IOSR Journal of Mechanical and Civil Engineering (IOSR-JMCE)
e-ISSN: 2278-1684,p-ISSN: 2320-334X, Volume 18, Issue 3 Ser. I (May – June 2021), PP 16-29
www.iosrjournals.org

Design and Analysis of Electric Wheelchair cum Stretcher

Siddhant Pawar[1], Yogesh Kanade[2], Hemant Singh[3]
[1]*(Mechanical Department, MCT Rajiv Gandhi Institute of Technology, Mumbai University, India)*
[2]*(Mechanical Department, MCT Rajiv Gandhi Institute of Technology, Mumbai University, India)*
[3]*(Mechanical Department, MCT Rajiv Gandhi Institute of Technology, Mumbai University, India)*

Abstract: *As electric wheelchairs are on a raising trend, there is plenty of scope in achieving a strong, lightweight and cost-effective wheelchair. This project illustrates some elegant engineering innovations in the chassis, motor and powertrain system that can help us achieve the goal. Furthermore, the project describes how the wheelchair can be converted into a stretcher and be used for transportation of patients and goods in hospitals or industries. As we proceed, we will see the designs, calculations and analysis for all the mechanical components such as power transmission, chassis and stretcher mechanism.*

Key Word: *Wheelchair, electric, stretcher, hospital environment, chain drive.*

Date of Submission: 08-05-2021 Date of Acceptance: 23-05-2021

I. Introduction

As per Census 2011, about 2.68 Cr persons are 'disabled' out of the 121 Cr Indian population, which is about 2.21% of the total population. The mobility of physically challenged people becomes a serious concern. In addition, not all families can afford to buy the expensive power wheelchairs available in the market. Mainly the individuals with neurodegenerative disease such as MND (Motor Neuron Disease), ALS (Amyotrophic Lateral Sclerosis) etc. needs a guardian to physically lift the patient every time the patient needs to move from wheelchair to his bed or vice a versa. This process takes a lot of physical tolls on the guardian, according to a study about 40% of guardians are suffering from back pain and joint pain. To solve this problem many researchers and mechanical engineers have developed various designs where the wheelchair can itself be able to be converted into a stretcher, some of which are using mechanisms of hydraulic and pneumatic lift, while others are using a lead-screw mechanism. These mechanisms are very efficient but the drawbacks are that it either requires servo motors which increases the manufacturing costs or are a noisy. To overcome this problem, we have proposed a new system which is purely mechanical based for the stretcher conversion. Having a purely mechanical mechanism helps in lowering the cost and at the same time to achieve a simple mechanism with less moving parts which shall help in reducing noise.

Also, a lot of electric wheelchairs available in the market are very expensive for a middle-class person to afford, thus to overcome this problem our objective is to choose a low power dc motor in order to lower the cost and compensate for its power by designing an efficient power transmission system.

II. Literature Review

DR. Sukanta [1], discusses the need of a wheelchair cum stretcher mechanism. This report has discussed the conversion by implementing a lead screw mechanism. The drawback of this system is the cost and the increased overall weight of the wheelchair.

Prof. Nikhil V. Bhende [2], discusses the design and fabrication of a lever propelled wheelchair. Although it reduces human effort considerably, individuals with neurodegenerative disease such as MND (Motor Neuron Disease), ALS (Amyotrophic Lateral Sclerosis) etc. will not benefit from this wheelchair.

Thomas Paul [3], describes the design for his approach in the wheelchair cum stretcher mechanism with lead screw connected with a hinge joint. The Lead Screw translates turning motion into linear motion, this provide an advantage that the wheelchair can be kept at any angle required for the comfort of the patient. A hydraulic jack is also added to have an adjustable height of the stretcher.

P. Swapna [4] in her report has discussed her approach to create the stretcher mechanism, the stretcher mechanism is achieved by using a scissor jack. The scissor jacks are advantageous in lifting heavy load, in this report the jack is capable of lifting load upto 1000kg. A 12V, 10 Amps motor is used to expand the jack. This is a great approach but the drawback is the weight of the jack and high cost of the overall design.

Mohan Kumar R. [5] has discussed a design of a multipurpose wheelchair, the market study of existing competitors and their cost analysis is done. Also, an ethnography study is done to address the design gap and importance of the existing product. The final output of their designed wheelchair has multiple options such as ease of defecation, cleaning and changing of clothes. The design also incorporates adjustable backrest, armrest and leg rest.

A design manual written by Amos Winter [6] has explained various mechanical principles in a wheelchair design. This report talks about the forces, center of gravity, free body diagram, moments, inertial forces, axial stress, behavior of metals, strain modulus, shear stress, bending stress, moment of inertia, (stiffness vs strength), failures and other important design parameters in a wheelchair.

Xin Chen [7], in his report has given a detailed study on modeling and design of a mechanism which is capable of having a sliding seat which is helpful for patients for moving themselves from the wheelchair to any other seat close by. Further this report discusses the static and dynamic analysis of the chair.

Parag Nikam [8], the report explains the tooth force calculation and its static structural analysis using Ansys workbench. Though the calculations were perfect, the simulation is not precise enough as boundary condition involves fixed constraints at bolt holes.

III. Proposed System

3.1 Motor:

Selecting an appropriate motor is extremely important as it is the powerhouse of the entire system. Considering its working environment, following would be the suitable parameters:

Since the wheelchair is designed to run in an indoor environment, the velocity can vary from 0.5m/s to 1.5m/s. We have considered the higher velocity of 1.5m/s. Also, the standard slope in Indian hospital environment is of 7.1˚.
The total mass of the wheelchair = 145Kg. (self-weight = 45kg, Load carrying capacity = 100kg)
Radius of the wheel = 10 inches = 0.254m based on availability and cost.

