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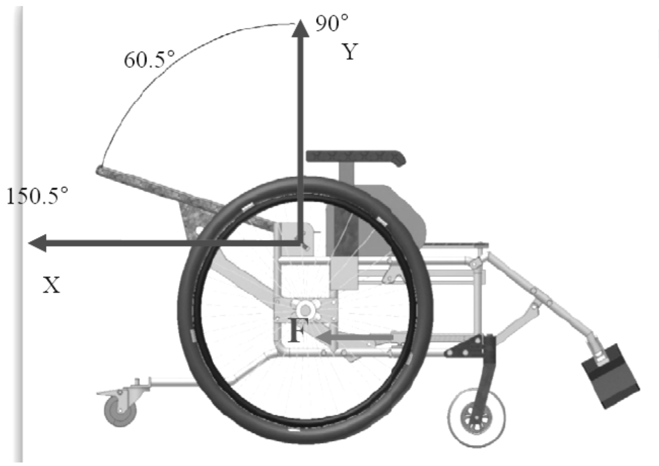
An Optimization Design for the Standard Manual Wheelchair



Xin Chen

Zhong Wu

Hongyu Deng

Department of Mechanical Engineering
Blekinge Institute of Technology
Karlskrona, Sweden

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Supervisors: Armando Leon, BTH
Sharon Kao-Walter, BTH

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Thesis submitted for completion of Master of Science in Mechanical Engineering with emphasis on Structural Mechanics at the Department of Mechanical Engineering, Blekinge Institute of Technology, Karlskrona, Sweden.

Abstract:

The standard manual wheelchair is commonly found in hospitals, rehabilitation centres, senior housing and airports for allowing patients, people with physical disabilities or elder people to move. In this thesis, a CAD model of the standard wheelchair is first obtained. Then, an optimization design is carried out based on this model. A sliding seat is designed to facilitate people moving by themselves from the wheelchair to any chair close by. On the other hand, a reclining backrest is also designed to make users feel more comfortable. Computer Simulations are performed to evaluate this design under static and dynamic conditions. A displacement range for the sliding seat in which the wheelchair will not overturn is determined. The optimization design shows significant improvements that can conveniently be introduced on the mechanical wheelchair in order to make it more useful and comfortable to the users.

Keywords:

Wheelchair, CAD Modelling, Optimization Design, Simulations

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Xin Chen

Zhong Wu

Hongyu Deng

Contents

1 Notation

A	Shaft diameter
A	The cross-sectional area
b	Width of strip
I_z	The moment of inertia for section
J	The moment of inertia
Mn	Maximum allowed torque in Nmm
Nc	Life, number of cycles
N	The axial force on cross section
n	The rotate speed
R	Radius of gyration
R	Rate
r	Radius from spring centre to location centre
T	Torque
t	Thickness of material
W_z	Anti-bending section modulus
y	The allowable deflection
y_{max}	The biggest deflection
φ	Torque angle at Mn
σ	Normal stress
θ	The allowable unit length torsion angle
θ_{max}	The biggest unit length torsion angle

2 Introduction

Disability affects 15-20% of every country's population. There are at least 650 million people with a kind of disability worldwide.

The total number of the population with a long-standing health problem or a disability (LSHPD) in 25 European countries is estimated to account for more than 45 million citizens. These statistics only refer to the population that is 16 to 64 years old. This means that approximately one in six persons (15.7%) of the working age population (aged 16 to 64) has either a long-standing health problem or a disability. One European in four declares having a member of their family affected by a disability. The wheelchair might be one of the equipment to be used by these populations Therefore, it is a meaningful idea if the normal wheelchair can be modified that makes it more suitable for daily life, it will help a lot of people. [1]

In fact, the standard manual wheelchair is found in hospitals, senior housing, airports, etc. and it is an affordable choice, it is important for part of the disabilities and elder people. Therefore, improving its functionality and comfort becomes a clear society's demand. This project has as a main focus to improve a mechanical wheelchair, by designing a sliding seat to facilitate people moving by themselves from the wheelchair to any chair close by, and also, a reclining backrest to make it more comfortable for users. Fundamental concepts in solid mechanics and dynamics, and engineering design tools as CAD and Computer Simulations are used. A displacement range for the sliding seat in which the wheelchair will not overturn is determined

The body of the thesis is described as: In chapter 3 a brief description of different wheelchairs is presented. In chapter 4, the modelling of the standard manual wheelchair and optimization design are presented. In chapter 5, the simulation and analysis are showed. Finally in chapter 6 the concluding remarks are presented.

3 Background

The first known dedicated wheelchair was made for Phillip II of Spain by an unknown inventor. After this, Stephen Farfle, who is a paraplegic watchmaker, built a self-propelling chair on a three wheel chassis in 1655. Then, John Dawson of Bath who comes from England designed a chair with two large wheels and one small one in 1783. [2]

With development of technology, new hollow rubber wheels, push rims, spoke wheels were added and used on wheelchairs.

The earliest wheelchair similar to what is used in modern is built by an engineer named Harry Jennings. It is made of tubular steel and can be folded.

In modern society, the revolution in powered wheelchair design, control, styles, range or travel distance, suspension, manoeuvrability, seating and other user options lead to the different types of wheelchairs in the market. Which type the users need is depending on the requirements, the respective functionalities, costs, etc. The most common ones are the following:

(1) Manual wheelchairs

The manual wheelchair is moved by pushing down or pulling back the push rims so that the users can go forward and backward by the force they apply.

Figure3.1. Manual wheelchair [3]

(2) Electric wheelchairs

Electric wheelchair is powered by an electric motor. It is intended for use in daily life and some of them are made for indoor and outdoor use.

Figure3.2. Electric wheelchair [4]

(3) Wheelbase chair

This kind of wheelchair has four small wheels extending from a low platform and the type of chair mounted on this platform varies according to the needs of the users.

Figure3.3. Wheelbase chair [5].

(4) Sports chairs

This type of wheelchair is designed for athletes with disabilities. It has lightweight frames and it is stability for sudden turns.

Figure3.4. Sports chair. [6]

(5) Stand-up wheelchairs

The stand-up wheelchair, which makes users stand up, is fitted with a hydraulic pump that provides enough support for lifting the seat. Therefore, the users can get things on higher place.

Figure3.5. Stand-up wheelchair [7]

(6) Stair-climbing wheelchairs

The stair-climbing wheelchair is used for users to climb stairs indoor or outdoor. But users still need the help of attendants. Besides, it is necessary for users to grasp a suitable handrail. [8]

Figure3.6. Stair-climbing wheelchair [9]

4 Modelling and Design

In this thesis work, the first step for developing an optimization is to model a standard manual wheelchair like following figure which is photograph by author.

Figure4.1. A standard manual wheelchair

And the design framework is given below:

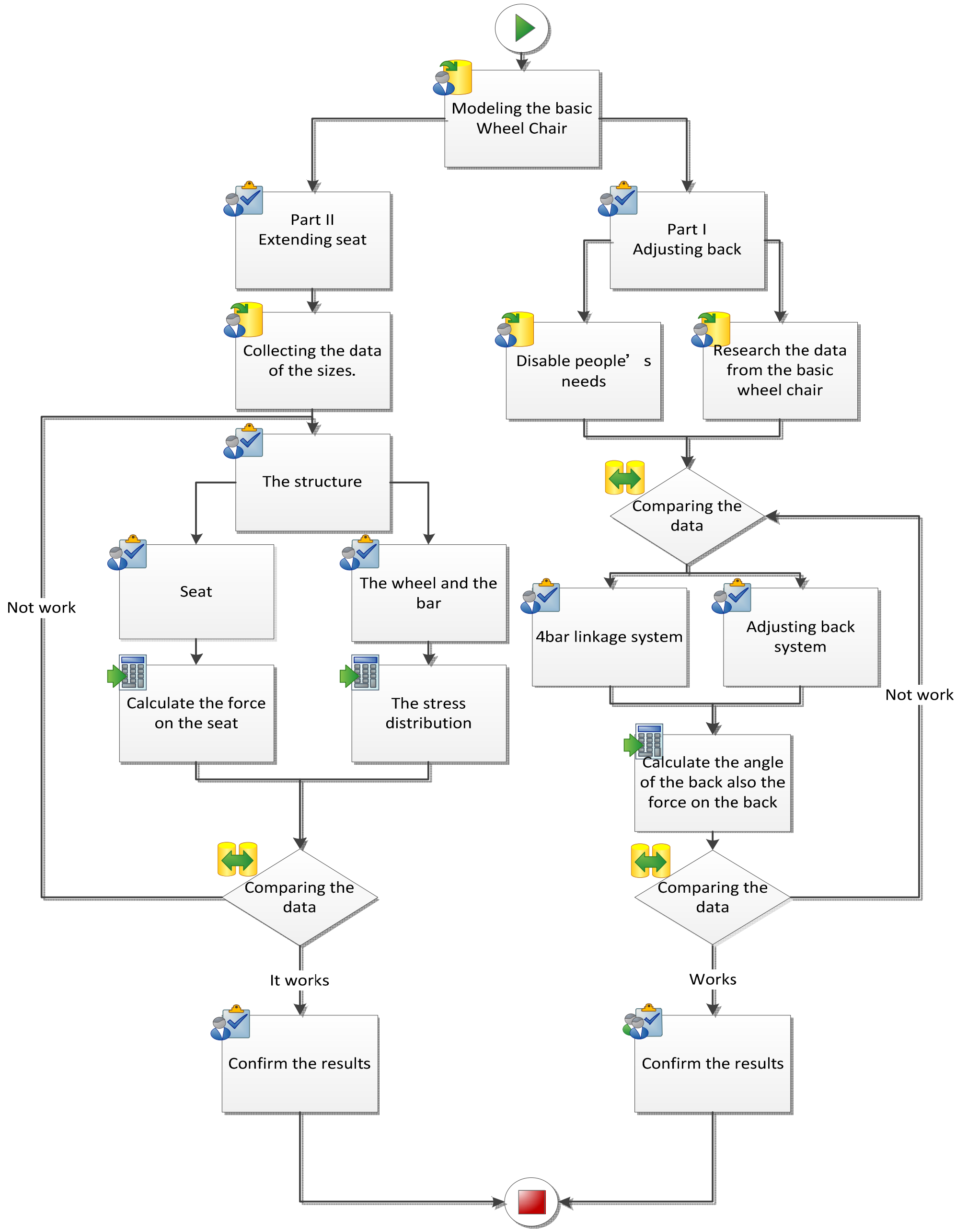

Figure4.2. Design framework

4.1 Software introduction: Autodesk-Inventor

Autodesk Inventor, developed by U.S.-based software company Autodesk, is 3D mechanical solid modelling design software for creating 3D digital prototypes used in the design, visualization and simulation of products. [10]

Part modelling of solid and surface modelling is combined to create a series of complex geometries. There are many tools for us to use like Zebra and Gaussian analysis tools which can be used to check the cutting, continuity and curvature. The project can be started from scratch, reuse the existing components and also changes the parts design, and incorporated into the Autodesk software to create a name for the conceptual design of curves and surfaces data.

This software is quite suitable and helpful for us to model a wheelchair.

4.2 Fundamental knowledge about mechanical wheelchair

A mechanical wheelchair is consisting of four main parts, which are framework, mat, backrest and adjusting device, and other parts.

4.2.1 Framework

The framework is used to support the whole seat and body mass. It consists of two parts which are mat framework and backrest framework. The strength and the stiffness of the framework should be enough in order to ensure security and load bearing.

4.2.2 Mat

In order to avoid pressure sores, the mat should be paid a high degree of attention. The ischia tuberosity bears great pressure when people on the seat; often beyond the pressure of the end of the normal capillary 1 to 16 times, which makes it easy to form a pressure sore. To avoid too much pressure here, the mat is often dug a small part in the corresponding position which makes ischia tuberosity overhead. The position of digging should be 2.5cm in front of the ischia tuberosity, the lateral should be 2.5cm in the lateral tubercle and depth of pad is about 7.5cm. The cushion should be concave shape after digging with the gap at the back. With the use of such mats incision, pressure sores can be quite effective in preventing.

