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Design, Development and Computational Finite Element Analysis (FEA) of an Electric Two-Wheeler Frame

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Abstract: The primary objective of this paper is to describe the complete design procedure for developing a frame of an electric two-wheeler. This chassis work concentrates more on the frame component and its functionalities. A complete and step-by-step approach is facilitated in the design. The process involves identification of the master layout of the vehicle with the determination of centre of gravity and dimensional constraints. Material selection for the frame is done by adopting Ashby's approach and principles of mechanics of solids are utilized in tube dimension calculations. The paper centralizes effectively on the Computer Aided Engineering (CAE) by adopting computational Finite Element Analysis (FEA). The FEA includes static structural analysis of the frame under various loading conditions, modal analysis, harmonic analysis and impact analysis. The computational aid combined with the analytical methods help the efficient and effective design, development and optimization of the frame structure.
Keywords: Structural analysis, Modal analysis, Ashby's approach, FEA, Electric two-wheeler.

I. INTRODUCTION

Carbon dioxide measured at NOAA's Mauna Loa Atmospheric Baseline Observatory peaked for 2022 at 421 parts per million in May 2022, pushing the atmosphere further into territory not seen for millions of years, scientists from NOAA and Scripps Institution of Oceanography at the University of California San Diego announced today. NOAA's measurements of carbon dioxide at the mountaintop observatory on Hawaii's Big Island averaged 420.99 parts per million (ppm), an increase of 1.8 ppm over 2021. Prior to the Industrial Revolution, CO2 levels were consistently around 280 ppm for almost 6,000 years of human civilization. Today India is the fourth largest emitter of Green House Gases (GHG) and contributes 7.08% of the global emissions. Transportation plays a crucial role in emissions. Globally, it contributes nearly 305.3 MtCO2e – 0.64 per cent of all GHG emissions. While in India, this sector is the fastest-growing source of carbon emissions. The world is purging towards electrified mobility, but that does not happen overnight. The average count of motorized two-wheelers in India is 49.7% of the total vehicles on road. This high influence of two-wheeler usage among the people can be the best entry level electrification. When this 49.7% is electrified completely, then there is a chance for 100% sustainable mobility and reduction in GHG. The Electric bike which will be running on battery, the power is supplied by the motor, thereby supplying this power to drive other gear components. The main purpose of using this E-bike is that it is user friendly, economical and relatively cheap. The efficiency of this system undeniable compared to conventional modes of transport. The Indian government aims to transform the future to be electric, and that by 2030 about 80% of the two and three wheeler market be electrified. Considering all these aspects into account engineers should design, develop, and optimise, prototype and manufacture electric two-wheelers of highest quality to meet the needs. The manufactures should ensure maximum durability and efficiency, with utmost importance to the static-dynamic stability, manoeuvrability and ergonomic comfort. This paper concentrates more on the chassis part of the vehicle, which plays the role of a backbone to any vehicle and helps to achieve dynamic stability. The chassis of two-wheeler comprises frame, wheels and brakes and suspension. The style of the two-wheeler depends upon its chassis and the chassis is the important carriage system of the vehicle.

II. FRAME OF THE ELECTRIC TWO WHEELER VEHICLE

The frame acts as a skeleton and supports the major components and systems by taking various loads of the bike. Different components are mounted on the frame providing them with strength to carry their specific individual loads. The frame also supports various components like seat, bodyworks, accessories, etc. Battery is also mounted on the frame making it even more crucial loading. The frame must be able to resist against shocks and impacts of the vehicle and provide stiffness thus protecting the user and vital parts of the vehicle.

The design of the frame also depends on the transmission, steering and suspension. The chassis is the backbone of the two-wheeler; it must support all the vehicle subassemblies as well as protect the driver. The chassis design is crucial to the success of any project because if the chassis fails, that puts the vehicle and the driver at tremendous risk. The goal of the frame will be to protect the driver, offer sturdy mounting for all subsystems, maintain all safety rules and regulations, and still be lightly designed (material optimization). This paper provides the work of computational design and analysis of the frame used in Electric bike in a detailed approach.

III. MASTER LAYOUT, MATERIAL SELECTION AND DIMENSIONS

Master layout of any vehicle is the schematic of the whole vehicle with the associated systems, to scale. The layout is a great value addition process in marketing, manufacturing and testing of the vehicle for further optimizations. In our project it consists of the location of all aggregate with respect to wheel centres, frame, suspension, brakes, fork, fender, handlebar, and battery pack with motor. It also specifies the vehicle dimensions with respect to wheelbase, ground clearance and handlebar height with respect to ground. The magnitude of these parameters, determine major performance of the vehicle under dynamic running.

A. *Master Layout Of The Electric Two-Wheeler Vehicle*

The master layout given here for the vehicle is a detailed description of the positioning and location of the various features of the frame. This layout helps the designer to plan the functional requirements of the frame. The analysis of the frame requires these force acting points for further study.

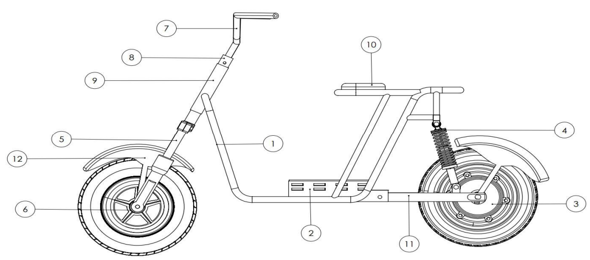


Figure 1: Master layout

TABLE 1
PARTS OF THE LAYOUT

Part	Part Name
1	Frame (tubular)
2	Battery pack
3	Motor (hub-drive)
4	Shock absorbers
5	Telescopic
6	Drum brake
7	Steering
8	Fork column
9	Headstock (frame)
10	Driver seat
11	Swing arm
12	Fender

TABLE 2
DIMENSIONAL CONSTRAINTS

Mark	Parameters	Dimensions
A	Wheel Base	1232.05 mm
B	Ground Clearance	278.12 mm
C	Seat Height	755.61 mm
D	Handle Bar Height	1063.52 mm

B. *Calculating The Centre Of Gravity From The Free Body Diagram*

The Centre of Gravity (CG) of any vehicle is the point that sums up the vehicle's mass in one central point. It is the average location of the theoretical focal point and enhances the stability of the ride. In our case we consider the kerb weight as 65kg and other relative component weights to find the coordinates of CG with respect to the front wheel centre.