Calculation of RPM of wheel:
The circumference of the wheel is the linear distance that will be covered in one revolution.
$2\pi r = 1.59m$
1RPS covers 1.59 meter in 1 second
Therefore, for 1.5meter in 1 second,
RPS = 0.94
i.e., 56.39RPM ≈ 57RPM

Case 1: Travelling on flat path:

Torque = (Push Force + Rolling resistance) * radius of wheel
Push Force is the minimum amount of force needed to start the motion of any automobile. It is given by the formula "m*a"
Rolling resistance is the resistance offered by the tire due to its Visco-elasticity. It is given by the formula "umgcos α"
where u = coefficient of rolling resistance = 0.01

Therefore, Torque = [(m*a) + (umgcos α)] *r
= (43.5+14.22) *0.254 = 14.66 Nm
Consider Factor of Safety (FOS) = 1.5, Torque = 21.99Nm
i.e 11Nm at each wheel

Calculating power required at one wheel:
Power = Torque* angular velocity (v/r) = 11 * (1.5/0.254)
= 64.9 Watts

Case 2: Inclined path

While travelling on an inclined path, the wheelchair will experience some Gradient Resistance. The Free body diagram for this condition can be illustrated in Fig. 1
Gradient Resistance is the resistance offered on slope due to gravity. It is given by the formula "mg sin α"
Considering that the gradient resistance will tend to oppose the motion, the velocity at slope decreases and the

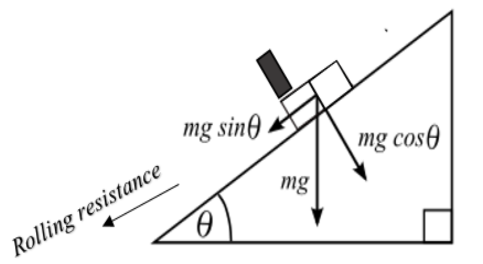

Figure 1

acceleration is almost zero. Therefore, push force becomes negligible
Therefore:
Torque = (Rolling Resistance + Gradient Resistance) * Radius
= [(umgcos α) + (mgsin α)] *r

= [(0.01*145*9.81*cos7.1) + (145*9.81*Sin7.1)] *0.254
Torque = 48.24Nm

Considering FOS = 1.5; Torque = 72.36Nm
i.e., 36.18Nm at one wheel

Power at one wheel = Torque * (v/r) = 36.18* (5.905)
= 213.64 Watts
Since Case 2 requires more power, therefore designing for Sloped Path. Hence, selecting DC motor MY1016Z2 that has a power capacity of 250W, 24 V and can run at 360 rpm. The maximum torque that can be achieved is 22 Nm. This motor can be powered by a 24 V Lithium-ion battery.

Figure 2

3.2 Transmission:
Purchasing a high torque motor would be heavy and extremely expensive. Thus, we have incorporated a transmission system which can multiply the torque to carry the design load.
We select chain drive over gear box to drive the wheelchair wheel which is powered by 250W motor due to following reasons:

- The chains drive has the center distance between input and output shafts is less
- Less slippage
- Velocity ratio can be achieved accurately
- Sprockets usually have low weight.
- Less maintenance and cost effective
- High efficiency

Further the detailed design of various parts of transmission is discussed.

Design of Chain Sprocket:

Material Selection:
Based on availability and applications we have selected C-45 (Mild steel) as Sprocket Material.
As per the standard Torque vs Rpm curve [8] (as shown in Fig. 3) of a DC motor, the maximum power of the motor is obtained at nearly half the rated RPM. It indicates that the motor has the maximum efficiency at this RPM.
The efficiency of the motor is important as it can lead to major heating issues, battery drainage and insufficient power.
Since the selected motor has a rated RPM of 360, we plan to run it at 180 RPM to obtain maximum power and efficiency.

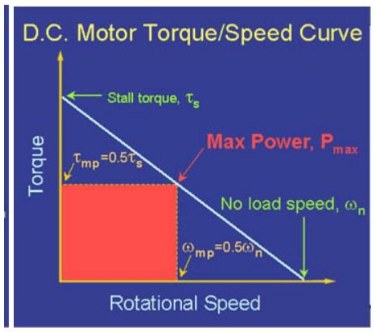

Figure 3

Therefore,
N1 = 180 rpm, No of Teeth = Z1 = 9 (Driver pinion)
N2= 57 rpm (Driven sprocket)
Therefore, Reduction ratio = N1/N2 = 180/57 = 3.157
Z2 = Z1*3.15 = 28.35 ≈ 29 Teeth
Using formula, D = p / sin(180/z) --------(Pitch – 12.7 mm)
We get D1= 37.13 mm, D2 = 117.463mm
We know that, chain length is pitch times the no of links, i.e., L = l_p *p
No of links is given by; $l_p = 2\ a_p + (Z1 + Z2)/2 + ((Z2 – Z1)/2*pi)^2 / a_p$
L = 635 mm --------- (Calculated Chain length for a_p = 15)
Actual Centre distance is calculated as
a= (e + sq root (e^2-8*m)/4 * p = 196.26 mm

Dynamic Calculation:
To check whether the design sprocket is safe or not in dynamic condition:
Assume Power = 250 W = 0.25kW
For dynamic conditions, we need to calculate first Service Factor (Ks) and it is given by
Service factor, Ks= k1 * k2 * k3 * k4 * k5 * k6

From PSG Pg. No – 7.76, the constant is shown in Fig. 4
Therefore,
Ks = 1*1.25*1.25*1*1.5*1
= 2.343
Chain velocity = pi*D1*N1/60 = 0.349 m/s
Breaking load (Q):
We know that, as P = $\frac{Q*v}{102*n*ks}$
Q = 1206.91 kgf
Going for safer side we select 08B-1 Roller chain with following specifications:
08B-1 with Breaking load = 1820 kgf, Area = 0.5 cm^2

- Load factor as Constant load, K1 = 1
- Factor of distance regulation for Fixed distance, K2=1.25
- As we consider before ap = 15 which is ap < 25
- Therefore, Factor for center distance between sprocket, K3= 1.25
- For horizon ıl position of the sprocket, K4= 1
- Lubrication ıactor for periodic lubrication K5= 1.5
- Operating as Single sift 8hrs/day, rating factor K6 = 1|

Figure 4

Table no 1

Chain No ISO/DIN	Rolon	Pitch (mm)	Roller Dia (mm)	Bearing Area (cm^2)	Wt./m (kgf)	Breaking Load (kgf)
08-B1	R1278	12.7	8.51	0.5	0.7	1820

Sprocket Dynamic Analysis:

