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"Design and Analysis of Four Wheeled Electrically Powered Mobility Vehicle"

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Abstract: *People with movement disabilities have numerous challenges in life, two of the most significant of which are first dependency on others and secondly mobility restrictions. As a result, people are unable to enjoy simple and personal activities such as shopping, going for a drive, riding on off-road terrain, and going on camping vacations, among other things. Though there are already various solutions available, such as wheelchairs, manual and electric wheelchairs, in this paper, we design and develop an open, four-wheeled, electrically powered mobility vehicle to address the aforementioned difficulties. This approach will seek to address the inadequacies of present choices by offering people with movement limitations a secure, comfortable, and relatively cost-effective form of transportation. The goal of this project is to include the advantages of current available options, such as wheelchair seating, compact design of mobility scooters and wheelchairs, electric power source and speed of mobility scooters, and overcome their challenges, such as long range, more speed, stability of a four-wheeled chassis, off-road driving capability, sufficient storage space, comparable price range, and so on.*
Keywords*: Movement Disability: Mobility Vehicle: Electric Vehicle: Ladder chassis: FEA Analysis*.

I. Introduction

In this project we are going to design and analyse the four wheeled electrically powered mobility vehicle for the people with movement disabilities. Currently, there are a few solutions accessible on the market for impaired people's movement. Manually operated wheelchairs, electrical wheelchairs, and mobility scooters are the three types. However, a quick investigation reveals that each of them has its own set of flaws and issues. It was observed that the most of the movement disabled people cannot afford the mobility scooters because of its high cost so they are using the manual wheelchairs. Efforts required for moving the manual wheelchairs are more and it is so much tiring for the movement disabled people. So to overcome this problem we are going to design an electrically powered mobility vehicle whose price is between the price of manual wheelchairs and mobility scooters. So this project is designed to give the comfortable ride at considerably good speed and with less maintenance and the main thing is that most of the cycle parts are used in this vehicle so the person can easily repair and maintain the vehicle in the nearby cycle shop if any problem has arise. The main advantage of electric vehicle is that it is silent and smooth to drive and comfortable chair with sufficient space is provided with some storage space at the rear end of the vehicle.

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Proposed Setup

Figure no 1. Actual model of electric vehicle

Problem Definition:

As we all know, people with movement disabilities confront a slew of issues, the most serious of which being mobility limits. According to the 2011 census, there are around 54 lakh people with a mobility disability in India. As a result, many people are unable to enjoy the same mobility, flexibility and independence as others. Wheelchairs, manual wheelchairs, and other assistive devices are available. Wheelchairs, both manual and electric, mobility scooters, and other aids are available, but they are not without their drawbacks as an example:

1. Assistance is required.
2. Cannot be used over long distances.
3. Must rely on others for long distance travel or public transportation, which has unique characteristics for disabled people but is not always functional.
4. Manual effort is required to drive wheelchairs or tricycles.
5. Most of the current options are immobile on different terrains.
6. No storage space is available on current options.

Objectives:

1. To design and develop an electrically powered four wheeled mobility vehicle for people with movement disabilities.
2. To eliminate challenges regarding manual wheelchairs and the shortcomings of an electric wheelchair.
3. To provide an easy means of transport and hence independence to people with movement disability.
4. To provide better speed, maneuverability and transport means with no pollution.
5. To make the vehicle compact and provide sufficient storage space.
6. To give a good battery life and easy repair and maintenance.