According to the composite body equations, (All Dimensions in mm)

X Component (X') = ∑x *WI ÷ ∑WI

Y Component (Y') = ∑y *WI ÷ ∑WI

- CG coordinate from ground (vertical) = 395.54 mm
- CG coordinate from front wheel (Horizontal) = 654.72 mm

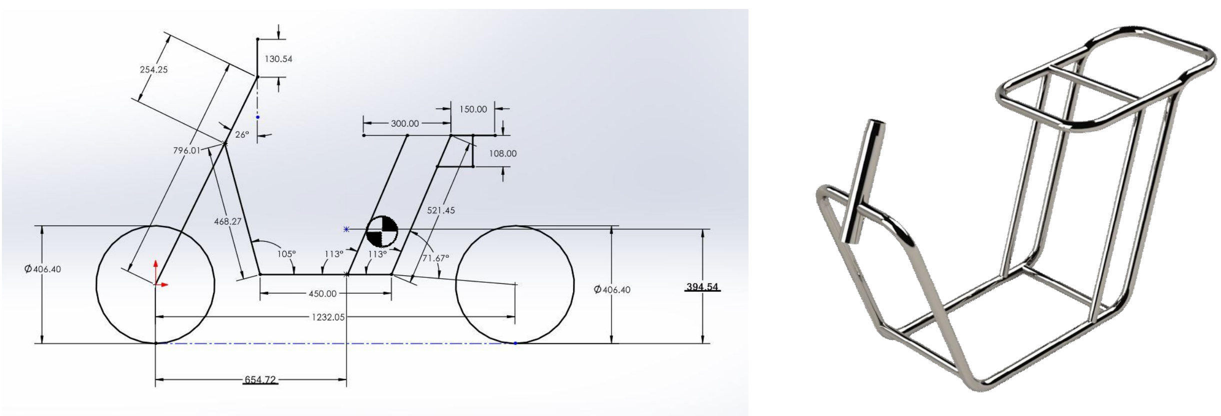


Figure 2: Free Body Diagram of the Vehicle and CAD model of the frame

C. *Material Selection for the Frame*

Frame is a crucial component from safety point of view, the material used for frame in the vehicle should have very stable and reliable mechanical properties under varying conditions of load. A powerful tool for helping in materials selection is offered by the Ashby plot. This is a scatter plot displaying at the same time two properties of materials or classes of materials. It is convenient because it gives useful information not only on which material displays the highest (or the lowest) property reported on the x- or y-axes, but also which one presents the highest ratio between the two properties. Materials and manufacturing are closely linked in determining final product performance. Two Primary Requirements of Our Design: *Stiffness and Strength.*

Candidate Materials: Chrome-moly (chrome molybdenum) steel-AISI 4130, Mild steel (AISI 1020 steel), Aluminium, Titanium, CFRP (carbon fibre reinforced polymer) and KFRP (Kevlar fibre reinforced polymer).

Considering stiffness and light materials for cantilever design, deriving the material Parameters using deflection equation and substituting for mass, to attain greater stiffness for lesser weight of material. Material Index needs to be minimized and performance index needs to be maximized.

$$m = \pi L\left[\frac{4FL^3}{3\pi\delta}\right]\left(\frac{\rho}{E^{0.5}}\right)$$

TABLE 3
Analysis of Candidate Materials

Materials	Young's Modulus, E	Density, ρ	Performance Index, √E/ρ	Performance Index
	G-pa	**g/Cm³**	**C**	C
Chrome-moly Steel (AISI 4130)	200	7.85	1.80155	1.8
Mild Steel (AISI 1020)	205	7.87	1.81929	1.81
Aluminium	69	2.7	3.07653	3.07
Titanium	116	4.5	2.39341	2.39
CFRP	190	1.6	8.61503	**8.61**
KFRP	40	1.4	4.51754	4.51

Based on Stiffness-considering Young's Modulus and Density, maximum performance index, c = 8.61 (CFRP), best cost to performance ratio = 35.91 (AISI 1020 mild steel). The derived Performance Index can be written. The intercept of this Equation is 2LogC. Considering C of AISI 1020 Steel we get 2LogC=0.51 and Slope=2

$$LogE - 2Log\rho + 2LogC$$

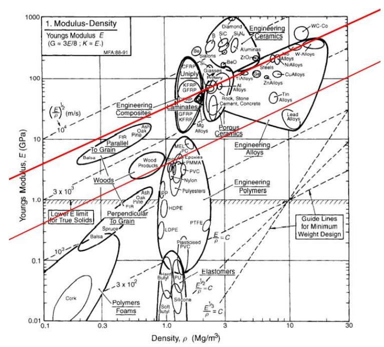


Figure 3: Ashby's performance mapping

The plot area gives us the best possible materials for the design. Ashby's approach paved us the way to filter 3 material candidates, AISI 1020 mild steel- ERW, Aluminium and Chromium steel. Now considering these three for availability and cost analysis we got,

