CHALMERS
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Development of virtual method for prediction of handling forces using Adams

Master's thesis in the Applied Mechanics Master Degree Program

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Abstract

The master thesis was performed at the department of Applied Mechanics at Chalmers University of Technology, in cooperation with Volvo Car Corporation and MSC Software, with the goal to develop a method for virtual prediction of handling forces using Adams.

When developing a car, well-adjusted handling forces for different functions such as forces for door closing, seat adjustment and hood release, are very important. Usually, these handling forces are verified physically and since Volvo Car Corporations strategy is to decrease dependency of verification through physical testing, handling forces must be verified virtually during development phases. The master thesis will cover the method for virtual verification of the force needed to pull up the handle for adjustment of the back, for a rear seat system.

When modeling the seat in Adams, the objective was to resemble the seats mechanical mechanisms as close to reality as possible. The model was therefore constructed in three different phases, where each phase gradually resembles reality more accurately. Phase one of the modeling was the most simplified one. Its main purpose was to achieve the correct motions of the seat’s different parts and thereby assure that the parts affected by the applied force were moving correctly. In phase two, a more advanced implementation of constraints and forces was performed and thus resembled the actual seat motion in a more accurate way. Finally in phase three, the seat’s parts were made flexible i.e. made into non-rigid parts, which allowed for possible deformations. By performing this transformation, the final phase represented a model that practically resembles reality.

When the model was built in Adams, it was possible to measure the handle force and also to perform a contribution analysis to find the parameter that contributed the most to the magnitude of the handle force. By identifying this parameter, a redesign was performed in order to evaluate the possibility of achieving a lower handle force. Geometrical variations were also implemented to examine how possible canting of the parts affects the magnitude of the force.

The simulations showed that the needed handle force was 73-90 N compared to Volvo Car Corporations physical tests which were 70-120 N. From the contribution analysis it was possible to determine that the recliner affected the handle force the most and a redesign of it resulted in a lower value. Implementation of geometrical variations showed an increase of the handle force magnitude.

From the simulations, it can be concluded that it is possible to model handling forces virtually and obtain good results. This means that there is potential in eliminating the experimental need for measuring handling forces and in turn decrease dependency on physical product testing.

Keywords: *Handling Force, Handle Force, Adams, Adams/Insight, Contribution Analysis, Geometrical Variation, Virtual, Finite Element Analysis, CATIA, Simulation.*

Preface

This master thesis was made with the goal to develop a method for virtual prediction of handling forces using Adams. The master thesis was carried out at the department of Applied Mechanics at Chalmers University of Technology, in collaboration with Volvo Cars Corporation and MSC Software, during the spring 2015.

The authors would like to thank everyone that has been involved in the master thesis. Special thanks to our supervisor Casper Wickman at Volvo Car Corporation, Fredrik Sjögren at MSC Software, Johan Elfström at Volvo Car Corporation and our examiner Anders Boström at Chalmers University of Technology. Also thanks to the group 91300 at Volvo Cars Corporation and the whole staff at MSC Software for welcoming and supporting us throughout the master thesis.

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Gothenburg, May 2015

Contents

Terms and Abbreviations

Adams	Automatic Dynamic Analysis of Mechanical Systems. A widely used multibody dynamics software.
Adams/Insight	A software used to study variations and optimize a design.
Brose	A German automotive supplier.
CAD	Computer Aided Design.
CATIA	Computer Aided Three-dimensional Interactive Application.
DOF	Degree Of Freedom
FEA	Finite Element Analysis
MSC Nastran	A FEA program
OEM	Original Equipment Manufacturer
VCC	Volvo Car Corporation
SimXpert	A preprocessor which allows the user to create a mesh on the parts which is necessary before running a FEA.

1. Introduction

This report describes the master thesis *Development of virtual method for prediction of handling forces using Adams,* which is performed at Chalmers University of Technology in cooperation with Volvo Car Corporation and MSC Software. The report covers the background, theory, method, results, discussion, conclusions of the project and recommendations for future work.

1.1 Background

Volvo Car Corporation (VCC) is an automotive Original Equipment Manufacturer (OEM) that acts in the premium segment [1]. One very important user aspect of a vehicle is consistent and well-adjusted handling forces for different functions of the product such as forces for door closing, seat adjustment and hood release. Handling forces are requirement-set by the technical product attribute Craftsmanship and Ergonomics at Vehicle Engineering [10].

