: Results for FEA Analysis after Optimization for Wishbone Modal Frequencies

Frequency
Mode 1: 423.3 Hz
Mode 2: 1036 Hz
Mode 3: 1287 Hz
Mode 4: 1974 Hz
Mode 5: 2151 Hz
Mode 6: 2621 Hz
Mode 7: 2965 Hz
Mode 8: 2991 Hz
Mode 9: 3054 Hz
Mode 10: 3477 Hz

Total Modal Displacement

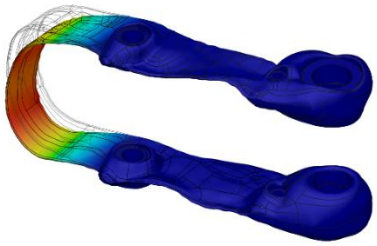

Figure 5.83: Mode 1: 423.3 Hz Total Modal Displacement

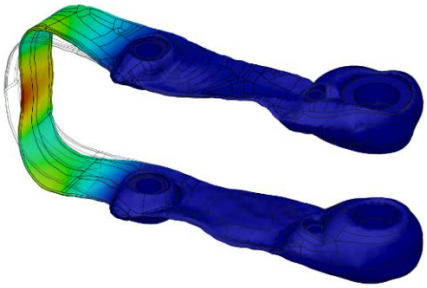

Figure 5.84: Mode 2: 1036 Hz Total Modal Displacement

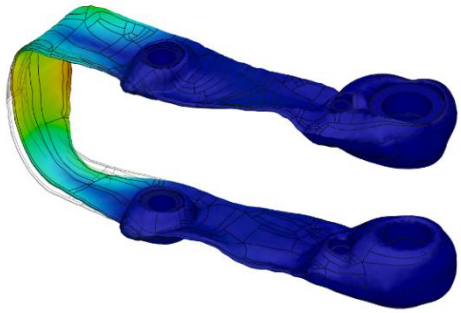

Figure 5.85: Mode 3: 1287 Hz Total Modal Displacement

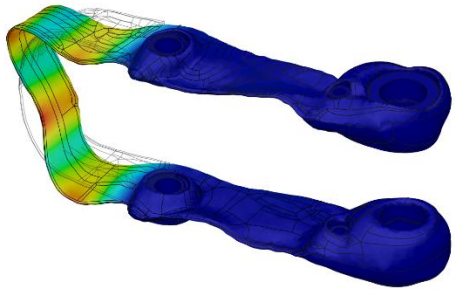

Figure 5.86: Mode 4: 1974 Hz Total Modal Displacement

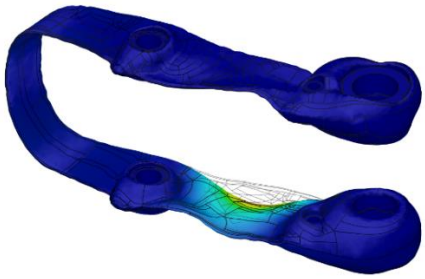

Figure 5.87: Mode 5: 2151 Hz Total Modal Displacement

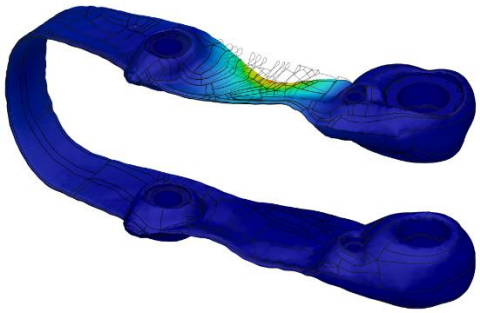

Figure 5.88: Mode 6: 2621 Hz Total Modal Displacement

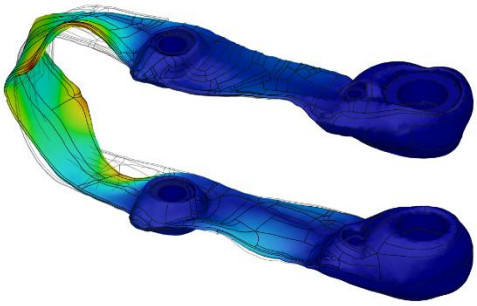

Figure 5.89: Mode 7: 2965 Hz Total Modal Displacement

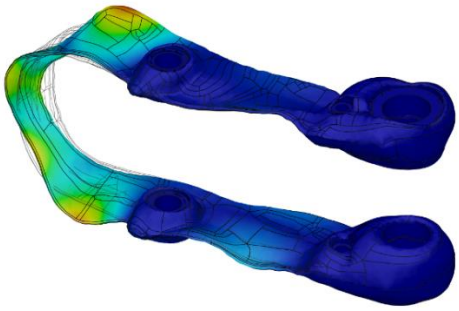

Figure 5.90: Mode 8: 2991 Hz Total Modal Displacement

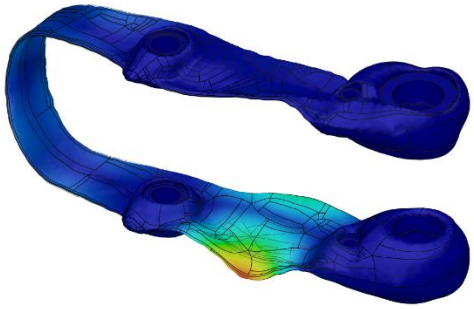

Figure 5.91: Mode 9: 3054 Hz Total Modal Displacement

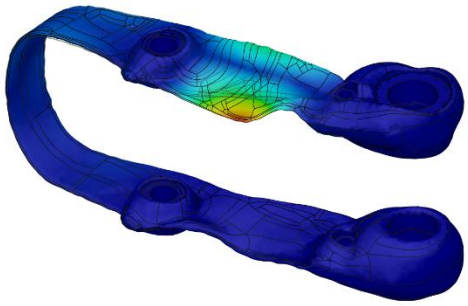

Figure 5.92: Mode 10: 3477 Hz Total Modal Displacement

5.5 COMPARISON STUDY ON BOTH PARTS:

Table 5.37: Comparison Study on Both Parts

CATEGORY	FORK BEFORE OPTIMIZATION	FORK AFTER OPTIMIZATION	WISHBONE BEFORE OPTIMIZATION	WISHBONE AFTER OPTIMIZATION
FOS	2.252	4	2.67	3.07
STRESS (MPa)	122.1	17.83	48.46	39.23
DISPLACEMENT (mm)	0.0033199	4.44*10^-4	0.0644	0.2211
REACTION FORCE (N)	123.4	46.54	275.6	221.3
STRAIN	0.001929	4.306*10^-4	0.001165	0.093
MASS (KG)	1.91	1.521	2.543	1.941
AVG. MODAL FREQUENCY (Hz)	2177.875	2093.681	3100.625	2997.7

Thereafter, we have compared the parts before and after optimization of suspension systems, after the comparison, the optimized fork and wishbone show better results but it has a complicated design. The design of the optimized double-wishbone shows excellent results to the two-wheeler, and the material Aluminium AlSi10Mg is used to make the suspension system light in weight. To design this suspension system we should know the dimensions of the chassis, their rack angle, wheelbase, and cornering speed, curb weight, etc. to get the perfect design and it minimizes the jerks, drag, and continuous surface contact with the track while racing.