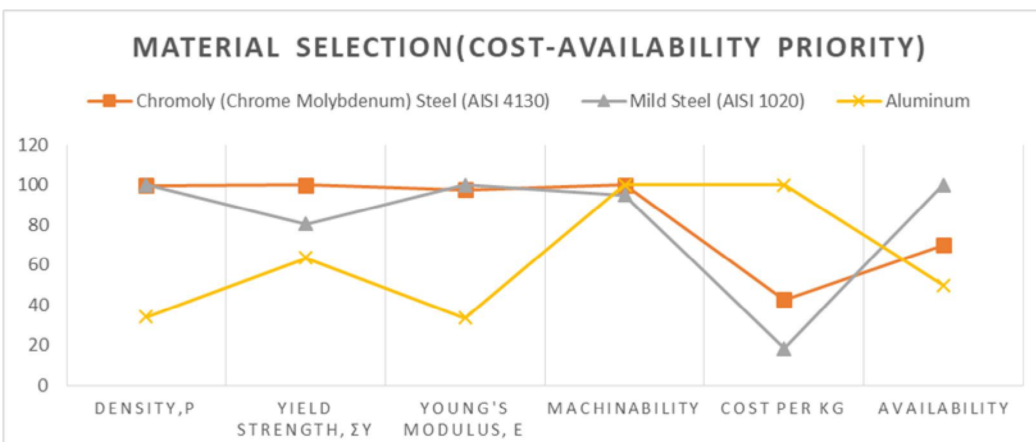


Figure 4: Cost Analysis of Materials

Based on the ASHBY's approach of Strength-Stiffness priority on Performance Index, the material candidates were sorted and analysed. The practical properties of 3 materials were taken into consideration, such as Cost, Availability and Machinability, and the optimal material for our project is selected, from the results. Material selected for the frame is Mild steel (AISI 1020 low carbon steel).

Mild steel is the type of carbon steel that has lower levels of carbon in it. When the carbon content in the alloy is 0.05 to 0.309%, it is considered as the mild carbon steel or low carbon steel. It doesn't include other elements and is purely an Iron-Carbon Alloy.

- Mild Steel is comparatively lighter than some High and Some Medium carbon steels, due to less carbon content.
- Considering our application this material is fore-front recommended due to their high Ductility, Malleability, Manufacturability and Weld-ability.
- The tubular forms of the material is available in the market in all standard dimensions, easy availability compared to Aluminium and Chrome-moly.
- Mild Steel is one material which keeps the project budget as low as possible, at the same time providing stiffness and strength. They are highly affordable to meet the required cost criteria.

After the detailed material selection analysing by ASHBY's approach and comparative study of the candidates by cost-performance indices, our team comes to the decision of using AISI 1020 Steel-Mild Steel (ERW-Electric Resistance Welded) for the Electric Two Wheeler's frame. This decision is based on analysis and practical research, by giving priority to cost, strength and availability. This material meets our design objectives and will perform well in the prototyping stage of the project.

D. Calculation for Tube Diameters

Let us consider three primary forces which are acting on the frame members,

- Force at the seat member 5 due to the rider = 1079.1 N
- Force at foot member 3 (40% of rider weight + battery weight) = 431.64 N
- Force at the Handlebar (30% of rider weight) = 323.73

These forces create stress on the frame and from which the moments can be calculated and the optimum diameter of the circular cross section can be determined.

Let R1 and R2 be the reaction forces at the front wheel and rear wheel centre of the vehicle, these reactions directly transmit to the frame endpoints. The relative distances are taken from the FBD of the master layout.

$$R2 = \frac{(348.95)(323.73) + (583.60)(431.64) + (862.44)(1079.1)}{1232.05} = 1051.55N$$

$$R1 = 323.73 + 431.64 + 1079.1 - 1051.55 = 782.92N$$

Stress calculation by distortion theory,

$$\sigma = \frac{0.57*UTS}{2} = 119.7\,\frac{N}{m^2}\ (UTS = 420\ \frac{N}{m^2})$$

We know that the bending moment equation can be used with respect to the polar moment of inertia (I) which for a circular section gives the diameter, here we consider the members of the frame as separate beams and applied, Where y=d, D=2.5+2.5+d (d=ID and D=OD)

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

From the values of the OD and ID calculated by assuming the members of the frames as beams, we can observe that the primary heavy load bearing members require higher values of OD and the values ranges between 19.75 mm to 32.98 mm. So, in order to have a stiff frame we must consider having circular tubes between this ranges.

For frame tube dimensions we can verify the manually calculated values by computational numerical analysis (FEA) using ANSYS mechanical.

Modal data, material and boundary conditions of the frame considerations are given by,

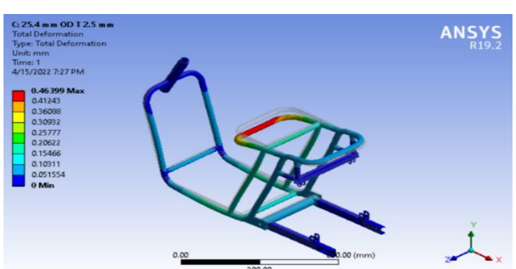
Material	: Mild Steel AISI 1020 (ERW)
Material Properties	: As Given before
Considered Outer Diameters (OD), mm	: 22, 25.4, 26, 28, 30
Considered Thickness (T), mm	: 2, 2.5 mm
Analysis System	: Static Structural (Linear Properties)
Mesh Type	: Program Controlled (Quadratic type)

Boundary Conditions (Fixed Supports) : Front Suspension Fixed, Rear Suspension Fixed
Degrees of Freedom : 6 Degrees allowed
Modal Analysis Boundary Condition : Rear Wheel Fixed
Number of Mode Shapes : 6 Mode Shapes
Natural Frequency for Comparing : First Mode Natural Frequency
Force 1 : 735 N (Person Weight)
Force 2 : 220.5 N (Load on Steering Column)
Force 3 : 414 N (Load on foot member & Battery Weight)
Force 4 :196 N (Luggage Weight)
Force Applied Position : Seat Member, Steering Column, Foot Member, rear- Luggage Member Along negative Y-Axis (Vertically down)
Acceleration due to gravity Condition : 1g

The study on the different outer diameter of the frame and the thickness, gives us comparison on the strength, stiffness, mass, Factor of Safety and vibrational characteristics. From the tabulated comparison, for our scenario of electric two-wheeler the best suited dimension based on the considered parameters are taken.