Teeth Force calculations & analysis for sprockets.
Following is the formula to calculate force acting on each tooth of a sprocket
$T_k = T_0 \times \{\sin ø \div \sin (ø + 2\beta)\}^{k-1}$
Where:
T_k = back tension induced in tooth k, T_0 = chain tension, ø = minimum pressure angle = (17 – 64)/N, N = No. of teeth, 2β = sprocket tooth angle (360/N)
k = the number of teeth engaged with chain (i.e., angle of wrap × N/360); rounding up to the nearest whole number.
For Driver Sprocket:
As per Dynamic calculation, Chain tension = 786N
Therefore T_0 = 724.88N
Ø = 17 – 64/9 = 9.88; N = 9; 2 β = 360/9 = 40
K = 200*9/360 = 5 Similarly for Driven Sprocket is given in Table 3

Table No 2

Teeth No.	Force (N)
1	724.88
2	162.59
3	36.469
4	8.18
5	1.834

Table No 3

Teeth No	Force (N)
1	724.8
2	404.77
3	226.02
4	126.21
5	70.47
6	39.35
7	21.975
8	12.27
9	6.85
10	3.82
11	2.136

Now, Performing FEA Simulation in Ansys Workbench. As boundary condition, giving fixed constraint to the holes and applying calculated tooth loads. The solution for driver and driven is shown in Fig 5 and 6 respectively. Also the corresponding values are tabulated in table no. 4.

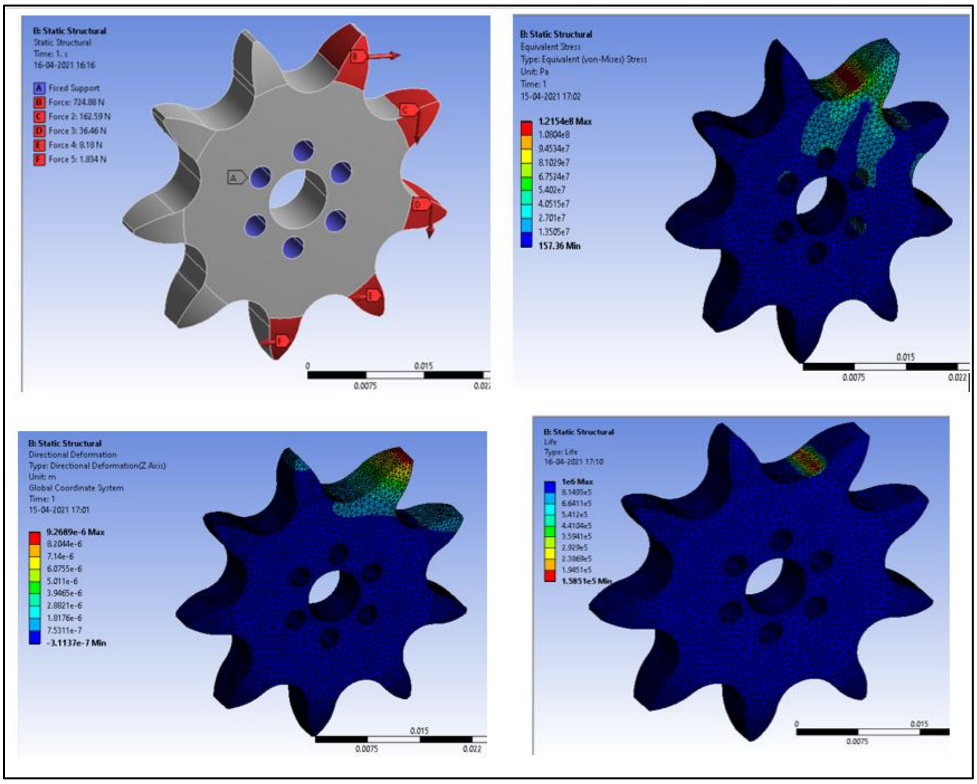

Figure 5

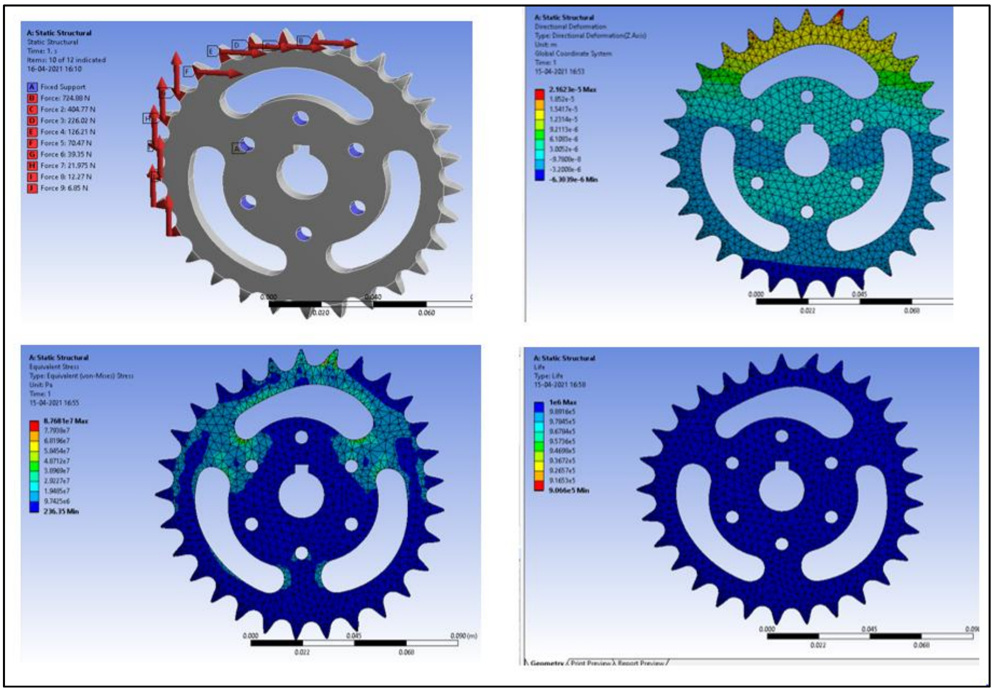

Figure 6

Table No 4

Type	Minimum	Maximum
Equivalent Stress (Driver)	157.36 Pa	1.2154e+008 Pa
Deformation (Driver)	3.1137e-007 m	9.2689e-006 m
Life (Driver)	1.5851e+005 cycles	
Deformation (Driven)	6.3039e-006 m	2.1623e-005 m
Equivalent Stress (Driven)	236.35 Pa	8.7681e+007 Pa
Life (Driven)	9.066e+005 cycles	

Design of Shaft:

Shaft is rotating machine element, sometimes stationary, which is used as support element. Here, shaft is used to drive the chain sprocket fixed to it. Therefore, the only weight acting on it is the sprocket, the shaft is supported at both the ends with roller support. As it as to transmit torque too, the shaft will undergo both bending and torsion.