II. Literature Survey

Table no 1: Summary of literature survey

Sr. No.	Name of the Author	Year of Publication	Work Done
1.	Yagyansh Mishra, et. al	August 2020	The authors have provided methodology for selection of various materials and cross sections for the chassis. Also procedure of design and analysis of a ladder chassis is

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			explained.
2.	Francis, et. al	April 2014	The authors have provided a reference for the structural analysis carried out in ANSYS simulation software.
3.	Kumar. A. Hari et. al	2016	The authors have studied the design and analysis of Eicher E2 truck and provided the calculation methodology for design of ladder frame chassis and analysis
4.	Ramesh kumar, et.al	March 2017	The authors have provided a reference for design procedure and optimization of a steering system using Ackermann geometry, also work has been done on optimization of current steering geometry.
5.	Khan noor mohammad, et. al	May 2018	The authors have given a simplified method of designing a steering system using Ackermann steering geometry. The paper describes process of design and fabrication of a steering system.
6.	Pinjarla, et.al	May 2018	Studied various analysis on shock absorber having different material and applying different loads.
7.	V.B. Bhandari	2014	The above mentioned book gives the best idea about selection of material and the design procedure for various systems
8.	Vinayak Gaikwad, et.al	2020	Provided a review on the components of the braking system and also the calculations for the system.
9.	Vivek Singh Negi, et.al	Jan 2018	Presented a methodology for the design of brake disc, considerations while designing and the logic behind them.
10.	T.Porselvi, et.al	2017	According to the author a BLDC motor was employed as the electric motor. Many resisting forces like rolling, gradient and aerodynamic forces were taken into account in order to determine the power.
11.	M. Koniak et.al	2017	The numerous types of batteries used in electric vehicles are discussed in this study with the specifications of each battery and the simulations which were conducted for the performance of the battery in various conditions.

III. Design Calculations and FEA Analysis

1. Design of chassis:

To design the chassis, loads acting on it need to be assumed. The following are the assumptions made for the loads acting on the chassis:

1. Load due to passenger (considering a heavy male) = W1 = 100 kg
2. Load due to full storage ie. Storage capacity = W_2 = 30 kg
3. Load due to battery packs and motor (safe estimate) = W_3 = 40 kg
4. Miscellaneous load (seat and steering system etc.) = W_4 = 30 kg

Hence total load acting on chassis = W1 + W2 + W3 + W4

Total load = W_T = 200 kg

Considering Factor of safety (FOS) as 1.5

The Design load will be –

W_T * FOS = 200 * 1.5 = 300 kg

300 kg * 9.81 = 2943 N or 2.943 kN

The final material selected was the Aluminium alloy Al 6061 T6. In the context of this project it is to be considered that weight of the chassis will be of utmost importance, as it should be very light in weight so as to allow people with disabilities to manually handle the vehicle. This would not be possible in a Steel

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chassis. Also the selected Aluminium alloy will provide enough strength as the load on the chassis is not very heavy.
The beam was loaded with a UDL
The total load on the chassis is 300 kg ie 2943 N, this will be carried by two such longitudinal beams, hence, one beam will carry 2943/2 = 1471.5 N.
This will be on a span of 48in out of the total span of 72 in as the front overhang will not carry any load. Hence the UDL will be = 1471.5 / 48 = 30.6562 N/in ie. 1.2069 N/mm.

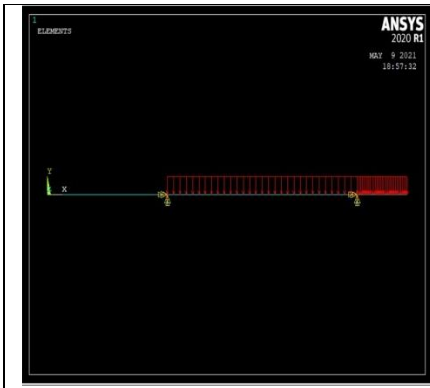

Fig No 2: Simply supported beam with UDL.

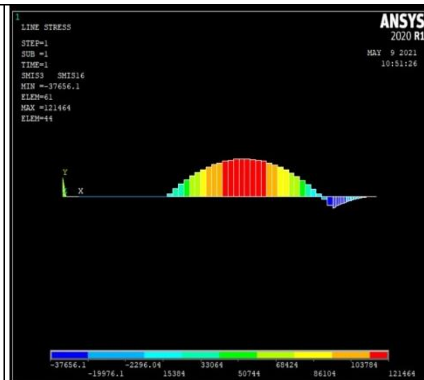

fig no 3: BMD Of the beam.