Table 4
Tube diameter by FEA. comparing different combinations of od and thickness of the tubular pipes

S. No	OD	Thickness	Maximum Stress	Maximum Deformation	Volume
	(mm)	(mm)	(M-pa)	(mm)	(mm³)
1	22	2	73.404	0.8903	9.61E+05
2	25.4	2.5	50.48	0.4639	1.39E+06
3	25.4	2	74.59	0.5983	1.25E+06
4	26	2	40.887	0.4431	1.26E+06
5	28	2	60.927	0.3747	1.35E+06
6	30	2	71.858	0.3153	1.42E+06
S. No	Mass	Seat Stiffness	Factor Of Safety	First Mode Natural Frequency	Mode 1 Maximum Deformation
	(Kg)	(N/mm)	(no unit)	(Hz)	(mm)
1	7.567	825.56	4.7	10.88	25.161
2	10.937	1584.39	6.9	10.651	18.713
3	9.8187	1228.48	4.6	10.565	19.664
4	9.877	1658.76	8.5	11.139	20.04
5	11.6	1961.56	5.7	10.877	18.565
6	12.18	2331.11	4.8	10.649	17.477



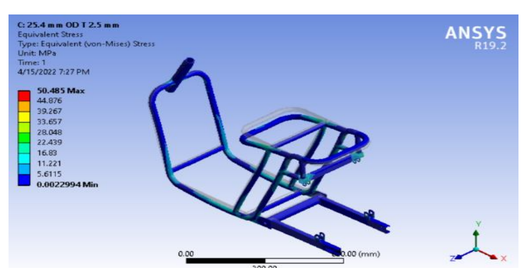


Figure 5: Total deformation and equivalent stress of 25.4 mm OD 2.5 mm thick tube

From the manual calculations and the FEA analysis, the tube diameter and the thickness of the frame for our application in electric two-wheeler can be concluded as 25.4mm outer diameter and 2.5mm thickness.

IV. CAE ANALYSIS OF THE FRAME

A. Aim and application of CAE in a two-wheeler vehicle

The chassis of a two-wheeler comprises of frame, suspension and brakes. The whole dynamics of the vehicle depends on the chassis and more importance must be given to frame. Our aim is to have an optimized design of the vehicle frame, which is the load bearing component of the vehicle. The deflection, vibration and energy absorbing characteristics are to be studied and the design must be optimized. The primary objective is to analyse the frame of the two-wheeler. These analysis results yield, immense value addition to the design and development.

B. Finite element analysis (FEA) software

ANSYS Mechanical is an easy-to-use finite element-based simulation tool that is very robust and tightly integrated with ANSYS' broad array of Multi-physics solvers. With a range of analytical tools, users can take their mechanical design end-to-end, from the geometry phase through initial proof of concept simulations, optimization, and validation. For this project we use ANSYS mechanical for better and faster design decisions while solving complex structural engineering problems.

C. Material data of the frame

The material used for the manufacturing of the frame of two-wheeler is AISI 1020-ERW (Electric Resistance Welding)-Mild Steel and the properties of this material are. For all the analysis performed on frame uses this material.

Table 5
Material Properties: AISI 1020 MS ERW

Material Properties	Metrics
Condition	Linear Elastic
Directional Property	Isotropic Elastic
Density	7870 Kg/m³
Young's Modulus	205 G-pa
Poisson's Ratio	0.29 µ
Bulk Modulus	162 G-pa
Shear Modulus	79 G-pa
Tensile Strength, Ultimate	420 M-Pa
Tensile Strength, Yield	350 M-pa
Elongation at break (in 50mm)	15%
Hardness, Brinell	121
Thermal Expansion Coefficient	13.13E-6 (1/K)
Thermal Conductivity	24.5 J/m. s. K
Specific Heat Capacity	7.2 J/Kg. K
Carbon	0.2
Silicon	0.21
Copper	0.01

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D. FEA Of the frame using ANSYS

1) Static structural analysis of the frame under 1g condition

A static structural analysis calculates the effect of steady (or static) loading conditions on a structure, while ignoring inertia and damping effects, such as those caused by time varying loads. The results help us to determine the maximum stress acting region and the load-deflection characteristics. Model details, material data, boundary and initial conditions are given as,

Material:	Mild Steel AISI 1020 (ERW)
Material Properties:	Table 7.1
Considered Outer Diameter (OD), mm:	25.4
Considered Thickness (T), mm:	2
Analysis System:	Static Structural (Linear Properties)
Mesh Type:	Program Controlled (Quadratic type)
Boundary Conditions (Fixed Supports):	Front Suspension Fixed, Rear Suspension Fixed
Degrees of Freedom:	6 Degrees allowed
Force 1:	735 N (Person Weight)
Force 2:	220.5 N (Load on the Steering Column)
Force 3:	414 N (Load on the foot member & Battery Weight)
Force 4:	196 N (Luggage Weight)
Force Direction:	Along negative Y-Axis (Vertically down)
Force Application:	Seat Member, Steering Column, Foot Member, Rear Luggage Member
Acceleration Due to Gravity:	1g condition
Time:	1 s
Volume:	1.4127e+006 mm³
Mass:	11.118 kg
Nodes:	84092
Elements:	42800