Traditionally, handling forces are verified physically during test series. Since VCC strategy is to decrease dependency of physical product verifying tests, handling forces must be verified virtually during development phases. The master thesis will cover the method for virtual verification of a handling force on a rear seat system. The studied handling force will be referred to as handle force throughout this master thesis report.

1.2 Purpose

The purpose of this master thesis is to develop a method for virtual verification of handling forces for a rear seat system using primarily Adams, which is a software used for multibody dynamic simulations [2]. Understanding whether geometrical variations should be considered for the evaluated system will also be investigated. This will result in eliminating the experimental need for measuring handling forces and in turn decrease dependency of physical product testing.

1.3 Boundaries

The boundaries for this project are:

- The time limit is set to 20 weeks.
- The analysis will only be performed on the rear seat of the Volvo XC90.
- Suggestion for potential improvements should be made but no optimized redesign will be performed.
- The analysis will primarily be performed on a rigid system.
- Foam on the rear seat will be neglected and the simulations will only be performed on the seat's frame.

1.4 Problem definition

The main questions that need to be resolved are:

- How do the results from the developed Adams model differentiate with the results from VCC physical testing data?
- Which are the most contributing parameters given from the contribution analysis in Adams?

- Should geometrical variations be incorporated as a parameter in simulation if obtained results cannot be motivated by existing contributing parameters?
- Does non-rigid simulation have to be conducted?
- Which potential improvements can be suggested?

2. Case description and theory

This section describes the necessary descriptions and theory used throughout the project. This chapter covers the seat definitions, degrees of freedom, constraints, forces and non-rigid deformation.

2.1 Seat definitions

In this chapter the seat system's mechanical mechanisms are described.

Figure 1. The seat viewed in a 3D enviroment in the software Adams.

The seat consists of several hundred parts and can be viewed in a 3D environment in figure 1. When describing the seat function, the main focus lies on the mechanical parts which are activated when the user wants to adjust his seat position, more precisely the back of the seat. These mechanical parts are better viewed in figure 2.