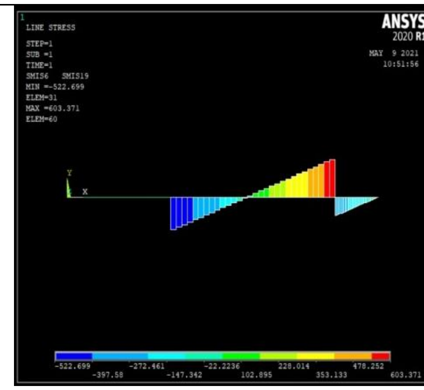

Fig no 4: SFD Of the beam.

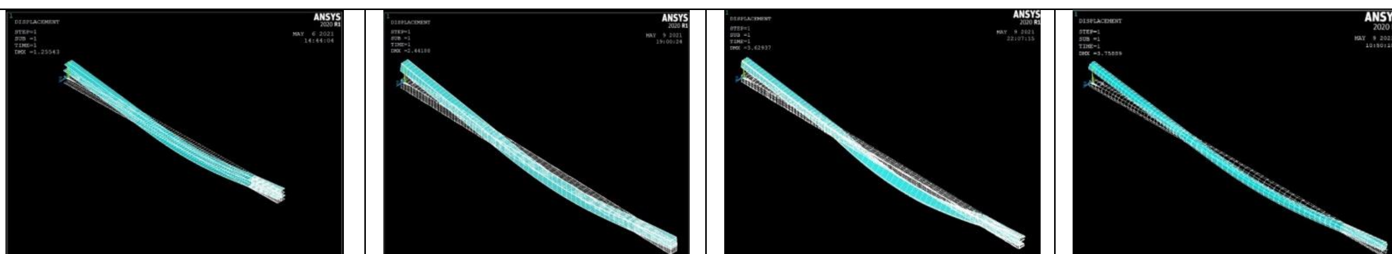

Fig no 5: Deformation analysis of various cross sections

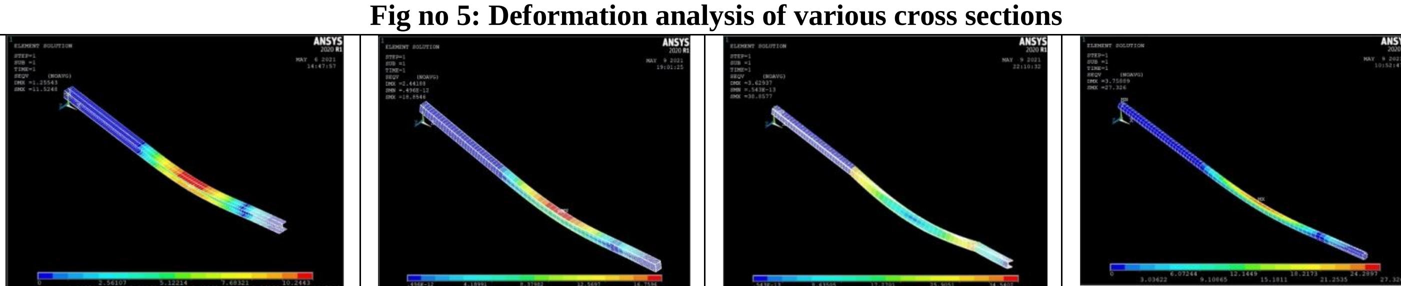

Fig no 6: Von-Mises stress analysis for various cross sections

From the above analysis in the figure no 5 and figure no 6, Square tubular section is selected as it gives the right balance between Availability, Cost, Dimensions, and the Structural properties.