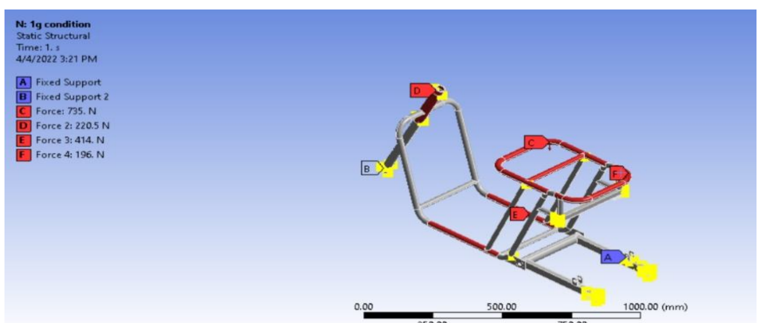

Figure 6: Boundary conditions for static structure

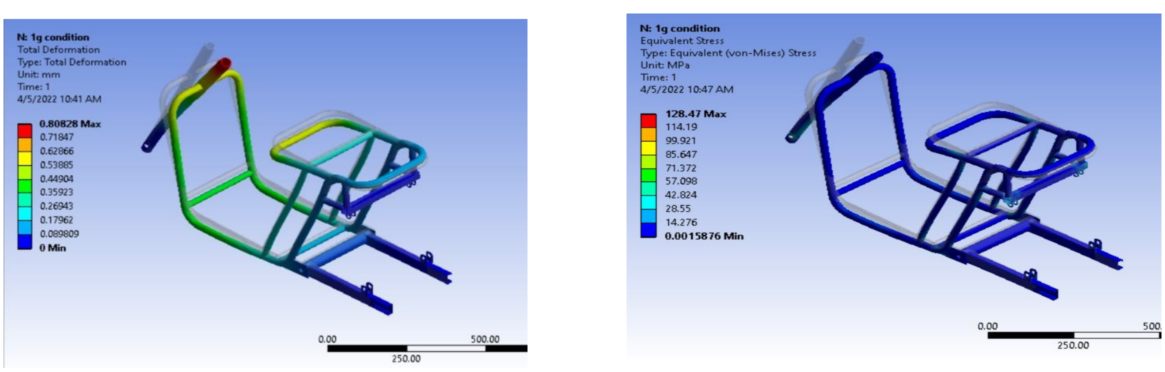

Figure 7: Total deformation and von-Mises stress under 1g condition as results

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Maximum Deformation: 0.828 mm
Average Deformation: 0.22087 mm
Maximum Stress Induced: 128.47 M-Pa

Maximum Stress Induced Point: From the static loading analysis it is found that the maximum stress is developed at the rear and front suspension mountings. The inference is that the sprung mass is taken by the suspension and the energy absorption takes place effectively. Such that sprung mass energy absorption is the role of the suspension system and with respect to these stress levels the suspension systems are designed.

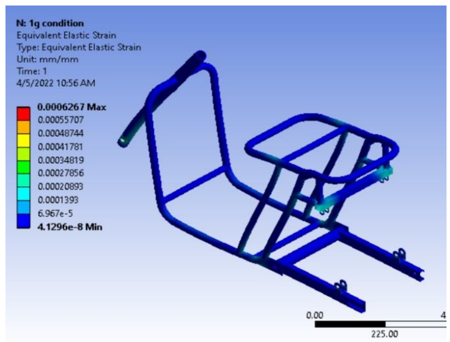

Figure 8: Equivalent elastic strain

Maximum Strain: 6.267e-004 mm/mm
Maximum Strain Induced at: Front and Rear suspension members

2) Conclusion of static structural analysis under 1g Condition

It is observed that the maximum resistance is provided in the form of stress at the front and rear suspension, and the members of the frame are very stiff and deforms very less with a factor of safety of 2.724. Hence the frame is stiff at 1g condition and more importance must be given for the suspension selection.

3) Results of static structural analysis of the frame under 1g, 2g, 3g and 4g loading

Comparing the response of the frame using same material and geometry, but by changing the loading condition with respect to the acceleration due to gravity: (1g,2g,3g,4g Conditions)

Table 6
Response of the Frame under 1g, 2g, 3g and 4g Loading

RESULTS	1g condition	2g condition	3g condition	4g condition
Max. Total Deformation (mm)	0.828	1.574	2.355	3.359
Max. Equivalent Stress (M-pa)	128.47	251.51	377.23	535.88
Max. Equivalent Strain (mm/mm)	6.27E-04	6.27E-04	1.84E-03	2.61E-03
Stiffness (N/mm) Seat Member	1170.38	1203.33	1203.33	1125.22
Factor of Safety	2.724	1.3191	0.92	0.653

The summary of these analyses is that the frame is structurally stiff and strong when loads under 1g and 2g conditions (i.e.1 and 2 riders) are under action. The FOS is 2.724 and 1.319 for the conditions respectively, and these values are practically significant. However, 3g and 4g loading is an assumed state and practically it is avoidable in a two-wheeler, their FOS's are minimum and structurally weak. Thus, the loading conditions conclude the frame design is structurally safe and optimum response is achieved.

a) Modal analysis of the frame

We can use Modal Analysis to determine the vibration characteristics (natural frequencies and mode shapes) of a part or assembly. We can also notice the response to the natural frequencies of our model when it is subjected time-dependent and/or oscillatory/vibration loads by running any dynamic analysis: like Dynamic Frequency analysis. A modal analysis is prerequisite for a dynamic analysis.