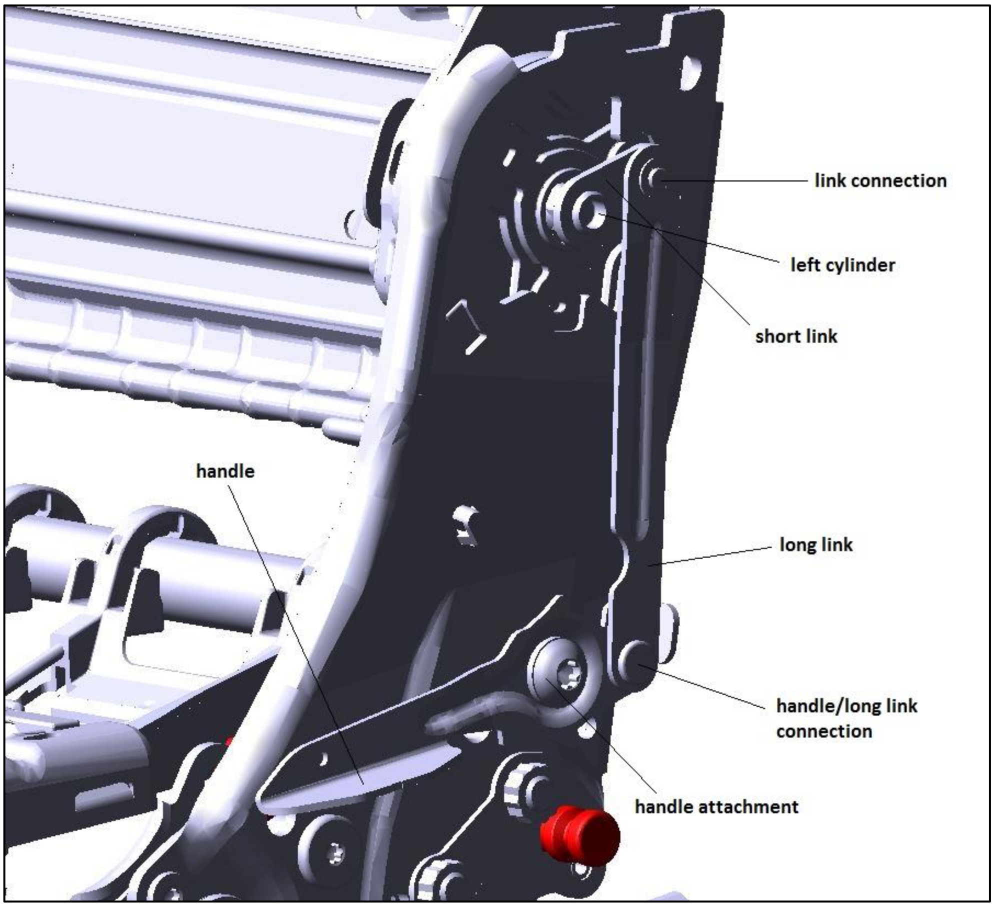

Figure 2. A more detailed view of the parts in the mechanical seat adjustement system.

If the user wishes to adjust the seat position, an upward force on the part referred to as the *Handle* which can be seen in figure 2 must be applied. When the *Handle* is lifted, a rotation is created between the *Handle* and the part referred to as the *Handle attachment* which is fixed in the lower part of the seat. This causes the opposite side from where the vertical force was applied i.e. where there is a connection, referred to as the *Handle/Long link connection*, between the *Handle* and the part referred to as the *Long link* to move downwardly which can be seen in figure 3. This leads to a rotation between the *Handle* and the *Long link* that in turn is fixed to the *Handle/Long link connection*. Due to these movements, friction forces will be obtained between the *Handle* and the *Handle* attachment, the *Handle* and the *Long link* and between the *Handle* and the *Handle/Long* link connection.

Figure 3. Movement of the seat´s parts when an upward force is applied on the *Handle*.

The movement continues with rotation of the part referred to as the *Short link*, which is connected to the *Long link*, due to the previous mentioned motion of the *Long link*. Here, the *Short link* is free to rotate around the *Link connection* while the *Long link* is fixed to the *Link connection* which prevents any rotation between these two. When the *Short link* rotates, it forces the part referred to as the *Left cylinder* to rotate simultaneously due to that the *Short link* is fixed to the *Left cylinder*. Rotation in the *Left cylinder* leads to three *Sliders* that are placed in the blocks on the part referred to as the *Left roll*, also known as the *Left recliner*, which has been highlighted in figure 6, to release. The parts in the *Recliner* can be seen in figure 5. All the parts in the *Recliner* serve different functions, but what is most important to notice is that when the *Left cylinder* is rotated, the *Steering ring* rotates, which in turn forces the *Sliders* to move inward. This is better viewed in figure 4, where it is easy to see that the shape of the curvature at the connection determines how the *Sliders* move. As the *Steering ring* rotates in a clockwise direction, this shape forces the *Sliders* inward. When the *Sliders* are released i.e. when the *Recliner* opens, the connection between the grooves will separate and the *Left roll* will be able to rotate. Once the *Handle* is released, the *Closure springs* forces the *Sliders* outward and in turn locks the *Recliner* [13]. These movements create friction between the *Long link* and the *Short link*, the *Short link* and the *Link connection* and between the parts in the *Left recliner*.

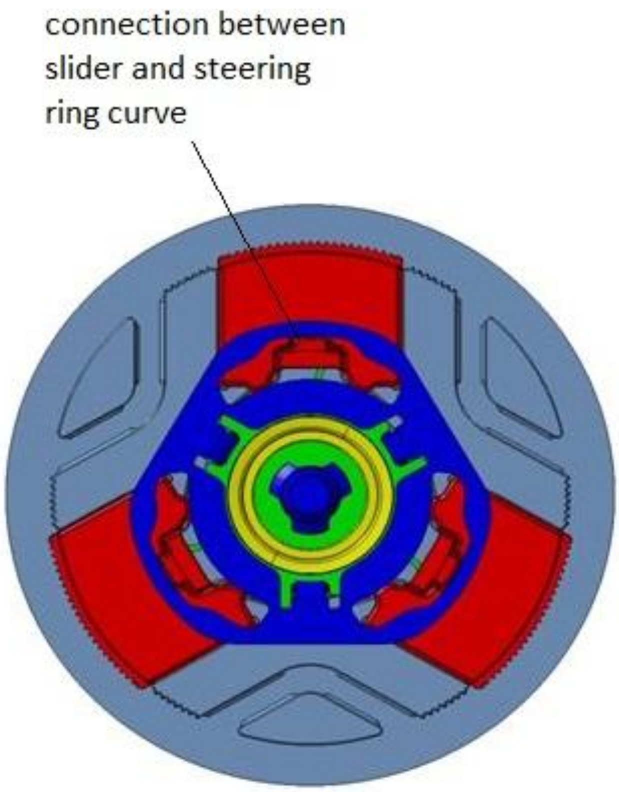

Figure 4. Connection between the three *Sliders* and the *Steering ring* [3].