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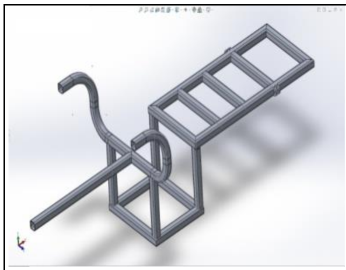

Fig no 7: CAD Model of chassis.

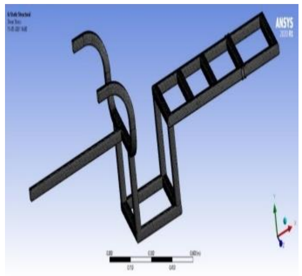

Fig no 8: Meshing of chassis

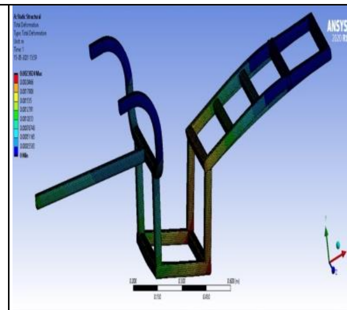

Fig no 9: Deformation analysis of chassis

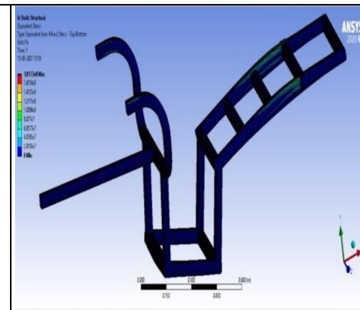

Fig no 10: Von-Mises analysis of chassis

The results show,
The total deformation in the chassis = 0.0023 m i.e.**2.3 mm**
Von mises stress acting on the chassis = **181.73 Mpa**
The Stress in the chassis is less than the yield stress of Al 6061 alloy (276 MPa).
The deformation is 2.3 mm for the span length of 1828.8mm. According to the general rule of allowable deformation in a beam, deformation should be less than (Span length / 300)
Here, 2.3 < (1828.8 / 300) i.e. 2.3 < 6.096 mm which is the allowable deformation.
Hence, from the figure no. 9 and figure no. 10 it is observed that the design of chassis is safe in both Deformation and Stress.

2. Design of steering system:

The steering system used in the mobility vehicle uses a version of Ackermann's steering geometry. The two main conditions in Ackermann steering are:

1. The steering pivot points are moved inwards such that their projections intersect at the centre of the rear axle.
2. The circles prescribed by all four wheels should be concentric.

Assumptions and required data:
Wheelbase (W) = 50 in. Trackwidth (T) = 24 in.
According to a general rule
Inner Wheel angle = $\theta_i < 44°$ and Outer wheel angle = $\theta_o < 30°$, also $\theta_i > \theta_o$.
The condition for correct steering is: $\cot \theta_o - \cot \theta_i = (T/W)$
Front Inner Wheel (FIW):
Here, using trigonometry,
Sin (34) = W / (FIW)
(FIW) = 50 / 0.5591 = **89.4145 in**
Similarly, all turning radius are found.
Front Outer Wheel (FOW):**110.132 in** Midpoint of Rear Axle (MRA) : **86.133 in.**
Comparing the values, the value of **86.133** in falls in between the given limits and hence is acceptable.

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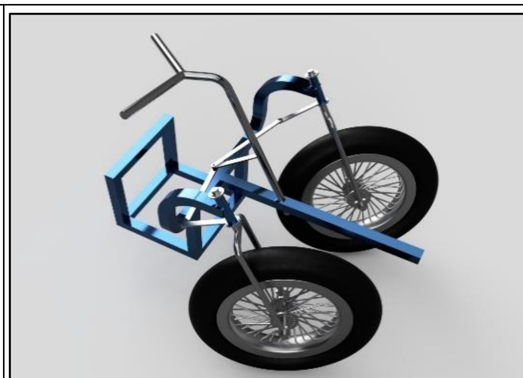

Fig no 11: CAD Model of steering system

3. Design of Suspension System:

The suspension of the vehicle designed in this project is implemented in:

- Front suspension: Two Forks are selected as front suspension.
- Rear suspension: Two coil spring type shock absorbers are used as rear suspension.