4.2.3 Backrest

Back can be high or low and tilt or cannot tilt. If the patient can control the balance of the trunk well, the low back of the wheelchair should be chosen so that patients have a greater activity. On the contrary, high back wheelchair should be chosen.

4.2.4 Adjusting device

It is used to adjust the wheelchair for different figures. Its structural form includes front and back, up and down, angle of the backrest, etc.

4.2.5 Big wheel

The big wheels are the major part to bearing weight. Besides a few wheels which have particular use for the environmental requirements to be solid tire, the multi-inflated tires are mostly used. The diameter of the big wheels has different sizes such as: 51cm, 56cm, 61cm, 66cm, etc.

4.2.6 Small wheel

Small wheels with big diameter ferry over obstacles and special carpet easily. But if the diameter is too big which will affect the entire wheelchair space to become larger that makes inconvenience. The diameter of the small wheels has different sizes such as 12cm, 15cm, 18cm, 20cm, etc.

4.2.7 Brake

The big wheel should have the brakes each round, of course, for those who have a hand hemiplegia, a single hand brake can be used, but it can also be installed to extend the rod, control both sides of the brake. Brake has two types: (1) notch brake. This kind of brake is safe and reliable, but it costs too much force. It can be stopped on the slopes after being adjusted and it is thought to be invalid if it cannot be stopped on the ground after being transferred to one level. (2) Toggle brake. It makes use of leverage and works with a few joints. The mechanical advantage of this brake is better than the notch brake, but its invalid speed is faster.

4.2.8 Feet and legs up care

There are two types of feet and legs up care: one is across both sides, another is separate both sides. Both are asked to be swung to two sides and can be removed for the best. The height of the feet care should be paid attention to. The higher the feet care is, the larger the hip flexion angle will be, and this will add more weight to the ischia tuberosity, which easy to cause pressure sores.

4.2.9 Handrails and arm care

Generally, handrails and arm care should be 22.5 ~ 25cm higher than the surface of the seat. For some arm care, its height can be adjusted. The lap board can be used on the shelf of arm care in order to reading, dinning, etc.

4.3 Modelling a basic mechanical wheelchair

A standard manual wheelchair will be modelled by using the software Autodesk-Inventor in order to get a deep understanding on the construction. Then, some structures are going to be modified.

4.3.1 Main parts of model wheelchair

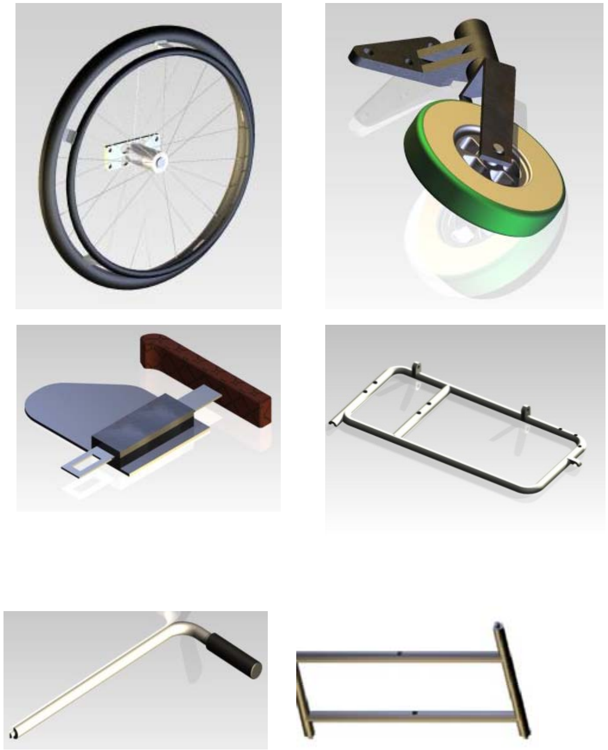

Figure4.3. Main parts of model wheelchair.

4.3.2 Whole model wheelchair

Figure4.4. Whole model wheelchair

4.4 Optimization design

The optimization design for the model mechanical wheelchair will be used to help most needed people to make the daily life easier and comfortable, especially for the elderly and the disabled people. There will be two functions which are seat sliding and backrest reclining. In order to carry out these two functions, a certain parts will be improved and some elements are going to be added on the basic mechanical wheelchair.

4.4.1 Sliding seat

The first approach is to help people to move from the wheelchair to any other one, for example to a toilet as shown in figure 4.5:

Figure4.5. Toilet designed for physically disabled people.

This kind of toilets for the disable person is everywhere in Sweden, but it is just simple construction with some handles on the both sides of the seats.

It may be possible for the people have long-time standing problem to get on the seats, but it is very difficult for the people who cannot stand to get on it, sometimes it maybe need someone else to help the disable person, which is very inconvenient.

For the seat adjusting system, we considered two different models.

1) The Dual Rail idea which came from the travelling crane from the manufacturer (Figure 4.6)

Figure4.6. Travelling crane [11].

2) The single Rail idea which comes from the Rowing machine (Figure 4.7)

Figure4.7. Rowing machine [12].

4.4.2 Reclining backrest

The second idea is introducing a backrest. In summer, Sweden has a very beautiful and sunny daytime. It is also healthy for the disable person to have sunbath. Another reason, it is very tiring if one person sits for a long time, therefore it will be quite comfortable if they can lie down for a while. So the user can enjoy the sunshine comfortably.

4.4.2.1 Four bar linkage system

For the reclining backrest system a four bar linkage is used to satisfy the requirements

Figure4.8. Four bar linkage system.

In this case, the four bar linkage system was applied according to the back of chair lying down with a very low velocity. The merits of the four bar linkage system are that the pressure on unit area is less, and surface contact is easy for lubrication, so the wear decreased; it is easy to manufacturing and acquire high precision; the contact between the two components sustains by the geometric closed of itself, it does not need spring force to stay in touch. The demerits of the 4bar linkage system are that, normally, only the given trajectory and rule of movement can be achieved approximately, and the design is more complicated; when given movement has more requirements or more complex, the number of components is more which makes the structure complex and the working efficiency lower, not only the possibility of self-locking is increased, but also the sensitivity of mechanism motion rule for manufacturing and installation error increase; the inertial force which is produced by the components for complex movement and reciprocating motion in mechanism is hard to balance, , it will cause larger vibration and dynamic load under a high speed, so linkage mechanism is often used to occasion of lower speed.

4.4.2.2 Calculation for torque of the backrest

According to the formulas:

$$T = J \cdot V / R \tag{4.1}$$

$$J = \frac{(m \cdot R^2)}{2} \cdot (kg \cdot m^2) \tag{4.2}$$

$$V = 3.14 \cdot D \cdot n / 60 \tag{4.3}$$

Where

R is the radius of gyration

T is the torque

J is the moment of inertia

n is the rotate speed

From the inventor, m=14.157kg

Estimate the mass of users is 80kg

The torque of the backrest is:

$$T=788.407947 \text{ Nmm}$$

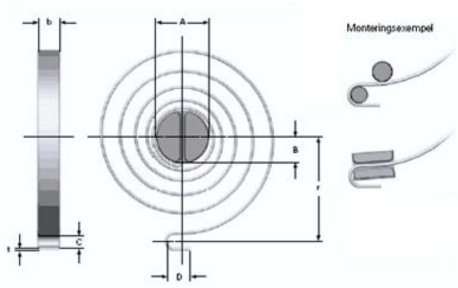

Figure4.9. Clock spring [13].

All dimensions in mm

t=Thickness of material

b = Width of strip

A = Shaft diameter

r=Radius from spring centre to location centre

n = Number of coils

φ=Torque angle at Mn

Mn = Maximum allowed torque in Nmm

R = Rate, Nmm per degree

Nc = Life, number of cycles

Tolerance: Tolerance for the position between inner and outer locating point is ±10 degrees for 5 coil springs and ±15 degrees for 8coil springs.

1 kp = 9.80665 Newtons

1 Newton = 0.10197 kp

The dimensions of scroll spring needed here are b<5; r<25; A<=8.

Cat.no	Description	t	b	A	r	n	B	C	D	R	Nc max 10.000 torque at Mn	Mn Nmm	Nc max 100.000 torque at Mn	Mn N
0900	SF-SF	0,5	3	7	13	5	2,5	2,7	3,5	0,56	354	198	284	198
0902	SF-SF	0,5	3	7	21	8	2,5	2,7	3,5	0,26	762	198	610	198
0901	SF-SF	0,5	5	7	13	5	2,5	2,7	3,5	0,93	354	329	284	329
0903	SF-SF	0,5	5	7	21	8	2,5	2,7	3,5	0,43	762	329	610	329
0904	SF-SF	0,6	4	8	16	5	3	3,2	4,5	0,9	416	374	332	374
0906	SF-SF	0,6	4	8	25	8	3	3,2	4,5	0,43	862	374	690	374
0905	SF-SF	0,6	6	8	16	5	3	3,2	4,5	1,35	416	562	332	562
0907	SF-SF	0,6	6	8	25	8	3	3,2	4,5	0,65	862	562	690	562
0908	SF-SF	0,7	4	10	19	5	3,5	3,7	5	1,43	354	506	283	506
0910	SF-SF	0,7	4	10	29	8	3,5	3,7	5	0,67	761	506	609	506
0909	SF-SF	0,7	7	10	19	5	3,5	3,7	5	2,5	354	886	283	886
0911	SF-SF	0,7	7	10	29	8	3,5	3,7	5	1,16	761	886	609	886
0912	SF-SF	0,8	5	12	21	5	4,5	4,2	6	1,79	456	816	364	816
0914	SF-SF	0,8	5	12	34	8	4,5	4,2	6	0,83	986	816	789	816

Figure4.10. Data of clock spring [14].

From the table above, the Cat.no 0912 should be chosen and its dimensions are:

$$Mn=816>788.407947,\ b=5;\ r=21<25;\ a=10>8$$

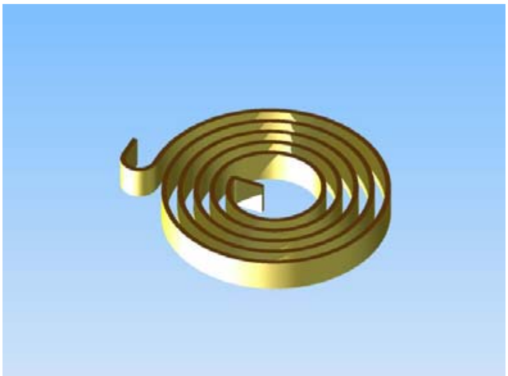

Figure4.11. Spring in the backrest system.

4.4.3 Remodelled wheelchair

Based on the two functions and the calculations made above, the wheelchair is remodelled with the help of Autodesk-inventor. As mentioned above, we have two solutions for the seat adjusting system, dual rail and single rail sliding seat, hence we remodel them both, and discuss the advantageous and disadvantageous of them in the chapter 5. The structure is showing as following figures:

Figure4.12. Whole remodelled wheelchair of the dual rail sliding seat system.

Figure4.13. Whole remodelled wheelchair of the single rail sliding seat system.

The following figures will make a clear show of two functions:

Figure4.14. Dual rail sliding seat

Figure4.15. Single rail sliding seat

Figure4.16. Reclining backrest of the dual rail sliding seat system

Figure4.17. Reclining backrest of the single rail sliding seat system

4.5 Material

4.5.1 Green materials

The choice of materials in frame, to some extent, affected the comfort of wheelchair because of the effect of elastic and vibration absorption of the material. From the angle of vibration absorption, the difference is small for stainless steel, aluminium alloy and pure titanium, but from the angle of the elastic rate, stainless steel is 193GPa, aluminium alloy is 72GPa, industrial pure titanium is 105GPa, so the vibration of the wheelchair which is made of aluminium alloy is minimal when the same load is given to these three different materials. Considering the lightweight, economy and its impact on the comfort of the influence of materials, the use of aluminium alloy is very reasonable. The usage of light metals in a wheelchair is now a tendency of development, if titanium can reduce costs, it will have very good development prospect on wheelchair market.