Designing the shaft for Bending and torsional.

Given: Power = 250W = 0.25kW, 57 rpm, Shaft length = 65mm = 0.065m

Material selection:

We select shaft material as C-45 due to its strength and applications with Shear stress strength = 450 MPa = 450 N/mm^2

Permissible shear stress strength = $\tau = \frac{450}{10}$ = 45 N/mm^2 (For FOS = 10)

We know that, Torque = $\frac{P*60}{2\pi N}$ = 41.88 N-m

T = 41.88 * 10^3 N-mm

Maximum Bending Moment:

We know that,

M = $\frac{W*L}{4} = \frac{0.400*9.81*0.065}{4}$ = 63.765 N-mm

Now, Equivalent Twisting moment,

$T_e = \sqrt{M^2 + T^2}$

T_e = 41880.048 = N-mm

Also, $T_e = \frac{\pi*\tau*d^3}{16}$

Therefore, $D^3 = \frac{41880.04*16}{\pi*\tau}$

D = 16.797 mm

D = 20mm …………………..(Referring to standard available diameter of shafts)

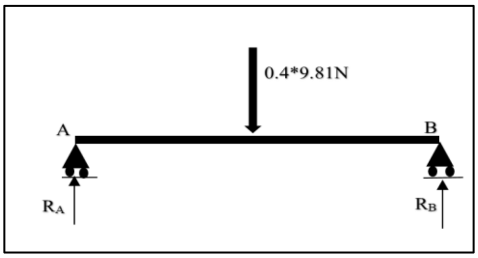

Figure 7 Shaft FBD

Analysis of Shaft:

We have performed static analysis of hub shaft in Ansys simulation software to check whether the analytically calculated dimension is safe or not in terms of structural, factor of safety and life of Shaft by fixing Support at one end. Applying bearing load as a 3.9 N and torque as 41.8 Nm. The solution for the shaft analysis is shown in Fig. 5. Also the corresponding values are tabulated in table no. 5

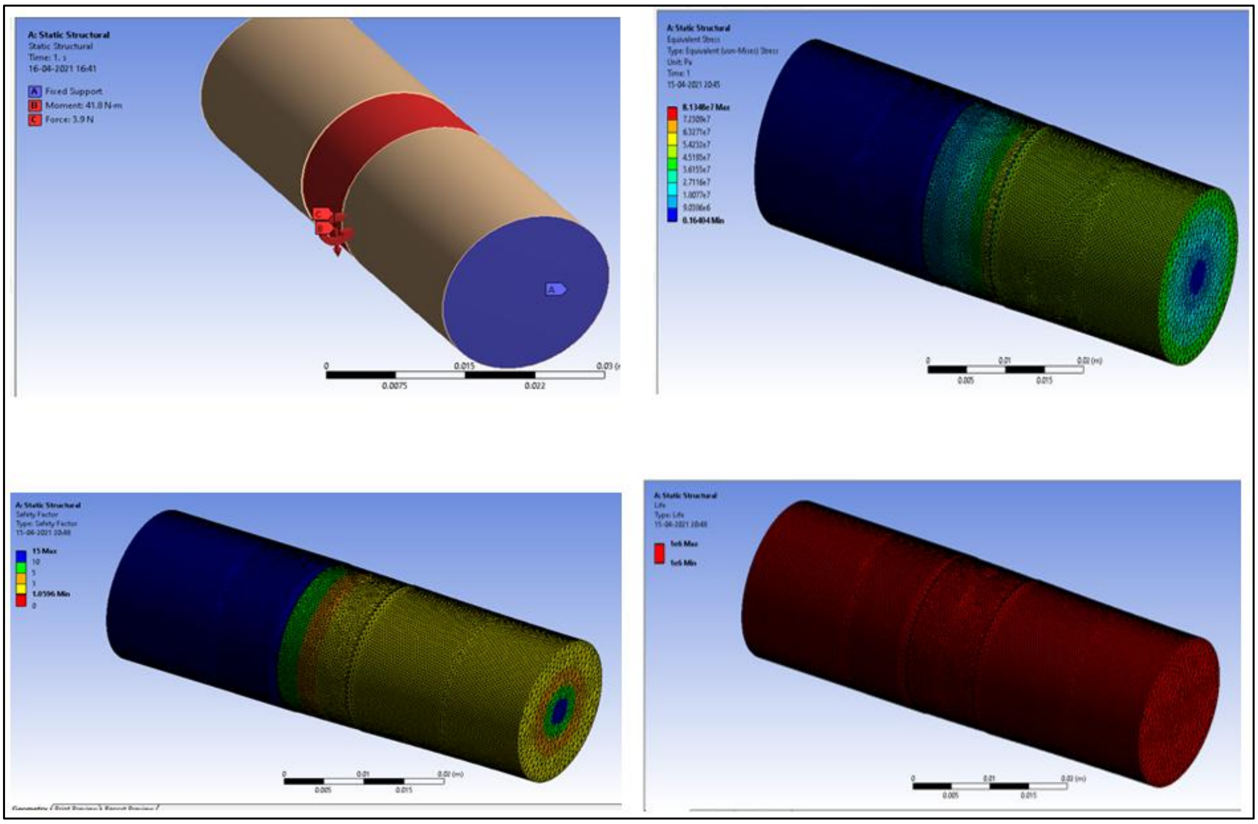

Figure 8 Shaft Analysis

Table No 5

Type	Minimum	Maximum
Equivalent Stress	0.16404 Pa	8.1348e+007 Pa
FOS	1.05	15
Life	-	10E6

Design of Key

A key is a piece of MS used to prevent relative motion between shaft and hub. It is inserted half in shaft and other half in hub. Keys are also subjected to crushing and shearing stresses.