Basics of Free Vibration Analysis » For a free vibration analysis, the natural circular frequencies, and mode shapes are calculated from:

$$[K] - \omega i^2 [M] \{\varphi i\} = 0$$

Assumptions:

- [K] and [M] are constant
- Linear elastic material behaviour is assumed
- Damping is not included
- {F} is not present, so no excitation of the structure is assumed
- The structure can be constrained or unconstrained Mode shapes {f} are relative values, not absolute

The results of modal analysis (deformation, Stress) are Arbitrary & are shown as scaled values by software for mathematical reasons & should not be considered real. These mode shapes help us to study, deformation patterns with increasing frequencies. Also, are used to determine frequency critical when the part or assembly is under working condition. In order to get realistic values for results, after modal analysis is performed where the geometry is subjected to actual load certain frequency.

b) Modal analysis of the frame under constrained no pre-stress condition

Pre-Stress: None
Mode-Shapes: 10
Constrained: Fixed support at the rear wheel mount (swing arm to wheel)
First Natural Frequency: 10.174 Hz

Maximum Deformation at Natural Frequency: 18.64 mm (23.3 times more deflection than that of the deformation caused due to normal leading conditions)

Table 7
Modal Analysis Results for the Frame (Non-Stressed)

Eigen-Vectors (Mode Shapes)	Eigen-Values(Natural Frequency)	Mode Type	Maximum Deformation
1	10.174 Hz	Bend Out	18.64 mm
2	16.403 Hz	Torsional	18.76 mm
3	29.682 Hz	Bend In	25.88 mm
4	35.134 Hz	Torsional	15.55 mm
5	44.715 Hz	Torsional	24.43 mm
6	51.895 Hz	Seat Bend	17.73 mm
7	73.562 Hz	Forward Bend	18.64 mm
8	91.909 Hz	Torsional	18.76 mm
9	94.412 Hz	Bending	25.87 mm
10	123.56 Hz	Torsional	15.57 mm

The vibratory behaviour of the frame under no pre-stress conditions leads to 10 mode shapes and the Eigen values are observed. From the 10 Eigen values the first natural frequency is 10.174 Hz and the deflection at this value is 23.3 times more than the usual loading conditions, so the excitation frequency must not match with this frequency and these values can further be used for dynamic analysis.

c) Modal analysis of the frame under constrained pre-stress condition

Load Application and Boundary Conditions similar to static loading under 1g condition.
Pre-Stress: Yes, 1g condition loads
Constrained: Fixed support at the rear and front suspension mountings
Mode Shapes: 10

Table 8
Modal Analysis Results for the Frame (Pre-Stressed)

Eigen-Vectors (Mode Shapes)	Eigen-Values(Natural Frequency)	Maximum Deformation
1	42.382 Hz	30.59 mm
2	64.212 Hz	29.16 mm
3	75.634 Hz	23.13 mm
4	150.6 Hz	16.21 mm
5	164.21 Hz	27.14 mm
6	174.58 Hz	22.82 mm
7	207.1 Hz	31.15 mm
8	253.22 Hz	35.09 mm
9	259.41 Hz	19.98 mm
10	294.87 Hz	29.90 mm

The results of this have the same behaviour as the previous one, but with loading conditions. The first natural frequency is 42.38 Hz and is 4 times more than the non-stressed condition, this shows that the suspension excitation plays a very crucial role in the dynamic conditions to avoid resonance under these frequency excitations

4) Harmonic analysis of the frame

ANSYS Mechanical APDL and Mechanical Workbench can perform harmonic analysis on a structure, determining the steady-state sinusoidal response to sinusoidal varying loads all acting at a specified frequency. In a harmonic analysis, the peak response will correspond with natural frequencies of the structure or the resonant conditions where the frequencies of Corresponding load matches with natural frequencies of geometry are critical zones where geometry is prone to damage. However, Results of frequency carried forward from modal to harmonic analysis gives us a rough idea of natural frequencies range of geometry & actual resonance conditions with harmonic analysis results.

Harmonic response analysis of the frame using the data from the modal analysis conducted under 1g loading and boundary conditions,

Forces acting on the Frame: Person weight on the seat, 30% Load on the Handle bar, battery weight on foot member of the frame and 40% load on the foot member

Physics Type: Structural
Frequency Spacing: Linear
Range: 0-60 Hz
Solution Method: Mode Superposition Method
User Defined Frequencies: 10.174 Hz, 16.403 Hz
Constant Damping Ratio: 0.02
Deformation Axis: Y-axis

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Tabular Data

	Frequency [Hz]	Amplitude [mm]
1	2.	20.172
2	4.	22.889
3	6.	29.554
4	8.	50.119
5	10.	367.62
6	10.174	476.44
7	12.	48.002
8	14.	20.924
9	16.	12.558
10	16.403	11.541
11	18.	8.5759
12	20.	6.286
13	22.	4.8255
14	24.	3.8386
15	26.	3.1737

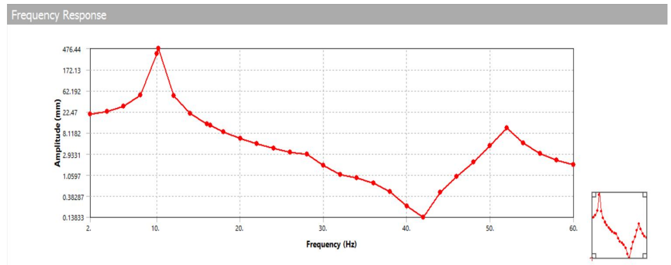

Figure 9: Frequency Response of the Frame with Respect to the Total Deformation along Y-Axis (Frequency vs. Amplitude)