In recent years, the green environmental protection materials begin to occupy the market, and with the popularity of green economy, new environmental materials are trending to sweep the globe. The thriving green wheelchair design also spurs a huge requirement of green materials. To be the basis and source of wheelchair manufacturing, the green environmental protection materials increasingly become the mainstream of marketing purchasing.

4.5.1.1 High-density composite materials

The main process principle of new high-density composite materials is that it is combined by making use of agricultural waste as raw material and cities waste such as polyethylene and polypropylene as adhesive materials with a certain amount of additives under a certain technology condition. This latest achievement is called "the 21st century new materials" by the United Nations Industrial Development Organization (UNIDO). It is of strong plasticity and good fire proofing.

4.5.1.2 High modulus fibers

At present, the tension and fracture strength of nylon and polyester tensile are only about 5% of its theoretical value. With the development of polymer fibre, the tension and fracture strength will be 40% of its theoretical value. The tensile modulus will be 90% of its theoretical value. As the development of polymer technology and the combining of organic and inorganic compounds, fibres with 40% theoretical strength is likely to be developed. This kind of fibre will be applied to various devices which have requirements of high strength and light quality.

4.5.2 Choosing wheelchair materials

4.5.2.1 Choose wheelchair materials according to strength theory

Axial tension and compression are two simple forms of the beam deformation. In machinery and equipment, beam is the part which mostly bearing the tensile and compressive function. Bolts are always used to connect wheelchair parts, they bearing tension after being tighten. To analyse the beam which bearing axial tensile or axial compression function, it will be turned out that the characteristics for the tension and compression of a beam is that two force functioned at two ends of the beam is equal and opposite, and action line is coincident with beam axis. The characteristic of its deformation is that the beam will elongation or shorten along the axis and its cross-sectional will be thinner or coarsen, as shown in following figure:

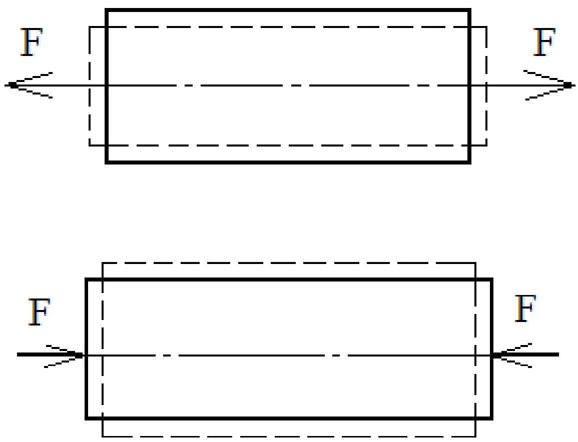

Figure4.18. Characteristic of deformation.

What talked above shows whether by stretching or compressing, the internal force on the cross section of the beam is along the axis of the beam, this force becomes axial force, it can be tension, also can be the pressure. The stress of each point on cross section is determined by the internal force distribution on the cross section of the beam. Under the action of the internal force, the beam will not only produce internal force, but also causes deformation and internal force are closely related to the deformation.

Take a beam which has the same cross section for example, two lines *ab* and *cd* which are perpendicular to the axis of the beam are drawn on the surface, and two vertical lines which are parallel to the axis of the beam are drawn between two horizontal lines. Then a pair of axial tensile force is added at both ends of the beam, which makes beam produce tension and deformation. It can be observed from the surface of the beam that the straight lines *ab* and cd are translated to the position of lines $a_1b_1$and $c_1d_1$respectively, but still the straight lines and vertical to the axis of the beam; two vertical lines are elongation with the same value and still parallel to the axis of the beam. So, the cross section which is plane before deformation will still be plane after deformation, but it translated along the axis, as shown in following figure:

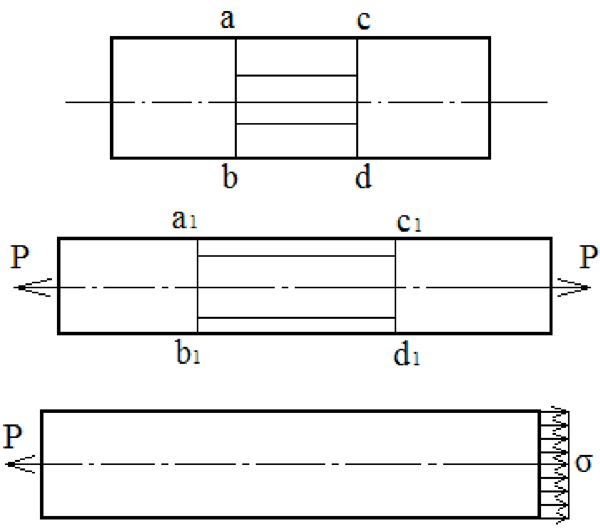

Figure4.19. Change of cross section.

According to the plane assumption, the elongation or shorten of each longitudinal line between two arbitrary cross sections is the same. From the uniform continuity hypothesis of the material, the internal force on the cross section is even-distributed, which is namely the stress at each point is equal.

Set the cross-sectional area of the beam is A, the axial force on cross section is N, then normal stress on this cross section is:

$$\sigma = N / A \tag{4.4}$$

As mentioned above, the stress that the material of the beam can withstood is limited. In order to ensure the beam can work normally, the work stress of the beam must not exceed the allowable stress of the materials. Therefore, conditions for stretching or compressing of the beam are:

$$\sigma = \frac{F_N}{A} \leq [\sigma] \tag{4.5}$$

Where the $[\sigma]$ means the allowable stress of the beam material.

At present, the material of wheelchairs on the market is mostly using hard aluminium alloy. Its biggest tensile strength is 370MPa, which is namely the maximum allowable stress is 370MPa, but the result obtained by calculation is far less than its maximum allowable stress. So the main structure of the wheelchair will use hard aluminium alloy.

4.5.2.2 Choose wheelchair materials according to the stiffness theory.

Under the action of different loads, the beam will produce different deformation. According to the different properties and the positions of the load, deformation can be divided into four basic types which are axial tension, shearing and torsion and bending.

The rigidity condition for circular shaft torsion is that the biggest unit length torsion angle θ_{max} of circular shaft should not exceed the allowable unit length torsion angle $[\theta]$:

$$\theta_{\max} = \frac{T}{GI_p} \leq [\theta] \tag{4.6}$$

The rigidity condition for beam bending is that when beam is bending, the deflection and corner for a specified section does not allow to exceeding the allowable value:

$$y_{\max} \leq [y]$$

Or

$$\theta_{\max} \leq [\theta]$$

Where $[y]$ and $[\theta]$ mean allowable deflection and torsion angles respectively.

(1) Stiffness calculation when torsion

If each end of the beam is acted on an external couple Me, and the size of them is equal while the direction of rotation is opposite, acting surface is vertical to the axis of the beam, then the cross section of the beam will occur relative motion around the axis, this deformation is called torsion, as shown in following figure:

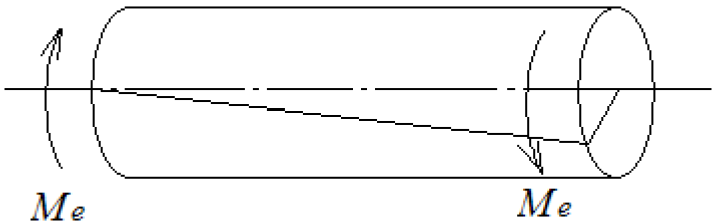

Figure4.20. Torsion deformation of the beam

The feature of its deformation is that the arbitrary two cross sections of the beam do relative rotation around the axis of the beam, the relative angular displacement φ between two cross sections is called torsional angle, *and* φ means the torsional angle of section B relative to the section A. The longitudinal line of the beam has a tiny tilt when torsion , and the angle of inclination of the longitudinal line of the surface is shown by γ. As shown in following figure:

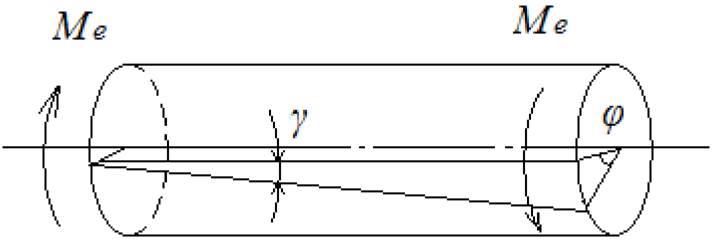

Figure4.21. Torsion deformation when the beam is torsion.

(2) Stiffness calculation when bending

If each end of the beam is acted on an external couple Me, and the size of them is equal while the direction of rotation is opposite, acting surface is coincident to a certain longitudinal plane which contains the axis of the beam, or when external force F which is located in longitudinal plane and vertical to the axis of the beam is acting on it, the axis of the beam will bend, this deformation is called torsion, as shown in following figure:

Figure4.22. Pure bending

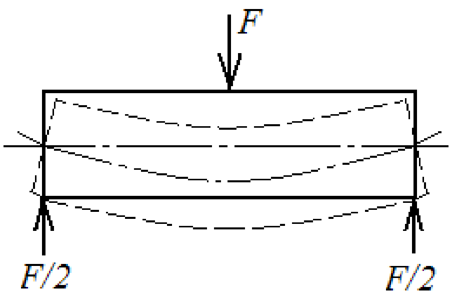

Figure4.23. Horizontal force bending

The shear figure and bending moment can be drawn as following figure:

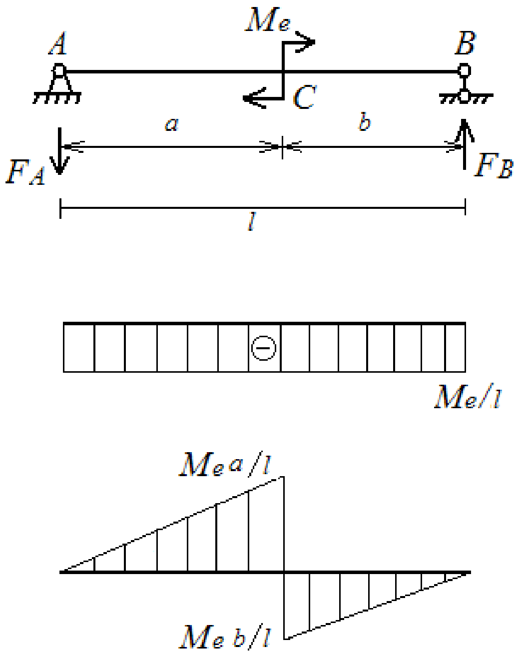

Figure4.24. Shear figure and bending moment

The allowable deflection of the beam bending is that the maximum deflection for the selected material should not exceed its allowable deflection. The maximum deflection for the main structural beam is:

$$y_{\max} = \frac{I_z}{W_z} \tag{4.7}$$

Where I_z means moment of inertia for section and W_z means anti-bending section modulus.

According to the bending rigidity condition $y_{\max} = \frac{I_z}{W_z}$, the maximum deflection for selected material should be less than its allowable deflection $[y]$.[15]

5 Simulations and Analysis

In order to evaluate our design, two different types of simulations were performed

Static Stress Analysis

Dynamic Simulation

The first one is carried out in order to verify that the wheelchair structure can resist the expected loads under static conditions. On the other hand, a dynamic simulation is performed to determine the safety displacement range in which the sliding seat can move without producing the overturning of the wheelchair.

5.1 Stress analysis under static conditions

For the static analysis, Autodesk-inventor is used to show the result.