Material selection:

We select key material same as shaft which has $\tau = 45N/mm^2$ for FOS = 10

We know that, Shaft diameter = 20mm; $\tau = 45N/mm^2$

Torque = $\frac{\pi*\tau*d^3}{16}$ = 70685.83 Nmm

For shaft diameter = d = 20mm; Width of key = 8mm; Thickness = 7 mm

l= length of key

Considering key for shearing failure,

70685.83 = l*w* τ* $\frac{d}{2}$ Therefore, l=19.63 mm

Considering key for crushing

Taking FOS = 5 for crushing stress

70685.83 = l*$\frac{t}{2}$*σc*$\frac{d}{2}$

L = 29.699 mm ≈ 30 mm

Therefore, the selected key dimension is L = 30 mm, W = 8mm T = 7mm

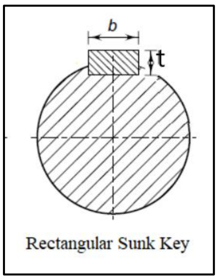

Figure 9 Shaft and Key

3.3 Chassis:

The main component which defines the quality, life and cost of any mechanical system or instrument is the chassis. Chassis is the basic skeleton structure of any vehicle and bears all the stresses, both static and dynamic, therefore the selection of material and the geometric design is one of the most crucial part in the designing process. The primary purpose of this body is to assemble all the components into a single unit and maintain its stiffness.

Material Selection:

The Material we have chosen for the chassis is AISI 1018 hot rolled steel square pipes with 25.4mm OD and thickness of 1. 6mm.The other commonly available thickness in the market are 1.2mm and 2mm, the former was not chosen as it has very little thickness which can be a problem during the welding process and the later was not chosen as it increased the overall weight of the chassis.

Another option was aluminum, though it is lighter in weight but it has very poor weldability and has high market cost. Therefore, though steel is heavier it is chosen for its good strength, welding capability, cheaper cost and market availability. The properties of the selected material are shown in table (6).

Table No 6

Material Properties	
Name:	AISI 1018 Steel, hot rolled bar
Yield strength:	1.8e+08 N/m^2
Tensile strength:	3.25e+08 N/m^2
Elastic modulus:	2e+11 N/m^2
Poisson's ratio:	0.29
Mass density:	7,870 kg/m^3
Shear modulus:	8e+10 N/m^2
Thermal expansion coefficient:	1.2e-05 /Kelvin

Chassis Geometric Design:

The basic dimensions of the wheelchair are designed close to the standardized dimensions used for most of the market available wheelchair, Fig. 10 shows the dimensions of standard wheelchairs [10]:

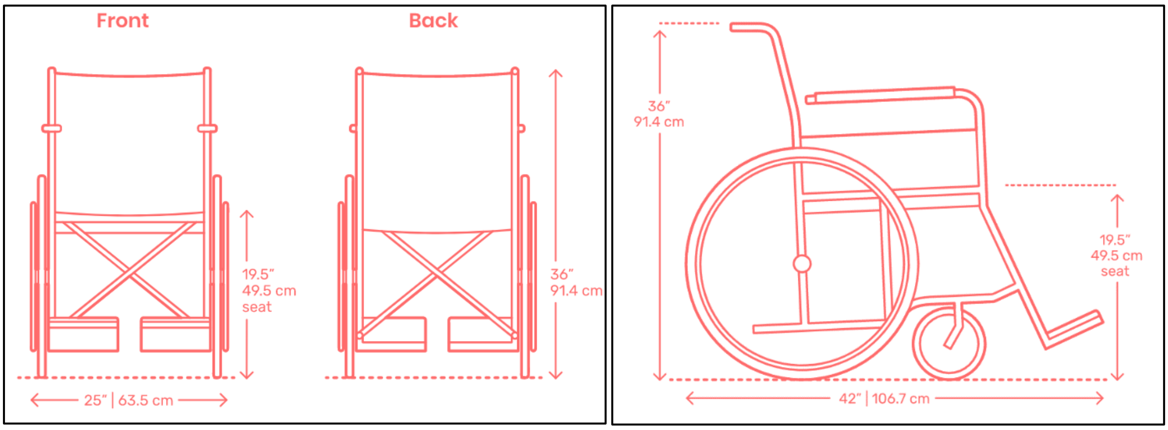

Figure 10

Since the wheelchair consists of a stretcher convertible mechanism, chain-sprocket system, DC motor, motor driver and a battery, the chassis is designed in order to achieve the packaging of all the mechanical parts and the electrical components. The cad model of the chassis with its packaging of the components can be seen in Fig. 8

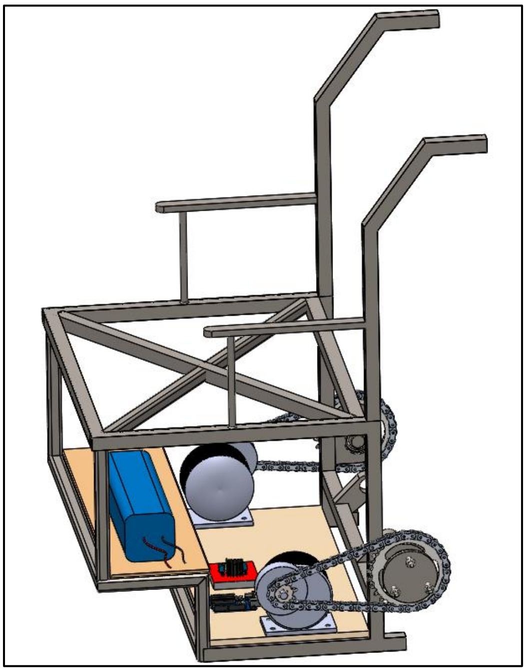

Figure 11

Chassis Design Calculation:

Chassis rigidity is an extremely important parameter while designing as if the system cannot handle the load it can lead to breakdown of the entire system. Hence it is crucial to assess the beam stiffness and torsional stiffness of the chassis. For that, we calculated the bending and compressive stresses:

Calculation for compression stress:

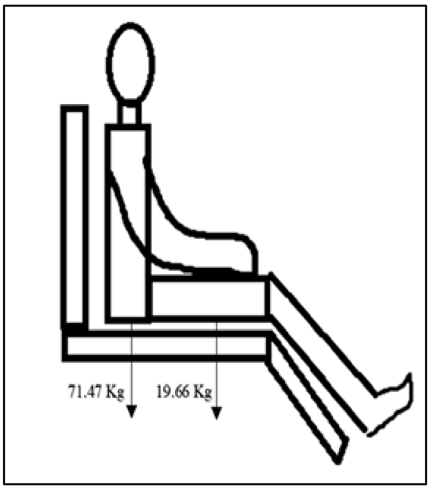

Figure 12

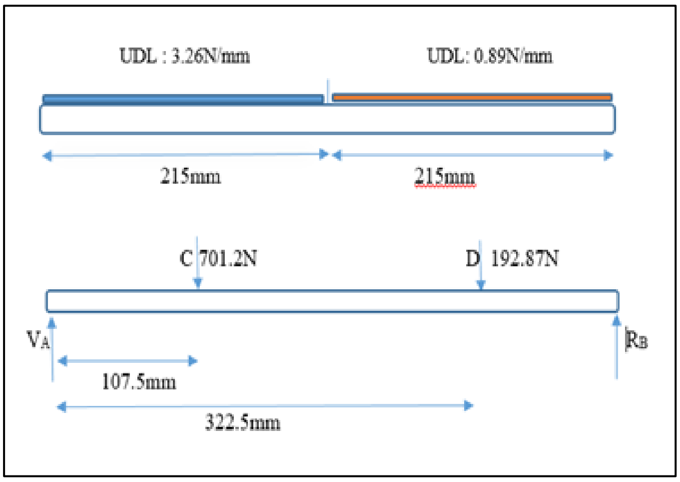

Figure 13

As per the standard body weight distribution, Upper body weight accounts for roughly 68.35%
Whereas thighs weigh nearly 18.8%. In sitting position, consider the upper body weight to act at the back half of the seat. The weight distribution according to it can be seen in Fig-12. This weight distribution is converted to uniformly distributed load as shown in the free body diagram of Fig-13.
Let V_A, R_B be the reaction forces acting on the Rear and the front member respectively. Assume weight of 104.5 Kg (according to average obese patient).

Applying Conditions of Equilibrium:
Sum of Forces along vertical axis should be zero
$V_A + R_B = 701.2 + 192.87 = 894.07N$...........(equation 1)

Sum of Moments about Point A = 0
-(701.2*107.5) -(192.87*322.5) +(R_B*430) = 0
R_B =319.95N
Therefore, V_A = 574.12N

Adding self-weight of the seat;
5.2Kg = 51.012N
Therefore,
Load on back two vertical members: 599.626N
Load on back two vertical members: 345.456N
Load on Each vertical member at the back = 299.8N
Load on Each Vertical member in the front = 172.72N.

Bending Moment Calculation:
From fig-D,
Bending moment at V_A =0
At V_C = V_A*107.5 = 61717.63.........(equation 2)
At V_D = (V_A*322.5) -(701.2*215) = 34395.7
At VB = 0

Therefore, maximum bending moment is at point C = 61717.63Nm
Calculation for Bending Stresses:

The bending stress is calculated for the pipes selected for the chassis refer Fig-14,
Total Pipe length = 430mm, Outer dia = 25.4mm, Inner dia = 22.2mm
Sectional Modulus of Hollow Square:
$Z = (BD^3-bd^3)/6D$
For Square Section, B = D
$= (25.4^4 - 22.2^4)/ (6*25.4)$ =173340.16/152.4 b =1137.4026
From equation two we know that ending moment is at point C = 61717.63Nm
Therefore, Bending Stress = M/Z = 54.26N/mm^2

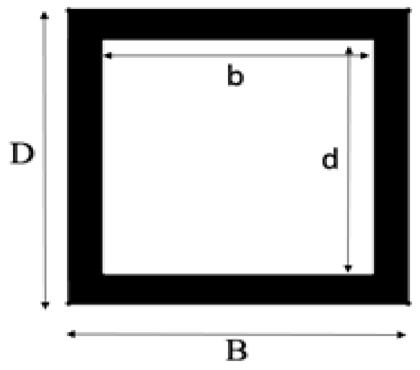

Figure 14

Chassis Structural Analysis:
Study boundary conditions:

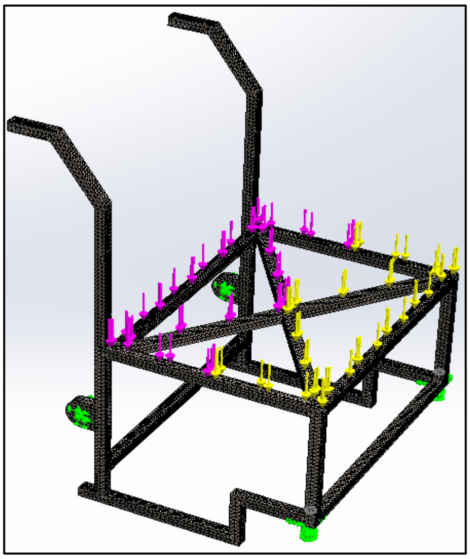

Figure 15

The structural analysis of the chassis is performed in solidworks-simulation study. In this study, four 'Fixed geometry type' fixtures are provided as shown in green in Fig-15, where two are at the front end of the circular section at the position where the castor wheels are to be mounted and the other two fixtures are at the rear end where the axle bearing of the rear wheels will be mounted. The loading conditions is applied for an assumption of a person weighing 120 kg. The forces are applied as per the calculation for the standard body weight distribution as discussed and calculated in previous section, in fig-F the load represented in pink have the value of 701.2 N and the load represented in yellow have the value of 192.87 N. The mesh as seen in Fig-15 used is of standard mesh of 'solid mesh type 'keeping the mesh quality plot as high having Element Size: 10.5145mm and Tolerance: 0.525735 mm.

Study results:
On applying all the necessary boundary conditions as discussed before the results are obtained, the following figures show the plot of Von Mises Stress (Fig-16), Equivalent Strain (Fig-17), Resultant Displacement (Fig-18) and Factor of Safety (Fig-19). The results are shown in the table-7.