As the total weight of the system is 310 kg, a factor of safety of 1.5 is used.
In this configuration, the suspension system has two shock absorbers and two forks. As 70 percent of the weight is acting on the back end, compute 70 percent of 310 kg, divide by 2, then convert this weight in kg to force in Newton.
W= 217 kg ---------------- acting on the shock absorber.
Total force= 2128.77 N
Force acting on single shock absorber is = 1064.385 N.
Oil hardened and tempered wire should be selected as material for coil spring. i.e. ASTM A229.
ASTM A229 have the following specifications:
Constant values (A and exponent m)
A=1855 mpa m=0.187
Carbon – 55% to 85%
E=30*10^3 mpa
G= 81370 mpa
Sut=1400 mpa.

Table no 2: Design calculations of spring [11]

Sr. No.	Parameters calculated	Calculated value
1.	Spring stiffness (k)	k= 42.575 N/mm.
2.	Maximum shear stress acting on spring wire (z)	z=700 N/mm^2
3.	Spring index value (C_i).	C_i=5
4.	Wahl factor (K_w)	K_w= 1.3105
5.	Wire diameter (d)	d= 6mm
6.	Mean coil diameter (D)	D= 30 mm
7.	Number of active coils (N)	N=12 coils
8.	Total number of coils (Nt)	Nt= 14 coils
9.	Solid length of the spring (S.L)	Solid length (S.L) = 84 mm

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10.	The actual deflection of spring (delta actual)	Actual defl^{n} of spring is 26.16 mm
11.	The total gap (Lg)	Total gap (Lg) = 13 mm
12.	free length of spring (Lf)	(Lf) = 122 mm.
13.	Pitch (p) of the coil	p=9.5 mm
14.	The actual spring rate (k)	k=40.685 N/mm
15.	The actual shear stress (zact)	zact= 493.33665 N/mm^2

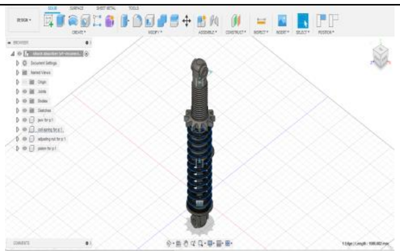

Fig no 12. Shock absorber design

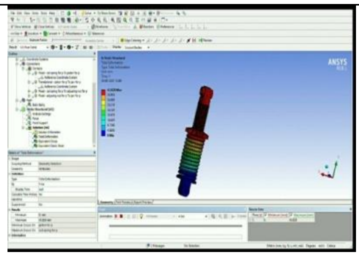

Fig no 13: Total Deformation Analysis of shock absorber

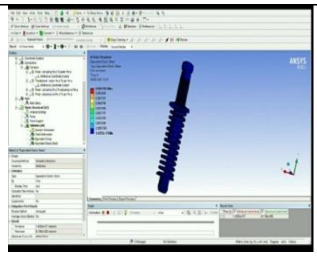

Fig no 14: Von mises strain on shock absorber

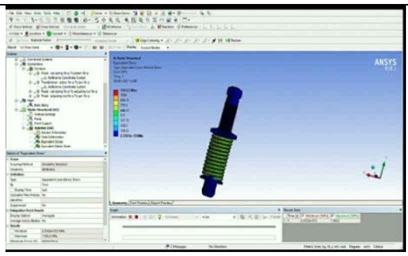

Fig no 15: Equivalent Von mises stres on shock absorber.[10]

Hence, from figure no. 13 and figure no. 15,it is proven that both the actual shear stress and actual spring rate are less than the maximum values. And thus the design is considered to be safe.