5.1.1 Geometry

For the simulations, a more simplified model than the obtained was used in order to reduce convergence problems when running the FEM analysis. This simplification also reduced the complexity of defining contact characteristics among different components by screws, or pressure pins. As we have two solutions to model the seat adjusting system, the dual rail and single rail, hence we analysis them respectively.

(1) Model 1: The Dual Rail seat adjusting system

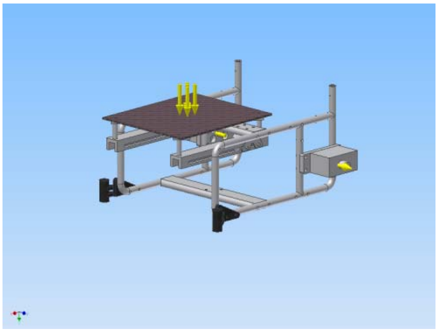

Figure5.1. Static analysis for the Dual Rail seat adjusting system

(2) Model 2: The single rail seat adjusting system:

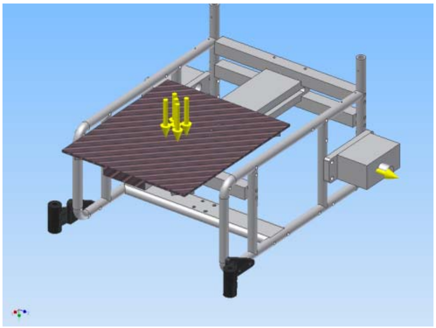

Figure5.2. Static analysis for the single rail seat adjusting system

It is easily found the size of the rail is wider.

5.1.1 Load

It is assumed that the person who seats on this wheelchair is about 85kg. The load is considered as a homogenously distributed load (pressure) over the seat

(1) For model 1:The Dual Rail seat adjusting system

The weight of the wheelchair is 12.9873kg. The Gravity is forced on the point of

$$x=-15.7691 \text{ mm}$$

$$y=71.9308 \text{ mm}$$

$$z=0.0235006 \text{ mm}$$

(2) For model 2: The single rail seat adjusting system:

The weight of the wheelchair is 19.4902 kg. The Gravity is forced on the point of

$$x=-72.0057 \text{ mm}$$

$$y=184.731 \text{ mm}$$

$$z=-50.1209 \text{ mm}$$

5.1.2 Boundary conditions

These two models have the same boundary conditions.

The chair is settled by the 4 points which is showed in figure 5.3:

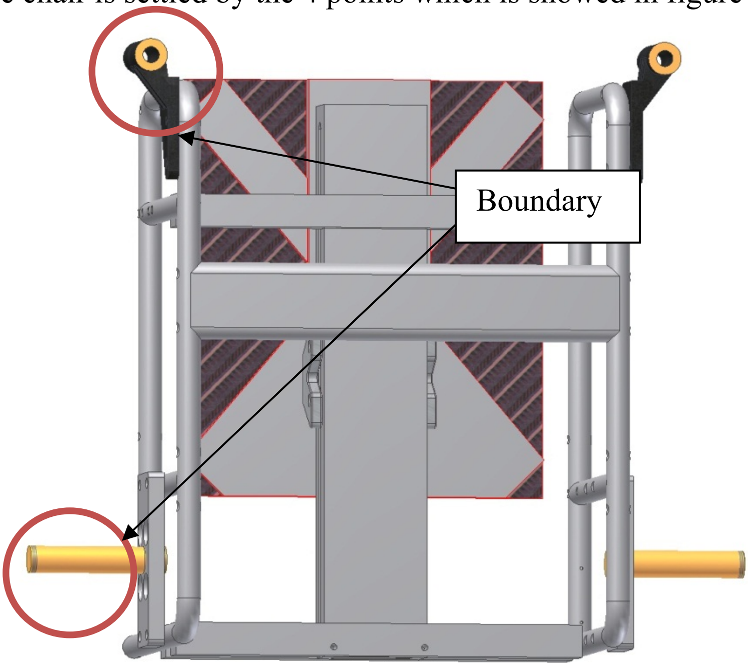

Figure5.3. Four points that settled the chair

5.1.3 Result

By applying the load, boundary condition and material into the simulation function of the Autodesk-inventor, the analysis result can be obtained. For the seat adjusting system, the two modes mentioned above are compared in two different situations. One is when the seat has moved to the front side of the chair, while the other one is back side. The result will be shown in two field, von Mises stress and displacement.

(1) For the front situation:

1) Mode 1 dual rail

Von Mises Stress

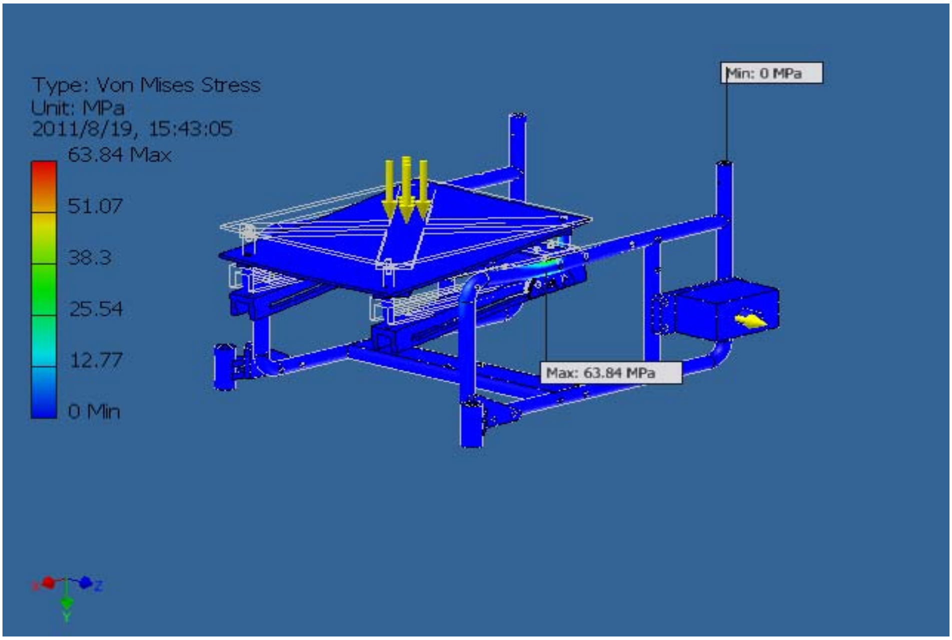

Figure5.4. Von Mises Stress of dual rail in the front situation.

Displacement

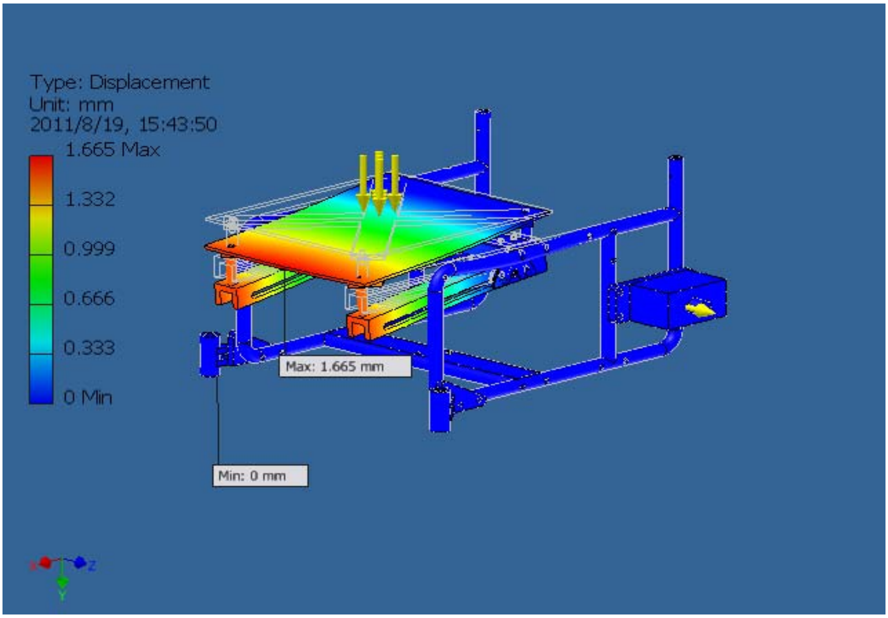

Figure5.5. Displacement of dual rail in the front situation.

2) Mode 2 single rail

Von Mises Stress

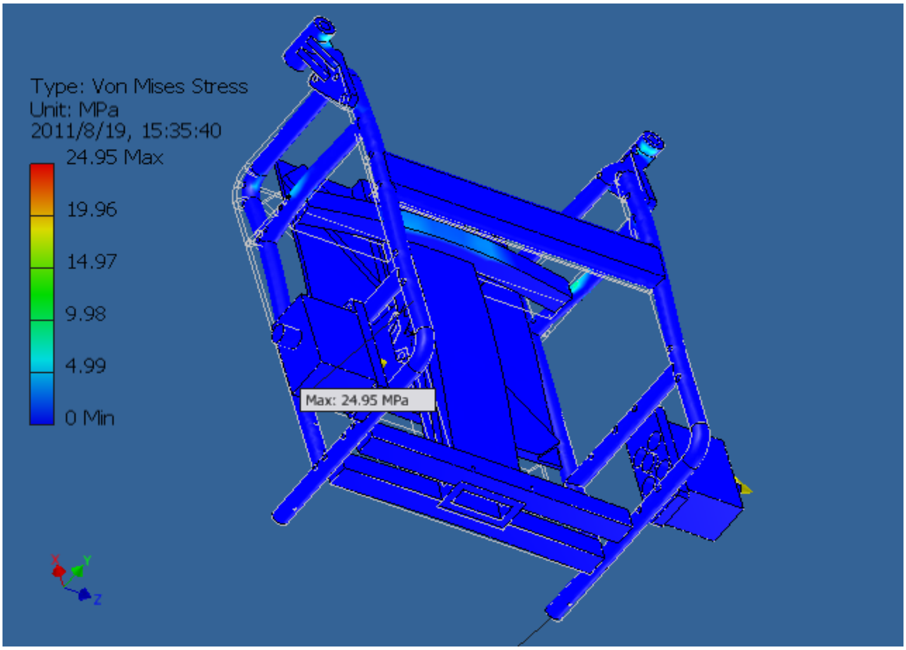

Figure5.6. Von Mises Stress of single rail in the front situation.

Displacement

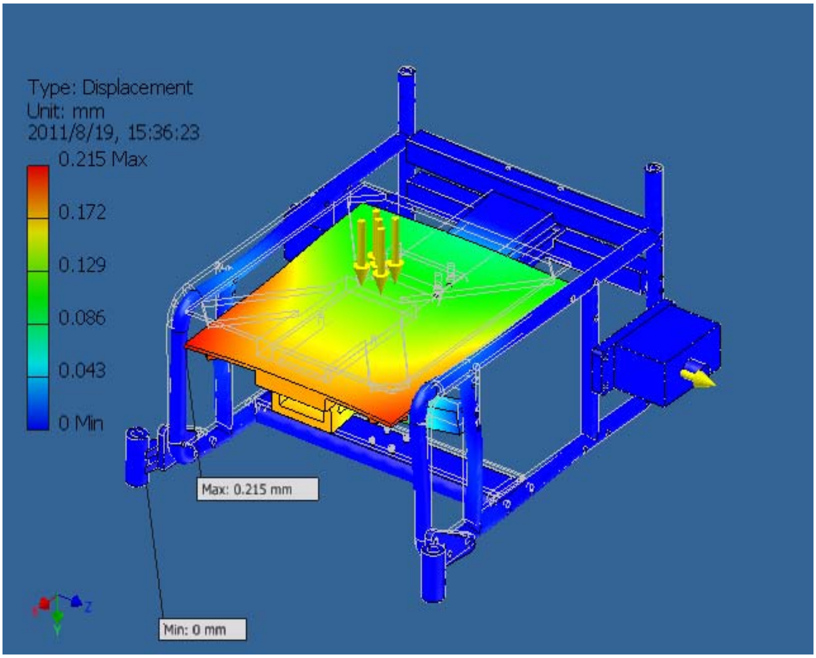

Figure5.7. Displacement of single in the front situation A.