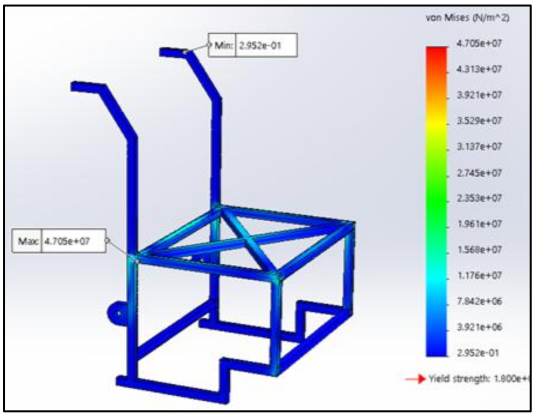

Figure 16: Von Mises Stress

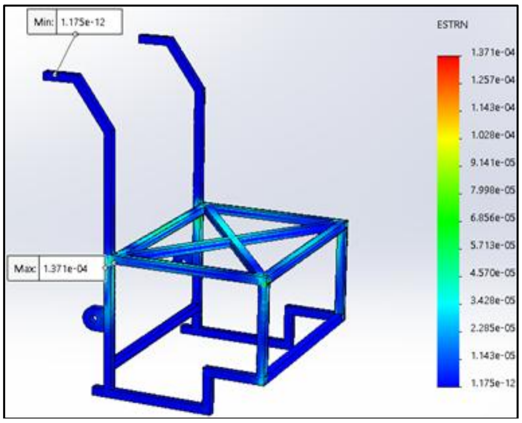

Figure 17: Equivalent Strain

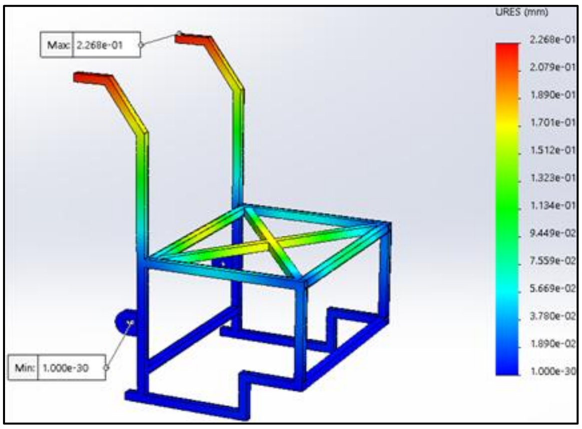

Figure 18: Resultant Displacement

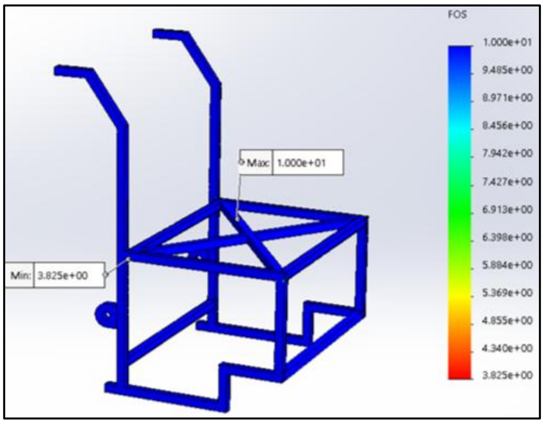

Figure 19: Factor of Safety

Table No 7

TYPE	MINIMUM	MAXIMUM
von Mises Stress	2.952e-01 N/m^2	4.705e+07 N/m^2
ESTRN: Equivalent Strain	1.175e-12	1.371e-04
URES: Resultant Displacement	1.000e-30 mm	2.268e-01 mm
Factor of Safety	3.825e+00	1.000e+01

Hence, from the analysis it is observed that the structure of the chassis is safe.

3.4 Stretcher Mechanism:
A stretcher mechanism is designed for the wheelchair which is sub-divided into 'Back-Rest conversion system' and 'Leg-Rest Conversion system '. Converting the chair to a stretcher will not only be useful in carrying patients but also creates more space for transportation of various types of medical equipment.
The conversion of the wheelchair into a stretcher is a two-stage process: -

- Back-Rest Conversion: -

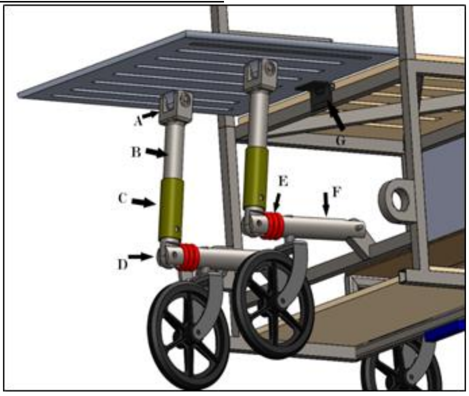

Figure 20

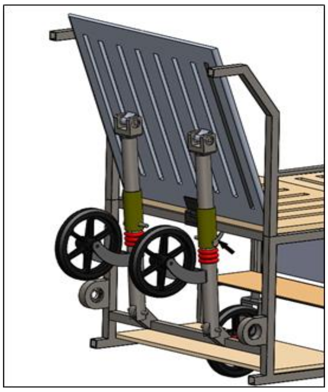

Figure 21

1. The back-rest conversion mechanism works mainly with a pair of two link mechanism. The pair of links are represented as (‘B and F’ in fig 20).
2. The seat is mounted to the chassis with a foldable hinge, as seen in (‘G’ fig 20).
3. The upper link (‘B’ in fig 20) is connected to the seat by a mount welded on the seat and socket structure at the end of the link (‘A’ in fig 20). This is fixed using two bearings and a stud, thus providing rotational motion to the upper link.
4. The lower link (‘F’ in fig 20) is connected to the mount in lower member of chassis, this connection is done by a bush insert of a larger diameter and a stud which provides in rotational motion of the lower link too.
5. The upper and lower links are joined together (‘D’ in fig 20) using a similar bush and stud mechanism as used above.
6. Hollow sleeves (‘C’ in fig 20) slide over the upper link, the sleeve acts as the main component to lock or unlock the links using a key. In Fig. 21, the wheelchair is in stretcher position where the links are bent in almost 90-degree angle until the castor wheels attached to the lower link meets the ground and acts as a support.
7. In Fig-21 the wheelchair is in sitting position, here the upper and lower pair of links are aligned in a straight line and are locked together using the sleeves and keys (shown by an arrow in Fig-20).
8. The key material is selected to have lower FOS than the sleeve and links, as in situation of failure the key should break instead of the primary parts. Rubber Grommets (‘E’ in Fig 20) are attached to the lower link using adhesives. The grommets will act as a stopper for the sleeve in case of failure of the key, thus acting as a safety mechanism.