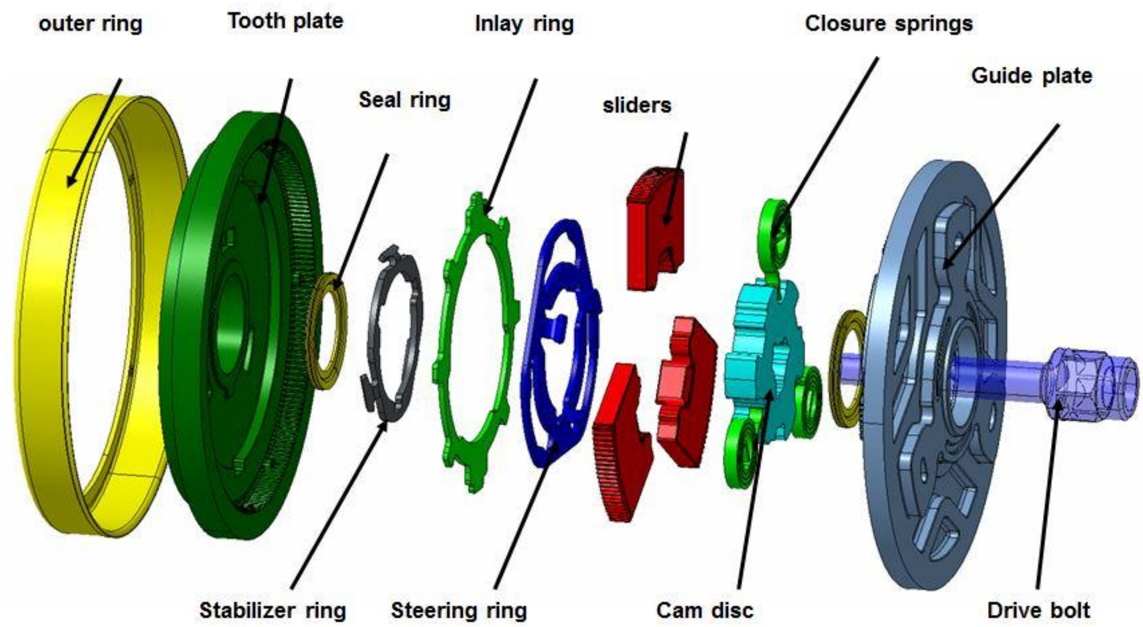

Figure 5. The complete detailed system in a *Recliner* [3].