(2) For the back situation:

1) Mode 1 dual rail

Von Mises Stress

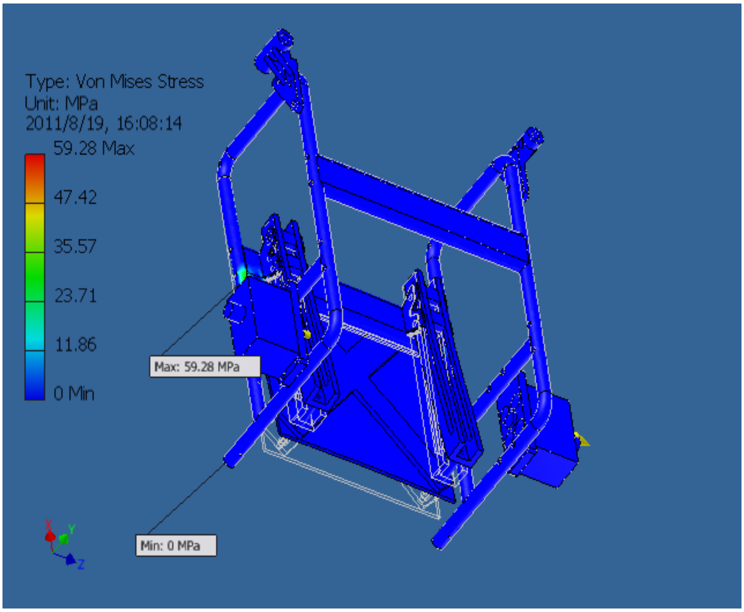

Figure5.8. Von Mises Stress of dual rail in the back situation.

Displacement

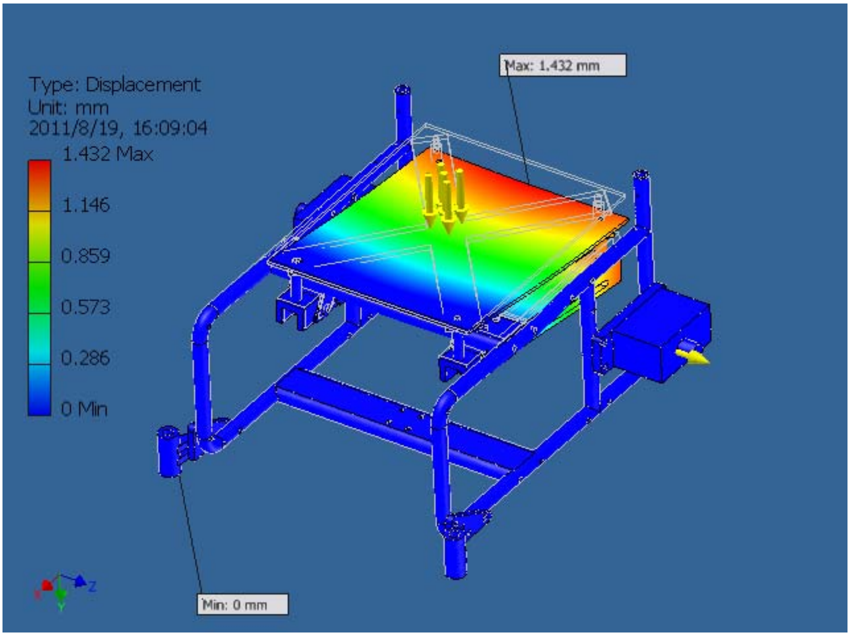

Figure5.9. Displacement of dual rail in the back situation A.

2) Mode 2 single rail

Von Mises Stress

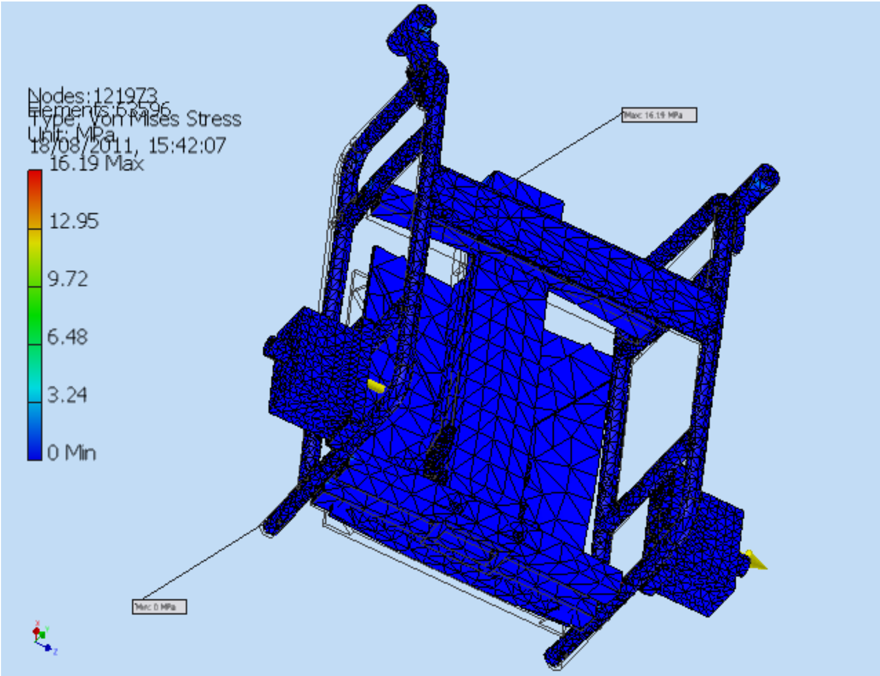

Figure5.10. Von Mises Stress of single rail in the back situation.

Displacement

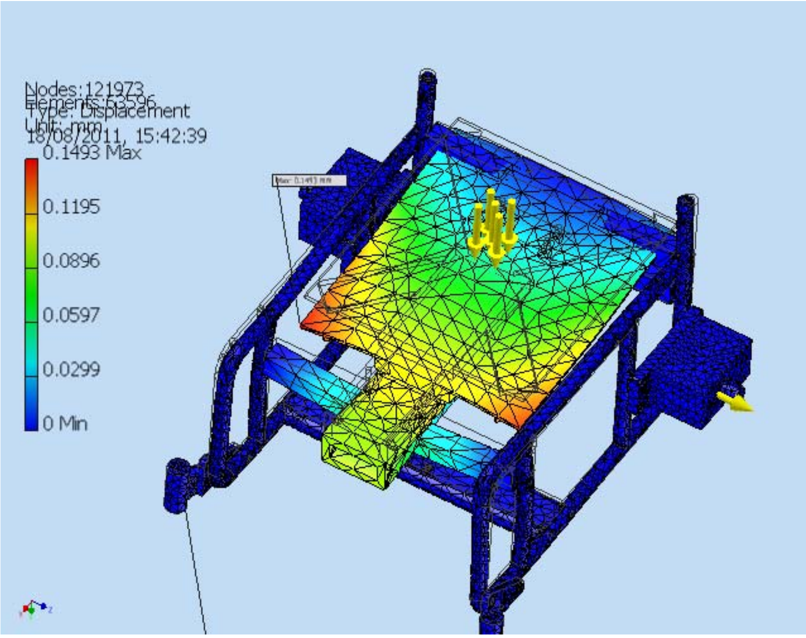

Figure5.11. Displacement of single rail in the back situation A.

From all the figures above, we collect and combine the data; following table can show the comparison more clearly.

Table 5.1 Comparison of two modes in two situations

Position	Mode	MAX Von Mises stress [MPa]	Max Displacement [mm]
Front	Dual rail	63.84	1.665
	Single rail	24.95	0.215
back	Dual rail	59.28	1.432
	Single rail	16.19	0.1493

The result was showed above. The single rail is safer than the dual rail. Also the seat is more stable. The single rail used too much material on the rail which causes the weight of the whole chair heavier (See appendix). So considering the safe factor, the single rail is more acceptable.

5.2 Dynamic analysis

Besides determining the safety range displacement for the sliding seat, we also run dynamic simulations to check the angle range of the backrest and the force at the respective spring connect to the four bar linkage. (Figure5.12)

In order to calculate this, we consider a constant speed reclining of the back is 5mm/s.

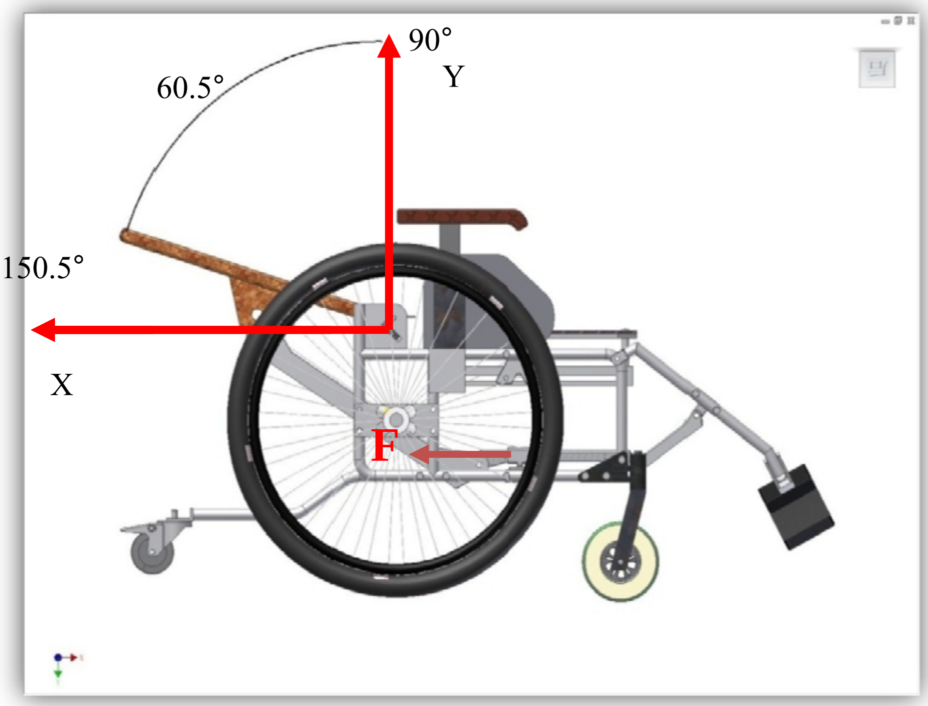

Figure5.12. Angle range of the backrest

The angle range is from 90 degree to 150.5 degree. The backrest can recline 60 degree.

The range of the force acting on the backrest is shown in figure 5.13:

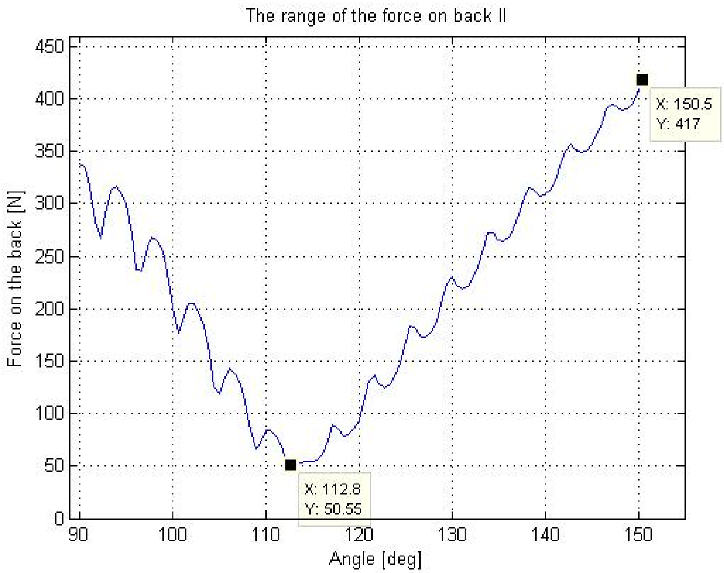

Figure5.13. Range of the force acting on the backrest in time domain.