From this, we can conclude that the optimized parts (fork & wishbone) have got improved than the previous design because the parameters show improved results.

CHAPTER 6

CONCLUSION

From the Finite Element Analysis or FEA Analysis we got to know that the optimized design shows us with good performance results for the Fork and Wishbone. After designing and simulation the optimized Fork and Wishbone geometry using Autodesk Fusion 360 software, mass and FOS are also obtained. The given stress and deformation results for the optimized parts of the double wishbone suspension system are within limit of acceptable factor of safety limit which confirms the design and analysis of the suspension components have been achieved. Designing the suspension is more complicated, it must have the proper procedure.

In this paper, we have discussed the important points for optimizing the design and analysing the double wishbone suspension system performance in various scenario. From the simulation results, it is clear that using generative design technique the weight of the double wishbone suspension system has been reduced according to the kerb weight of the motorcycle. Since, mass is one of the important criterion in the racing and cruiser bikes, as mass gets reduced the vehicles gets to travel faster, hence generative design can be helpful to reduce mass up to 8% of the double wishbone suspension system.

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APPENDIX

Plagiarism Report

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R (37150127) who carried out the project entitled ◆DESIGN OPTIMIZATION OF DOUBLE WISHBONE SUSPENSION SYSTEM FOR MOTORCYCLE◆ under my supervision from November 2020 to March 2021. Internal Guide Dr.G.Arunkumar, M.E., Ph.D Head of the department Dr.G.Arunkumar, M.E., Ph.D

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