Selection of front fork for the vehicle front suspension:

Front suspension is often implemented using a telescopic (i.e., telescoping) fork. As in the problem 70% of weight acting on the rear end of the wheels and shock absorber is already calculated so the remaining 30% load acts on the front forks.

The total weight acting on the front forks is 93 kg**.** So, the weight acting on each fork is 93/2.

Load acting on each spring is 46.5 kg.

The total force acting on the individual fork is 456.165 N. So the mild steel black suspension fork is most suited for the use.

4. Motor and the Powertrain

The motor is the most fundamental component of any basic electric vehicle. Brushless DC motors are commonly employed in these applications and for a decent driving experience.

Power Rating of Electric Motor

The power rating is based on various vehicle dynamic factors like rolling resistance, gradient resistance and aerodynamic drag.[16] The gross weight of the vehicle is considered 310 kg.

The force needed to drive it is,

Ftotal = Frolling + Fgradient + Faerodynamic Where,

Ftotal = Total Force (This is the tractive force that the motor's output must overcome)

Frolling = rolling resistance force

Fgradient = force of resistance to gradients

Faerodynamic = Aerodynamic drag force.

Table no 3: Rolling Resistance and Gradient Resistance.

$F_{Rolling}$	$P_{Rolling}$	$F_{Gradient}$	$P_{Gradient}$
16.72 N	0.1672 KW	1.592 N	0.15923 KW

Aerodynamic Resistance

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The resistance provided by the vehicle's drag is known as aerodynamic resistance. This drag should be as low as possible. But due to the low speed of the vehicle, the drag is very negligible. As a result, $F_{aerodynamic}$ is considered0.As a result, these are the vehicle's primary three forces. Aside from that, the inertia caused by the vehicle's acceleration and deceleration should be taken into account.[16].The power to overcome those forces is also regarded 1 kW when additional random factors are taken into account.

Therefore, total tractive power, P_{total} = 0.1672 + 0.15923 + 1 = 1.32 kW

~ 2 KW. Thus, for this purpose a 2 kW rated power motor was selected.

Table no. 4 lists the specifications of the chosen motor.

Parameter	Value
Rated Torque (Nm)	7.6
Rated Power (kW)	2
Max Power Output (KW)	3
No. of poles	8
Motor Diameter (mm)	145
Rated Voltage (V)	48

Table no 4: Motor parameters

Battery Selection

Based on the required power, a 48 V battery was chosen for the motor.[17] When the vehicle is travelling at full speed, which is 36 kmph, the battery rating is 75 Ah. The battery's other parameters are listed below in table 5. Table no 5. Battery Parameters

Parameters	Description
Nominal Voltage	50.4 V
Charge Voltage	14.4-14.6V
Capacity (Ah)	75
Capacity (W)	900
Chemistry	Lithium Iron Phosphate

5. Braking system:

Braking system is a very important and crucial system of every vehicle as it accounts for the control and safety of the vehicle. Apart from being a mechanism used of slowing down or stopping the vehicle, brakes are very essential for navigation vehicle as per user's intentions. To achieve braking force transmitting member are used which converts the applied force, amplify it and then apply it to generate a torsional moment which is in the opposite direction to the direction of motion of the vehicle.

Assumptions:

1. Diameter of brake disc = 180 mm
2. Diameter of the Brake pad = 12 mm
3. Diameter of Brake line = 1.5 mm
4. Wheel diameter = 24 inch
5. Maximum Speed of the vehicle = 36 kmph

Calculated values:

For Rear side:

Braking Torque = 345.59 Nm

Braking Time = 0.95 sec

Braking Distance = 4.75m

For Front side:

Braking Torque = 345.59 Nm

Braking Time = 0.409 sec

Braking Distance = 2.045m

IV. Analysis of Results

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Table no:6 Graph for chassis material selection