The maximum force acting on the backrest is 417 N while the minimum is 50.55 N.

Determining safety displacement range for sliding seat

Now we try to find the safe displacement range of the seat which will not cause the chair rotating (overturning). See Figures 5.24 & 5.25.

We set the seat moving from the rear edge to the unlimited front. The inventor will simulate the chair situation.

Two cases are studied:

1. The load is assumed to be on the frontal half of the seat (Figure 5.17)
2. The load is assumed to be on the rear half of the seat (Figure 5. 24)

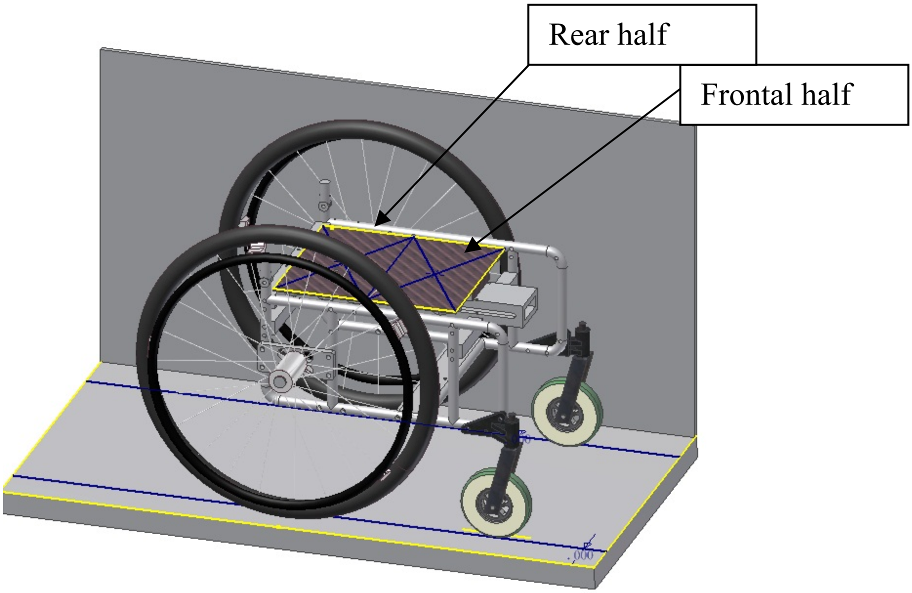

Figure5.14. Simulation of the chair from the original position

The idea is to estimate the critical position for the sliding seat in which the wheel chair will overturn (Figure 5.15).

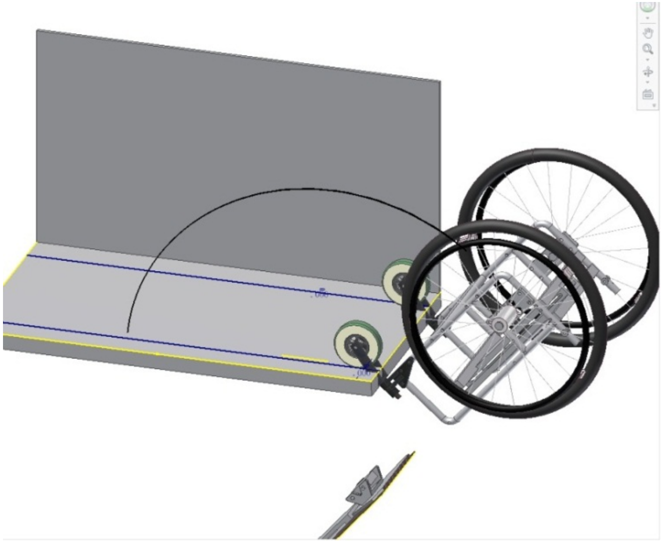

Figure5.15.Simulation of chair's Overturning

5.2.1 Load on the frontal half of the seat

Depending on the user's habits, some people are used to sit in the forefront of the chair; others like to sit in a whole chair.

In this section we consider the first case. We simulate the seat sliding from the original position to unlimited distance at a constant speed. Then we check the seat position in which the wheelchair will overturn. In this sense, we can determine the critical seat position and as a consequence to determine a safe displacement range position of the seat.

The seat is sliding from the original position (see figure 5.14) until its maximum outer position at a constant speed. The simulation time is 1.8 seconds (See figure 5.16). The load is considered as concentrated force of 850 N on point A (See figure 5.17). As mentioned before, this load corresponds to the weight of a person (approx. 85 Kg).

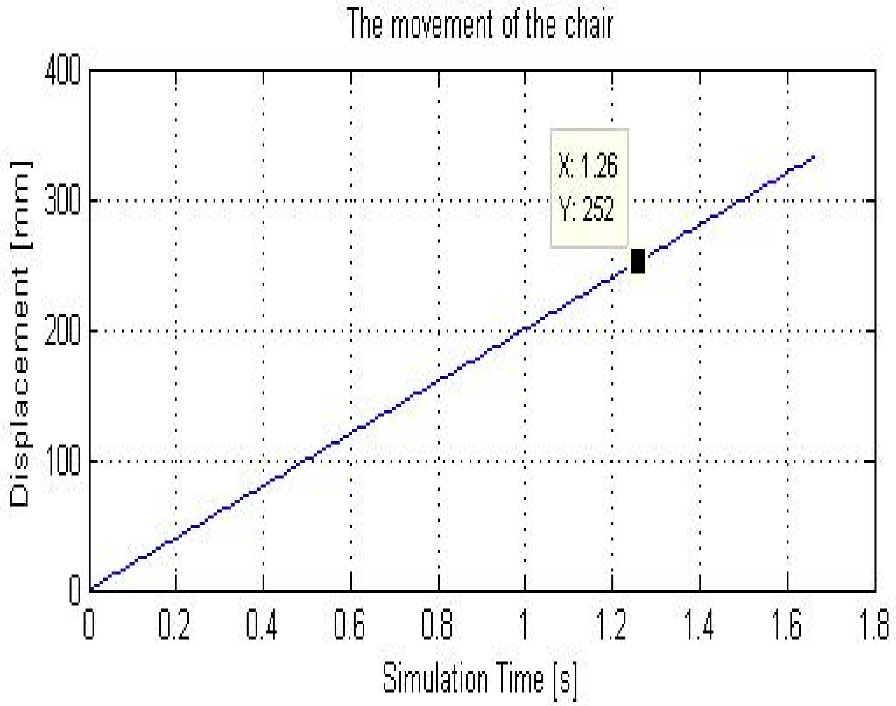

Figure5.16.The range of simulation time and displacement

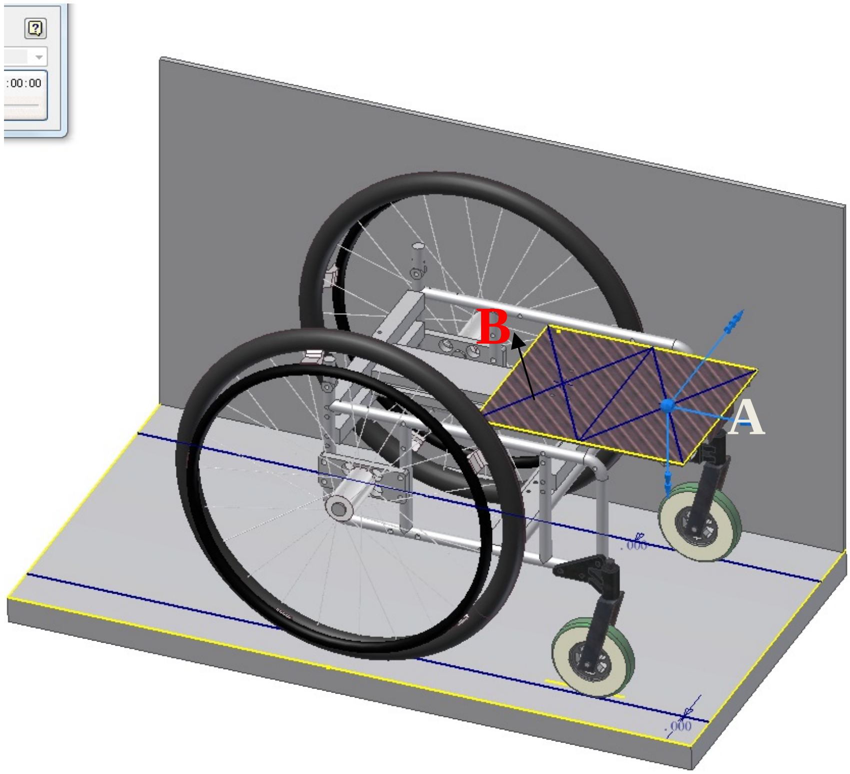

Figure5.17.Load position for simulations

In order to determine when the wheelchair will overturn, we measure the velocity in vertical direction of the driving wheel through its centre (see figure 5.18). Since it is expected the wheel chair to initially be on rest, as soon as this velocity changes from zero, it will indicate that the wheel chair is overturning.

The velocity of the big wheel and the displacement of the seat was calculated and plotted in figure 5.18. Figure 5.19 shows these results in one curve. Then, we can see directly at which seat position the vertical velocity of the big wheel start to increase from zero.

We should use force moment to find the point, but on the other hand the velocity is the same thing we need which is easy to show when the chair start rotate.

The range of time is limited between 0to 1.66 second.

The range of displacement is limited between 0 to 331.9999 mm.

In this way, when the velocity is higher than zero, we can determine when the wheelchair will overturn. Therefore we can obtain the safe range of displacement of the sliding seat.

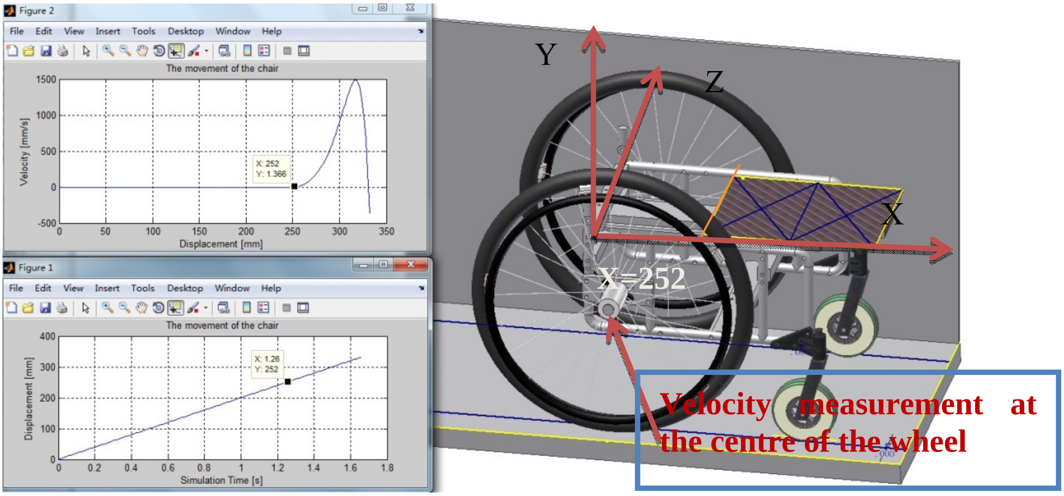

Figure5.18. Loading position of seat adjustment dynamic simulation

The velocity of the centre of the wheel can be seen in more details at figure below. We can notice that at 250mm the wheelchair can overturn. Therefore, our safety range should be below this value.