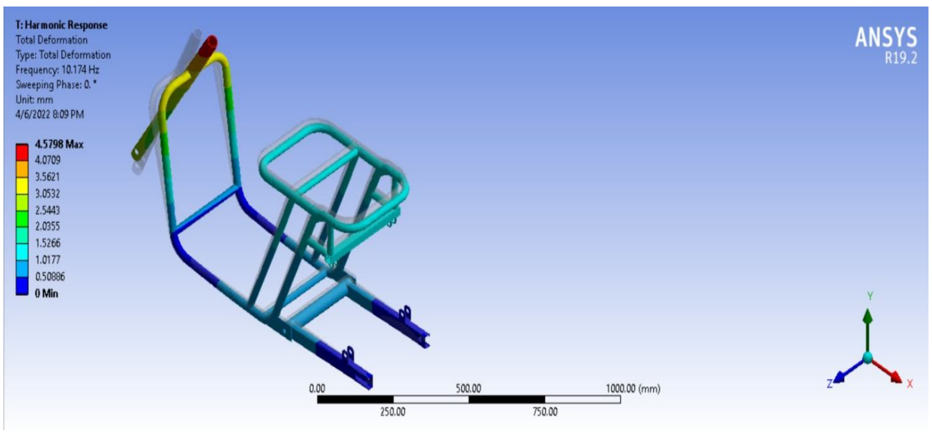

Figure 10: Harmonic Response- Total Deformation at 10.174 Hz

Total Deformation at 10.174 Hz: 4.57 mm
Maximum Stress Induced at 10.174 Hz: 291 M-Pa
From this it is clear that at the first mode natural frequency 10.174 Hz the amplitude-476.44 mm rises to the extreme deformation and thus the value represents the natural frequency, this frequency thus needs to be considered in the design and must be avoided. Thus the mode shapes and natural frequencies of the frame are determined using the modal and harmonic response analysis. These values help us in designing the optimum frame dimensions. The points and mesh where more stress are induced are taken into account and better manufacturing of those components are to be performed.

5) Frontal forced impact analysis of the frame

Weight of the Vehicle (Gross Weight):	150 kg
Maximum Speed of the Vehicle:	35kmph (9.722 m/s)
Impact Time:	0.85 seconds
Collision Type:	Perfectly Inelastic Collision
Fixed Support:	Rear End Wheel of the vehicle
Front Impact Loading:	Maximum stress is expected to occur at the front suspension mounting point
Formula for force of impact:	F= m*v/t
Frontal impact force:	1715.64 N
Force Direction:	+X-axis

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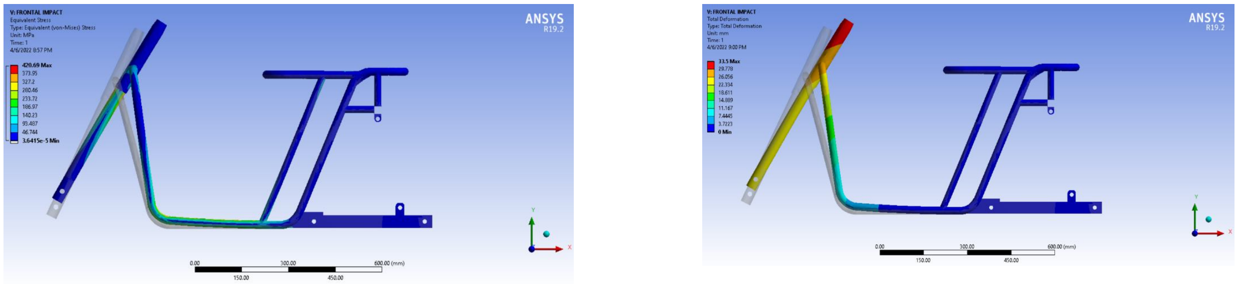

Figure 11: Total deformation and Equivalent-Stress Induced under Frontal Impact Condition (M-Pa)

Maximum Deformation: 33.5 mm
Maximum Stress Induced: 420.69 M-Pa

- Total Deformation under Frontal Impact condition leads to more deformation on the front suspension mountings as expected.
- Equivalent-Stress induced under Frontal Impact condition leads to more stress on the front suspension mountings as expected.

6) Rear forced impact analysis of the frame

Weight of the Vehicle (Gross Weight):	150 kg
Maximum Speed of the Vehicle:	35kmph (9.722 m/s)
Impact Time:	0.85 seconds
Collision Type:	Perfectly Inelastic Collision
Fixed Support:	Front End Wheel of the vehicle
Front Impact Loading:	Maximum stress is expected to occur at the Rear suspension mounting point and rear members
Formula for force of impact:	F= m*v/t
Frontal impact force:	1715.64 N
Force Direction:	-X-axis (negative x-axis)

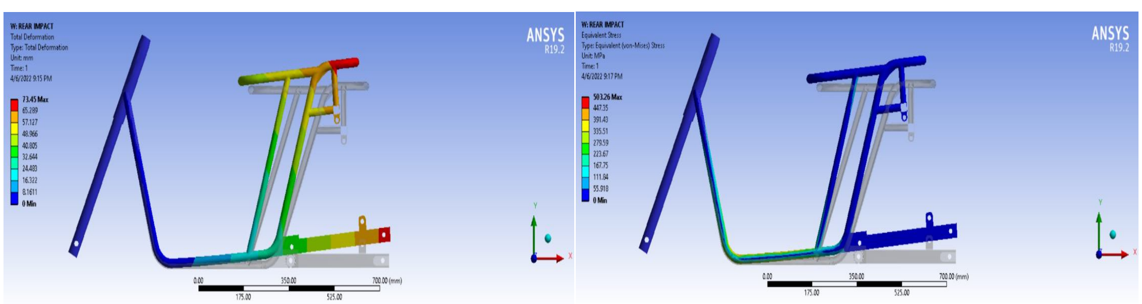

Figure 12: Total deformation and Equivalent-Stress Induced under rear Impact Condition (M-Pa)

Total Deformation and Equivalent Stress under Rear Impact condition leads to more deformation on the Rear members as expected. These analyses are very significant for crashworthiness. The safety under passive conditions is achieved only through structurally strong frames and these results pave us the way to better safety.