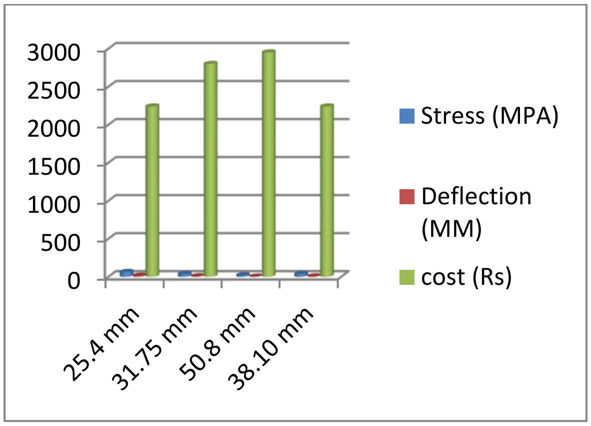

Table no 7: Result table of steering system

Turning radius at points	Turning radius in (mm)
Front inner wheel (FIW)	2271.1283
Front outer wheel (FOW)	2797.3528
Midpoint of Rear axle (MRA)	2187.2702

Table no 8: Result table of suspension system

	Permissible values	Actual Design values
Shear stress (z)	**700 MPA**	**493.34 MPA**
Spring stiffness or spring rate (k)	**42.575 N/mm**	**40.685 N/mm**

Table no 9: Result For Braking System Considering Rear Side and front side of the Vehicle.

Parameters	Calculated Values Obtained (Rear Side)	Calculated Values Obtained (front Side)
Braking Torque	345.59 Nm	345.59 Nm
Braking Time	0.95 sec	0.409 sec
Braking Distance	4.75 m	2.045 m

Table no 10: Parameters and values of Motor and Battery

Parameters	Value	Parameters	Values
Power required to overcome resisting forces	598 W	**Rated Voltage**	**12 V (of one battery)**
Rated Power	750 W	**Capacity**	**26 Ah**
Rated Voltage	48 V	**Time till which vehicle can run on full load**	**99 mins**

V. Conclusion and Future scope

Conclusion:

1) For material selection of chassis, the frequently used aluminium alloys were considered, the selection of final material was done by comparing properties like density, Young's modulus, ultimate tensile strength, weldability, corrosion resistance, weight, machinability and cost. The final material selected was Aluminium Alloy Al6061 T6.

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2) Selection of the cross-section of chassis was done considering the availability, cost, and analysing the deformation and stresses acting on the cross-section. From the analysis, the ideal cross-section for chassis was square cross-section.
3) The Steering System used in the mobility vehicle uses a version of Ackermann's Steering Geometry. From the calculations, the conditions for Ackermann Steering Mechanism were met. Thus, from the constructed geometry, correct Steering is achieved and turning radius is within sought limits.
4) For Suspension System, by considering properties like load, strength, stiffness, Modulus of rigidity, ultimate tensile strength, availability and cost the Oil Hardened and Tempered wire i.e. ASTM A229 was selected for coil spring and Mild Steel was selected for Suspension Fork.
5) The Motor selected was BLDC motor because of its properties like light weight, less complexity, high performance, less no. of moving parts, more resistant to heat and based on the power requirements to overcome all the resisting forces.
6) Based on the required power, a 48 V battery was chosen for the motor. When the vehicle is travelling at full speed, which is 36 kmph, the battery rating is 75 Ah, and parameters like nominal voltage was 50.4V, charge voltage was between 14.4 to 14.6 V, and capacity was 75Ah i.e. 900W.

Future Scope:

1) A roof can be included in the vehicle, in order to increase the comfort of the user as well as to protect from harsh weather conditions.
2) Also, the roof can be used to mount the solar panels which could provide an alternative power source to the vehicle.
3) Additional accessories such as headlights, taillights, mobile holder can be added in order to improve the ease of use.
4) The current stationary seat can be replaced by a rotary seat which will increase the accessibility of the vehicle.

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