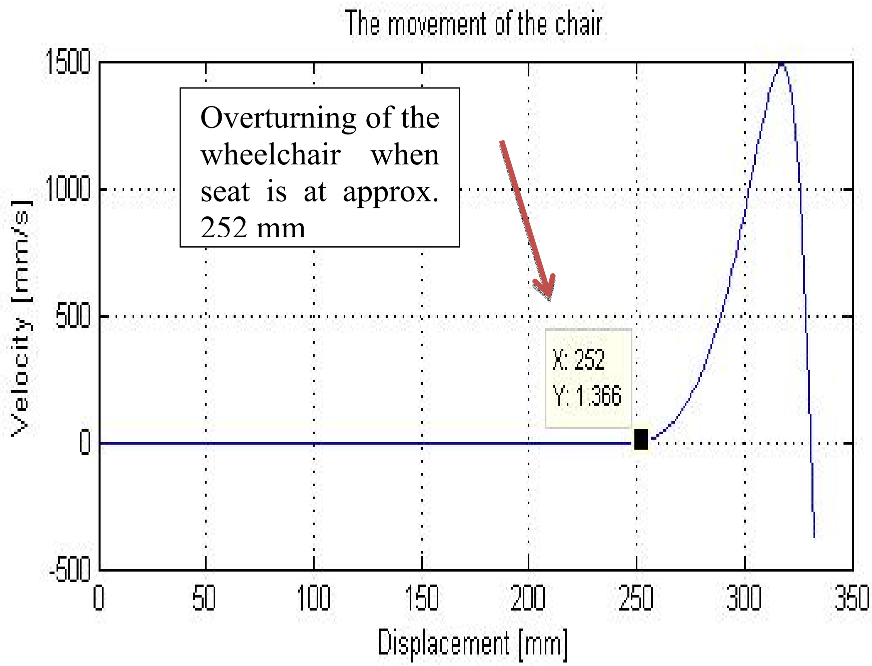

Figure5.19.The velocity of the big wheel

According to the model, the critical point of the seat is at point 252mm and the velocity is 1.366mm/s. This means that the seat position supposed to be always before this position to guarantee rest of the chair (See figures 5.20 and 5.21)

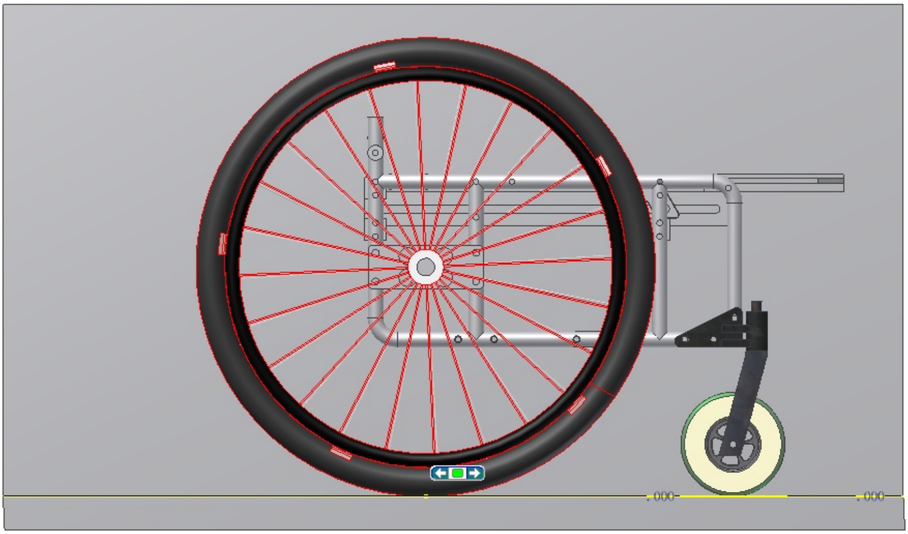

Figure5.20. the wheelchair is stable until 252 mm

The seat is stable now. (Figure 5.20)

If we set the seat displacement to 254mm the seat will be rotating.

The chair will be rotated as shown in Figure 5.22.

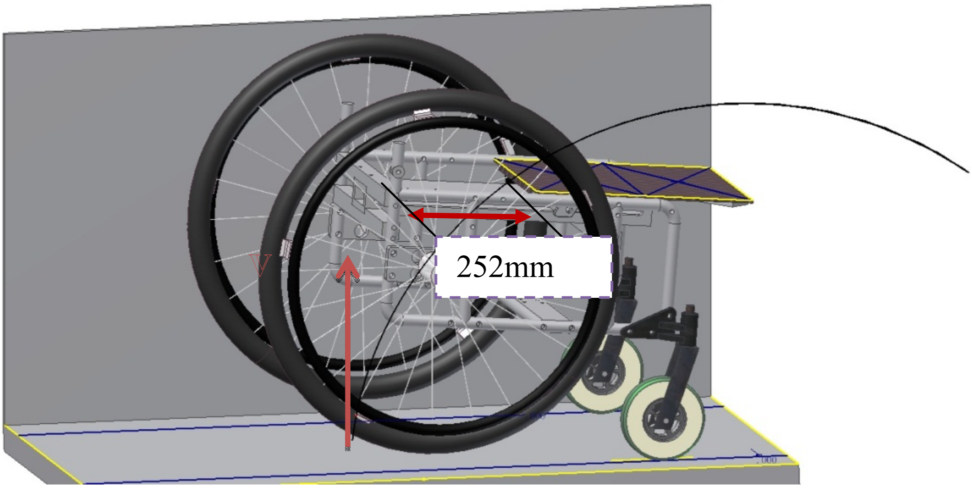

Figure5.21. The rotation starting moment of the wheelchair's velocity

We verify the previous results by fixing the seat at 254 mm. As a result, we noticed that the chair overturned inmediatly, as shown in figure 5.22

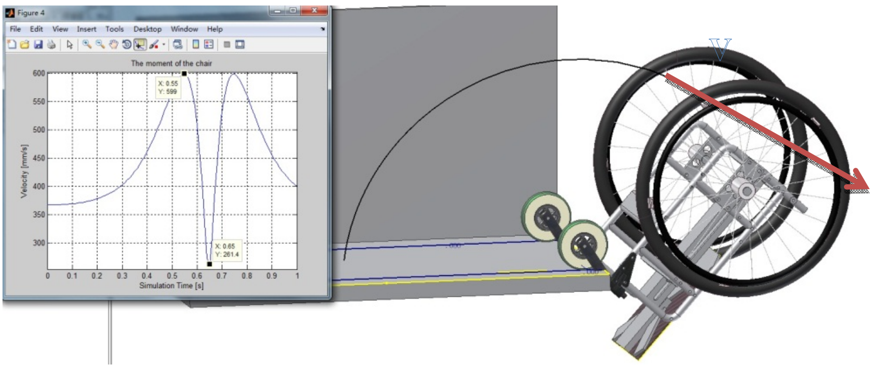

Figure5.22. Result of loading in the front.

Figure 5.23 shows the vertical velocity of the big wheel when the seat is at 254 mm from its original position

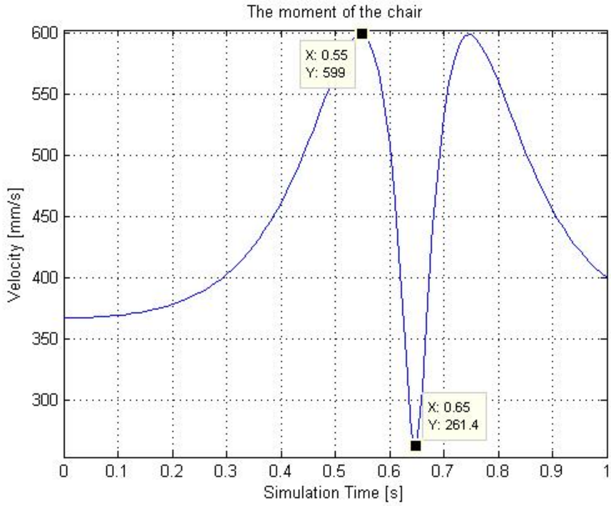

Figure5.23.Result of loading in the front

5.2.2 Load on rear half of the seat

Now we consider the case when the user is seated on the rear of the seat, (Figure5.24.)

As the previous case, the seat is sliding at constant speed during the simulation (Fig 5.25). The resulting velocity (in vertical direction) of the big wheel is shown in figure 5.26. According to this result, the wheel chair will overturn when the seat is at approximately 450mm from its original position. (Figure 5.27)

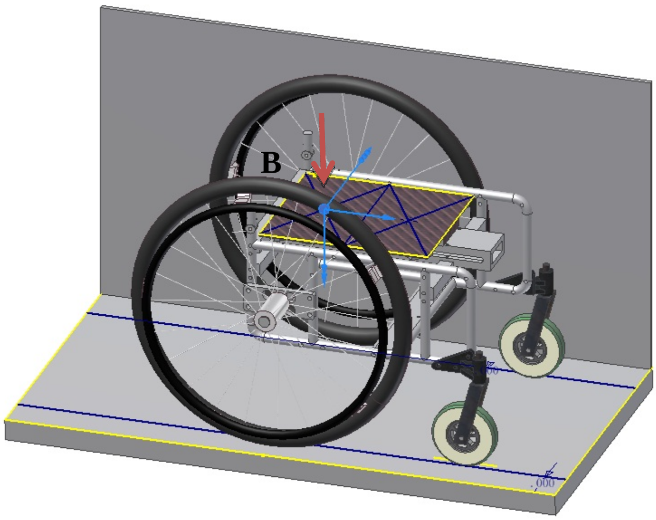

Figure5.24. Load position of seat adjustment dynamic simulation

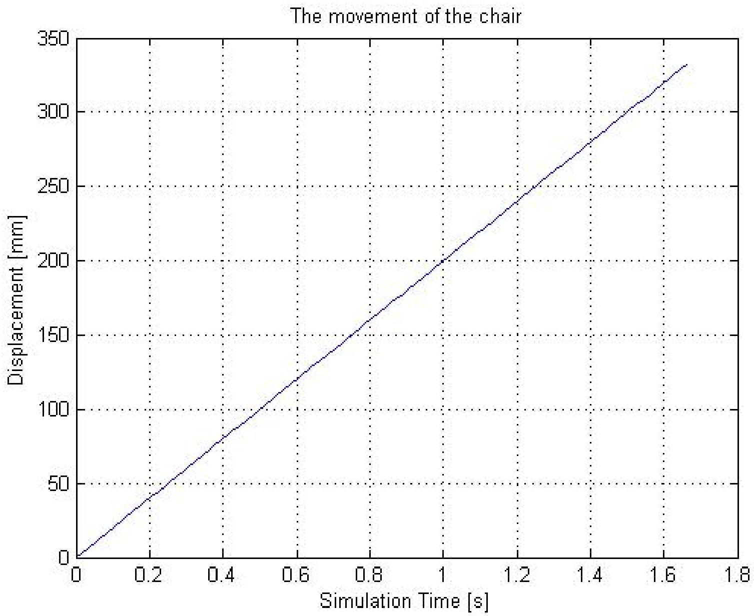

Figure5.25.The range of simulation time and displacement

Like what we have done in part I, we get results will be shown in the figures blow.