- Leg-Rest Conversion: -

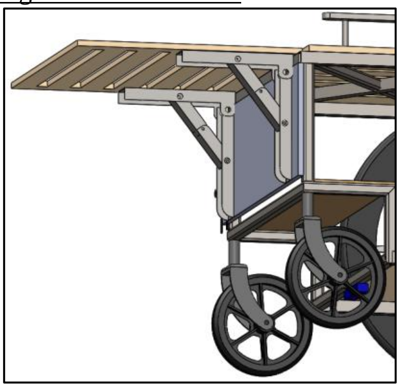

Figure 22

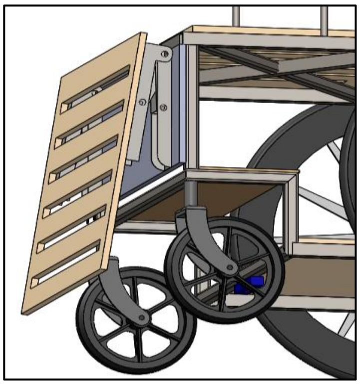

Figure 23

The conversion mechanism for the leg rest is achieved using a foldable hinge bracket, where one part of the hinge bracket is attached to the leg-rest and the other part is attached to the plate on the chassis. The Fig-22 and Fig-23 represent the hinge in sitting position and stretcher position. Such adjustable hinges are readily available in the market, are cheap in cost and, at the same time it helps in making the mechanism simple and less complicated. It also helped with providing compactness of the chassis and achieving sufficient amount of space for packaging of the battery, motors and other hardware.
The cad model for the wheelchair in stretcher and sitting position is shown in Fig-24 and Fig-25: -

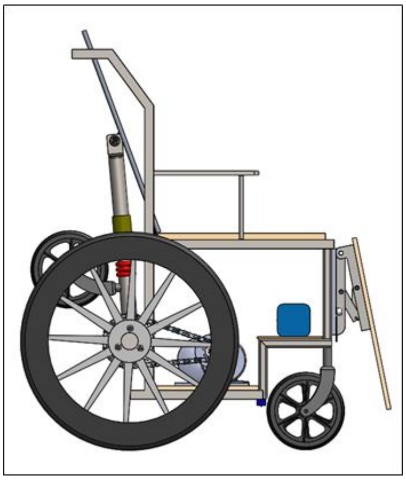

Figure 24

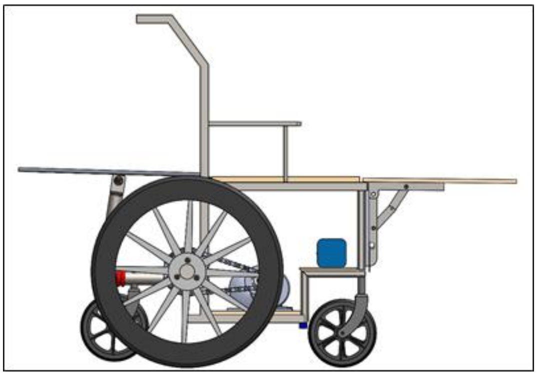

Figure 25

IV. Cost Estimation

The estimated cost of the actual designed wheelchair is nearly Rs 20,540. The cost of wheelchairs available in Indian market have a starting range of Rs 40,000 and go up to Rs 1,00, 000.Comparing with the starting range our design has achieved a price reduction of about 48.65%. The cost of each subsystem is given in the table below:

Table No. 8 Bill of Material

Department	**Component Name**	**rice**
Electric Hardware	Micro-controller	00
	Motor Driver	000
Power Source	Motor- MY1016Z2 (Qty = 2)	400
	Battery- 24V	000
Transmission	Sprocket	60
	Chain	80
	Shaft and key	00
Chassis (Stretcher Convertible)	MS AISI 1018 pipes	800
	Wooden Planks	00
Tyres	R10	00
Miscellaneous		00
Total		s.20,540

V. Conclusion

The goal of designing a strong and lightweight wheelchair was accomplished successfully. All the designs are safe as justified through simulations. The bill of material suggests the affordability of the product compared to other wheelchairs available in the market. The design eliminates the complexity of transferring the patients from wheelchair to stretcher. Thus, the wheelchair convertible to a stretcher eliminates the need to have both the products separately Also, the stretcher mechanism enables the wheelchair to be used in industrial applications for transportation. The wheelchair can be controlled either manually through joystick control, or through some automation.

The rendered image of the final CAD model of stretcher cum wheelchair is shown in Fig. 26

Figure 26

References

[1]. Dr. Sukanta Roga, Abhijeet Kumar, “Design and Fabrication of Wheelchair cum Stretcher with Multi Fold”, IJAIEM Volume 6, Issue 6, June 2017.

[2]. Prof. Nikhil V. Bhende, Mithun G. Kolhe, “Design and Fabrication of Lever Propelled Wheelchair”, IRJET Volume: 04 Issue: 03 | Mar -2017.

[3]. Jyothish K Sunny, Thomas Paul, “Design and Fabrication of Stretcher Cum Wheel Chair”, IJIRST Volume 2 | Issue 11 | April 2016.

[4]. P. Swapna , Dr. B. Sharmila, “Electric Wheelchair for Physically Challenged”, IRJET Volume: 03 Issue: 05 | May-2016.

[5]. Mohan Kumar R., Lohit H. S., “Design of Multipurpose Wheel Chair for Physically Challenged and Elder People”, SASTECH Volume 11, Issue 1, Apr 2012.

[6]. Mechanical Principles of Wheelchair Design, MIT.

[7]. Xin Chen, Zhong Wu, “An Optimization Design for Standard Manual Wheelchair”. Blekinge Institute of Technology, Sweden

[8]. Parag Nikam, Rahul Tanpure, “Design Optimization Of Chain Sprocket Using Finite Element Analysis”, IJERA ISSN : 2248-9622, Vol. 6, Issue 9,(Part-5) Sepember.2016, pp.66-69.

[9]. Center for Innovation in Product Development MIT, <http://lancet.mit.edu/motors/motors3.html>

[10]. Bryan Maddock,Dimensions.com,<https://www.dimensions.com/>

[11]. R.K. Bansal, 2009, Strength of Material.

[12]. R.S. Khurmi and J.K Gupta, 2005, Textbook of Machine Design.

[13]. PSG Design Data Book.

Siddhant Pawar, et. al. "Design and Analysis of Electric Wheelchair cum Stretcher." *IOSR Journal of Mechanical and Civil Engineering (IOSR-JMCE)*, 18(3), 2021, pp. 16-29.