E. Conclusion of CAE analysis

Computational Finite element Analysis methodologies are always a great means to solve mathematical models. In our paper these methods helped in solving many structural and crashworthy problems assisted with many sub-systems of the vehicle. Especially it played a very crucial role in the frame manufacturing, and required many simulations with respect to this structure and gave optimal results. The values obtained also helped us to change the design with respect various stress actions and total deformations. Thus, CAE Analysis helped us to visualize the problem and solve it in a very effective way with minimal consumption of time and cost.

V. CONCLUSIONS

"Science can amuse and fascinate us all, but it is engineering that changes the world", these were the words of Isaac Asimov. Engineering is any sufficiently advanced technology which is indistinguishable from magic. Electric two wheelers play the most role in transforming Indian mobility towards electrification and the chassis of the frame plays the utmost role in the quality of the process. This paper mainly focuses on the frame of the electric two-wheeler and its features. The process of the PLM in this case started with the development of master layout of the application, which includes consideration of dimensional constraints, components position and determination of the centre of gravity. This assessment helped in evaluating the stability and manoeuvrability of the vehicle. Then the process concentrates on the frame design. The frame design starts with the determination of AISI 1020 MS ERW as the optimal material for the application, followed by estimating the tubular dimensions as 25.4mm Outer diameter and 2.5mm thickness for the tubular frame. The results were optimized by using CAE FEA tools. Then the design was analysed using ANSYS FEA simulation, where in the static structural 1g analysis gave us the factor of safety as 2.72. The maximum loading under 2g, 3g and 4g conditions were also simulated. Vibrational resistance and resonance conditions simulated in modal and harmonic analysis were a great medium to identify the vibrational characteristics and their responses. The crashworthiness of the frame was done by forced frontal and rear crash tests. These values helped the work to identify that the headstock region of the frame was strained more and a change in design to it was adapted. Thus design optimisation and process improvement were justified under CAE. The holistic process workflow gave highest productivity in design, development and prototyping of the frame structure of electric two-wheeler. .

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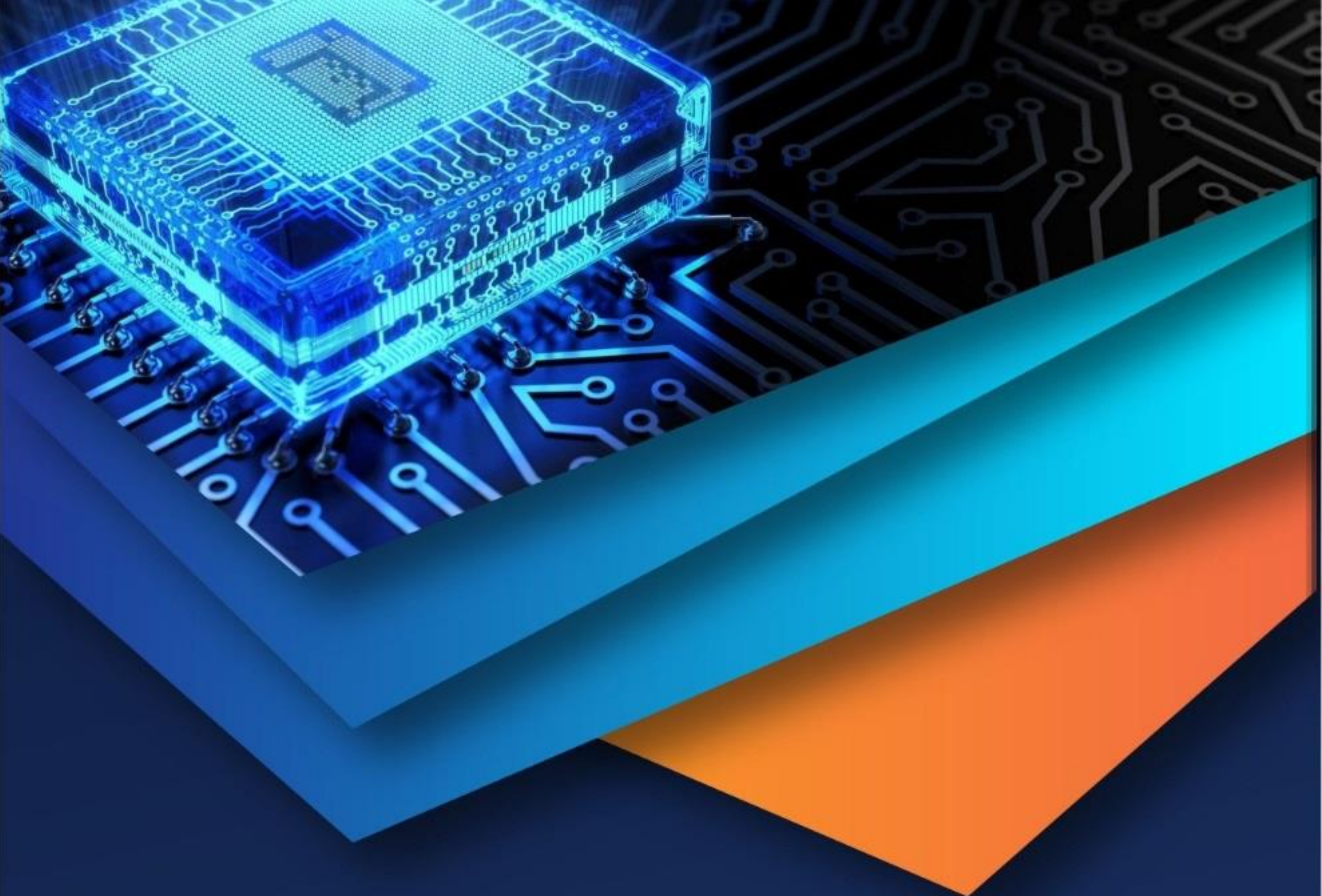

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