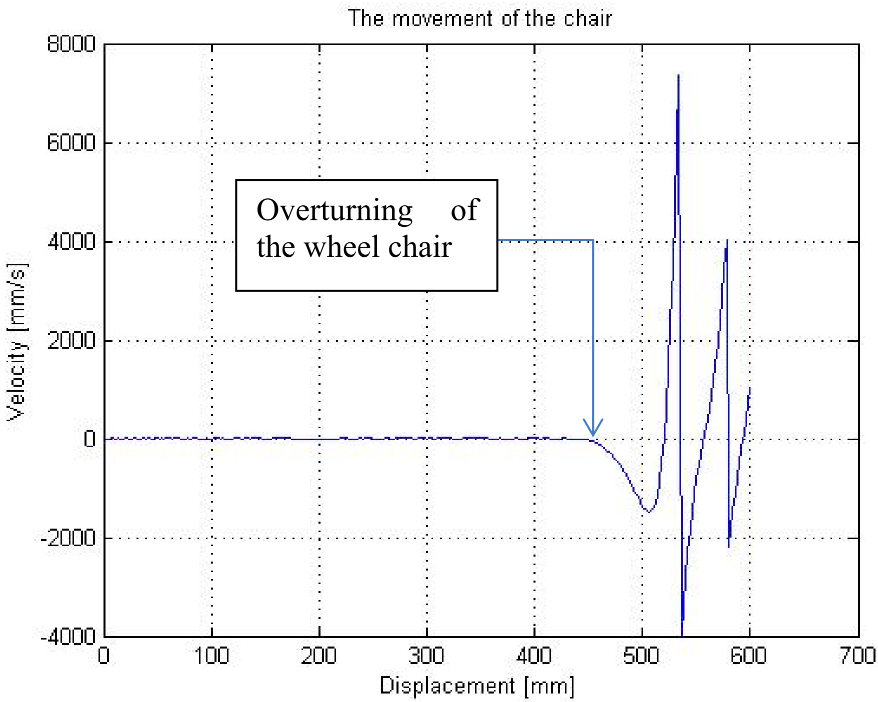

Figure5.26. the velocity of the big wheel and the displacement of the seat by loading at the rear of the seat

Now from figure we found between the two edges of the rail is the safety range. And the dangerous area is away from the forefront point.

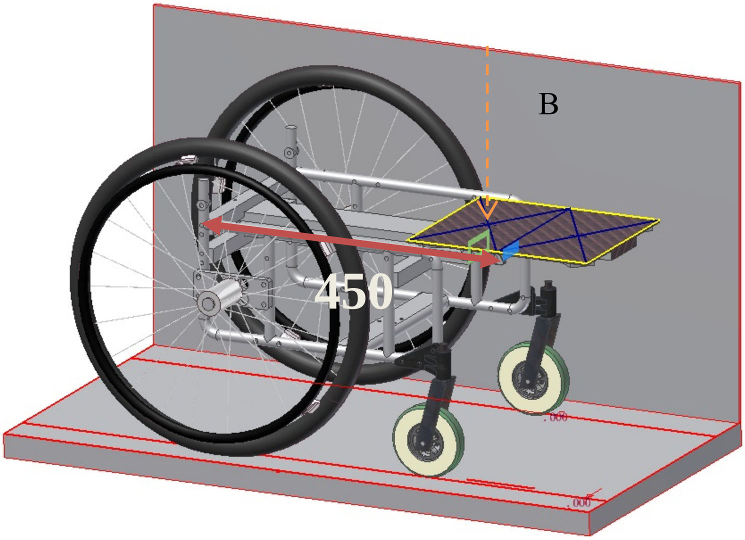

Figure5.27. Result of loading at rear of the seat.

6 Conclusions

This project is focus on modifying the manual wheel chair in order to make the user feel more comfortable and convenient. With the help of software Autodesk inventor we remodel the wheel chair and analysis it in both static and dynamic filed, the results we get are list as following:

- An optimization design was developed in CAD for the standard mechanical wheel chair.
- A sliding seat was designed to allow people moving from the wheel chair to any other seat.
- A reclining back rest was also introduced to offer more comfort for the user. This can recline up to 60.5 degrees.
- Two different models for the sliding seat were designed: dual & single rail function.
- Although the model with one rail is heavier than dual rail, but from the static stress analysis, it also showed to be safer, then, we consider it as our most convenient solution for the user.
- Dynamic analysis was also performed to determine a safe range of displacement for the sliding seat.
- According to the respective simulation results, the seat can move from its original position to 252 mm front without producing the overturning of the wheelchair.
- This range of displacement for the sliding seat represents a good enough distance for allowing the user to move from the wheel chair to any other seat.
- The optimization design shows significant improvements that can conveniently be introduced on the mechanical wheelchair in order to make it more useful and comfortable to the users.

7 Future works

We consider that there are some works need to be done in future about our suggested design. We mainly think that:

- To make a prototype and perform experimental test on it.
- After finished the prototype (figure 7.1), we found a few parts need to be modified.

Figure7.1 Prototype

- The wheels to keep balance
- The circuit still has some problem
- After discussed with the wheelchair user, they need a support on their back when they try to move.

- To focus on the design of the seat and back rest in terms of comfort

To incorporate electrical devices to move the slide seat and the back rest

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9 Appendixes

1. Result summary of dual rail in the front situation from Autodesk inventor

Name	Minimum	Maximum
Volume	4792360 mm^3	
Mass	12.9873 kg	
Von Mises Stress	0.000358548 MPa	63.8386 MPa
1st Principal Stress	-9.60003 MPa	59.5888 MPa
3rd Principal Stress	-61.0011 MPa	13.5743 MPa
Displacement	0 mm	1.66496 mm
Safety Factor	4.30774 ul	15 ul
Stress XX	-59.2133 MPa	52.304 MPa
Stress XY	-19.8064 MPa	17.6582 MPa
Stress XZ	-11.3729 MPa	21.2509 MPa
Stress YY	-30.4685 MPa	35.1848 MPa
Stress YZ	-26.0624 MPa	24.8618 MPa
Stress ZZ	-39.8903 MPa	38.8509 MPa
X Displacement	-0.18902 mm	0.190387 mm
Y Displacement	-0.00908187 mm	1.6549 mm
Z Displacement	-0.0655115 mm	0.0700374 mm
Equivalent Strain	0.00000000479083 ul	0.000828303 ul
1st Principal Strain	-0.00000443695 ul	0.000848065 ul
3rd Principal Strain	-0.00083261 ul	0.00000435296 ul
Strain XX	-0.000809363 ul	0.000732613 ul

Strain XY	-0.000382329 ul	0.000340862 ul
Strain XZ	-0.000219534 ul	0.000410213 ul
Strain YY	-0.000432762 ul	0.000461566 ul
Strain YZ	-0.000503091 ul	0.000479916 ul
Strain ZZ	-0.000342694 ul	0.000416394 ul
Contact Pressure	0 MPa	470.881 MPa
Contact Pressure X	-251.178 MPa	78.7192 MPa
Contact Pressure Y	-147.082 MPa	386.725 MPa
Contact Pressure Z	-95.2997 MPa	139.786 MPa

2. Result summary of single rail in the front situation from Autodesk inventor

Name	**Minimum**	**Maximum**
Volume	6742170 mm^3	
Mass	19.4902 kg	
Von Mises Stress	0.0000244995 MPa	24.9459 MPa
1st Principal Stress	-4.67513 MPa	16.7004 MPa
3rd Principal Stress	-21.3906 MPa	3.10833 MPa
Displacement	0 mm	0.215046 mm
Safety Factor	0 ul	15 ul
Stress XX	-9.48292 MPa	12.2199 MPa
Stress XY	-6.1951 MPa	6.38033 MPa
Stress XZ	-6.84762 MPa	5.81322 MPa
Stress YY	-15.5251 MPa	16.6773 MPa

Stress YZ	-6.66632 MPa	6.45388 MPa
Stress ZZ	-9.80505 MPa	8.17965 MPa
X Displacement	-0.0129752 mm	0.0167642 mm
Y Displacement	-0.00205596 mm	0.214556 mm
Z Displacement	-0.0506322 mm	0.0504508 mm
Equivalent Strain	0.000000000316235 ul	0.000323241 ul
1st Principal Strain	-0.00000236067 ul	0.000207698 ul
3rd Principal Strain	-0.000335127 ul	0.000000925423 ul
Strain XX	-0.0000890977 ul	0.000154349 ul
Strain XY	-0.000119586 ul	0.000123162 ul
Strain XZ	-0.000132182 ul	0.000112215 ul
Strain YY	-0.000221903 ul	0.000207252 ul
Strain YZ	-0.000128682 ul	0.000124581 ul
Strain ZZ	-0.000100674 ul	0.000119229 ul
Contact Pressure	0 MPa	362.273 MPa
Contact Pressure X	-250.5 MPa	213.133 MPa
Contact Pressure Y	-176.873 MPa	275.83 MPa
Contact Pressure Z	-22.6858 MPa	41.5783 MPa

3. Result summary of dual rail in the back situation from Autodesk inventor

Name	**Minimum**	**Maximum**
Volume	4792360 mm^3	
Mass	12.9873 kg	
Von Mises Stress	0.000312249 MPa	59.2784 MPa

1st Principal Stress	-9.30671 MPa	52.3456 MPa
3rd Principal Stress	-60.6254 MPa	7.56167 MPa
Displacement	0 mm	1.43241 mm
Safety Factor	4.63913 ul	15 ul
Stress XX	-51.1397 MPa	42.0725 MPa
Stress XY	-14.2327 MPa	32.596 MPa
Stress XZ	-13.7284 MPa	18.5782 MPa
Stress YY	-27.4169 MPa	31.6853 MPa
Stress YZ	-21.4803 MPa	26.3918 MPa
Stress ZZ	-46.5964 MPa	41.527 MPa
X Displacement	-0.161892 mm	0.161811 mm
Y Displacement	-0.0150423 mm	1.42408 mm
Z Displacement	-0.0690378 mm	0.0701892 mm
Equivalent Strain	0.00000000415612 ul	0.000765489 ul
1st Principal Strain	-0.00000754053 ul	0.000764274 ul
3rd Principal Strain	-0.00081961 ul	0.00000301787 ul
Strain XX	-0.000698947 ul	0.000602498 ul
Strain XY	-0.000274738 ul	0.000629211 ul
Strain XZ	-0.000265005 ul	0.000358621 ul
Strain YY	-0.000409878 ul	0.00045489 ul
Strain YZ	-0.000414641 ul	0.00050945 ul
Strain ZZ	-0.000548805 ul	0.000475804 ul
Contact Pressure	0 MPa	282.797 MPa
Contact Pressure X	-40.3231 MPa	133.515 MPa

Contact Pressure Y	-164.532 MPa	243.77 MPa
Contact Pressure Z	-159.199 MPa	75.0897 MPa

4. Result summary of single rail in the back situation from Autodesk inventor

Name	**Minimum**	**Maximum**
Volume	6742130 mm^3	
Mass	19.4901 kg	
Von Mises Stress	0.0000398702 MPa	16.1922 MPa
1st Principal Stress	-4.49075 MPa	14.3767 MPa
3rd Principal Stress	-18.9644 MPa	4.15509 MPa
Displacement	0 mm	0.149316 mm
Safety Factor	0 ul	15 ul
Stress XX	-9.27736 MPa	7.66109 MPa
Stress XY	-6.15328 MPa	4.66115 MPa
Stress XZ	-6.04079 MPa	6.71115 MPa
Stress YY	-17.4833 MPa	8.45355 MPa
Stress YZ	-3.71465 MPa	3.9307 MPa
Stress ZZ	-9.5906 MPa	9.18115 MPa
X Displacement	-0.00907488 mm	0.0191568 mm
Y Displacement	-0.00124956 mm	0.14816 mm
Z Displacement	-0.0240956 mm	0.0243274 mm
Equivalent Strain	0.000000000526636 ul	0.000215664 ul
1st Principal Strain	-0.0000027474 ul	0.000189648 ul

3rd Principal Strain	-0.000247476 ul	0.00000429265 ul
Strain XX	-0.000112366 ul	0.0000789399 ul
Strain XY	-0.000118779 ul	0.0000899758 ul
Strain XZ	-0.000116607 ul	0.000129548 ul
Strain YY	-0.000232775 ul	0.000105542 ul
Strain YZ	-0.0000717051 ul	0.0000758757 ul
Strain ZZ	-0.0000976662 ul	0.0000888431 ul
Contact Pressure	0 MPa	157 MPa
Contact Pressure X	-116.935 MPa	98.4206 MPa
Contact Pressure Y	-88.7889 MPa	117.789 MPa
Contact Pressure Z	-13.6019 MPa	32.0458 MPa

School of Engineering, Department of Mechanical Engineering
Blekinge Institute of Technology
SE-371 79 Karlskrona, SWEDEN

Telephone: +46 455-38 50 00
E-mail: info@bth.se