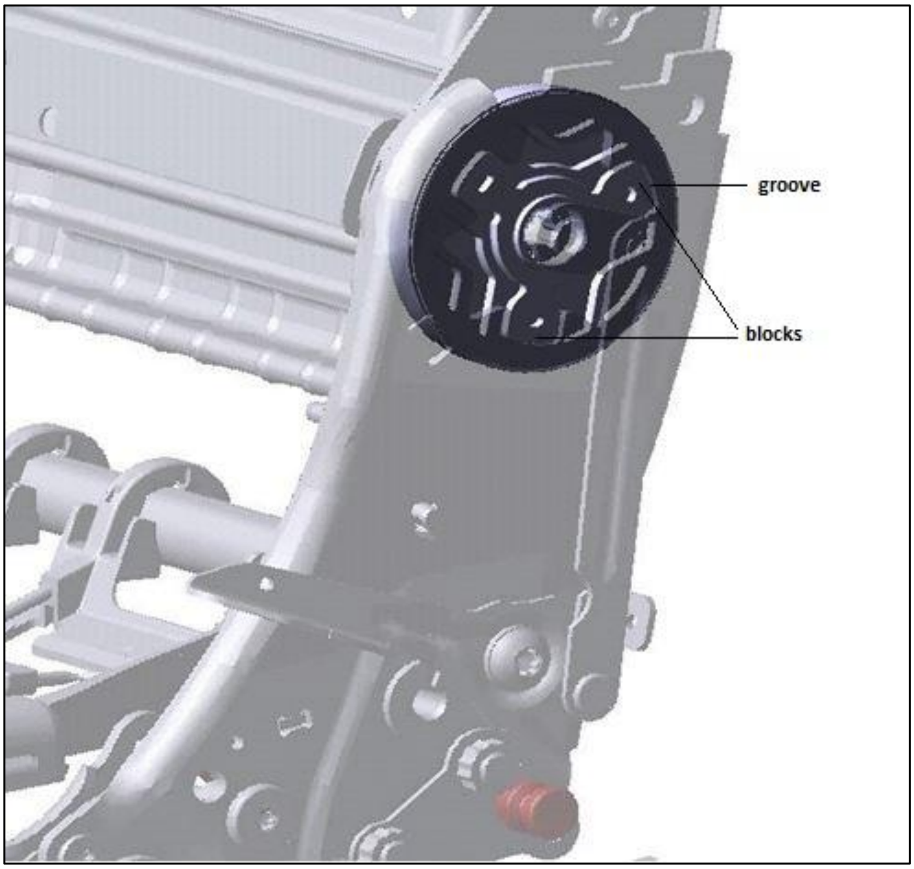

Figure 6. The left *Recliner* highlighted.

Lastly the *Left cylinder* is through the part referred to as the *Left wire holder*, linked to the part referred to as the *Wire*, which also is linked across the other side of the seat to the part referred to as the *Right wire holder*. The *Wire* and *Right wire holder* allows the *Left cylinder* to rotate simultaneously as the part referred to as the *Right cylinder* rotates with some transmission losses. This can be seen in figure 7. When the *Right cylinder* rotates, the same mechanisms operate as previously explained for the *Left cylinder*, making it possible for the user to adjust the back of the seat.

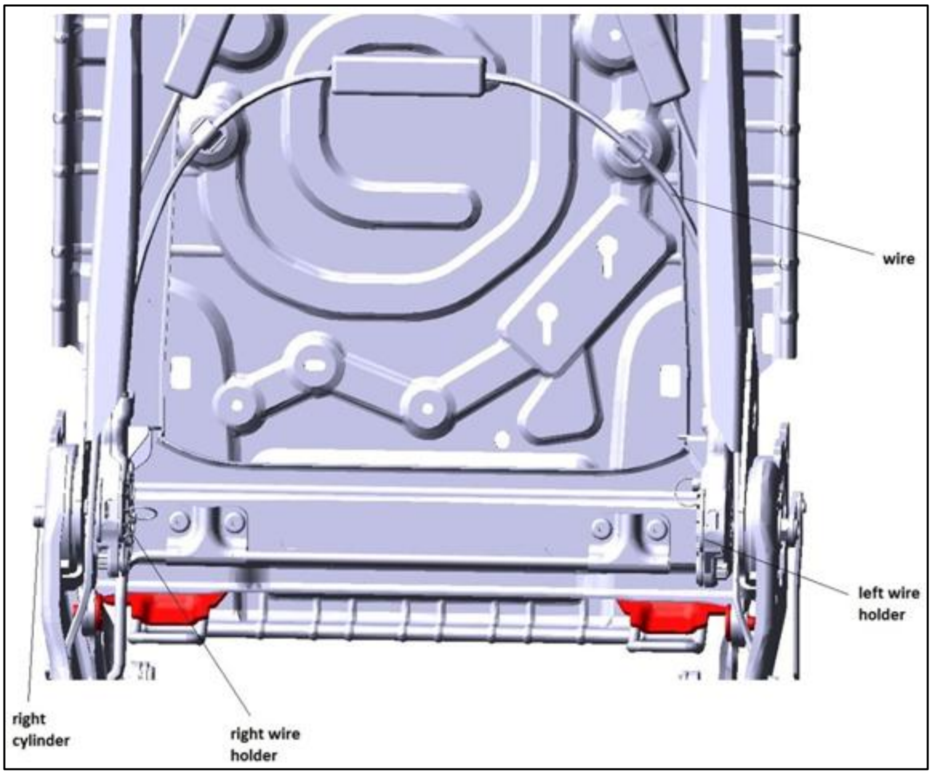

Figure 7. A front view of the seat.

The seat is held upwards due to a *Torsional spring* which is attached between the back and lower part of the seat. This can be seen in figure 8. The *Torsional spring* contains a preload which makes the back seat rise up automatically from a downward position when the user lifts up the handle [13].

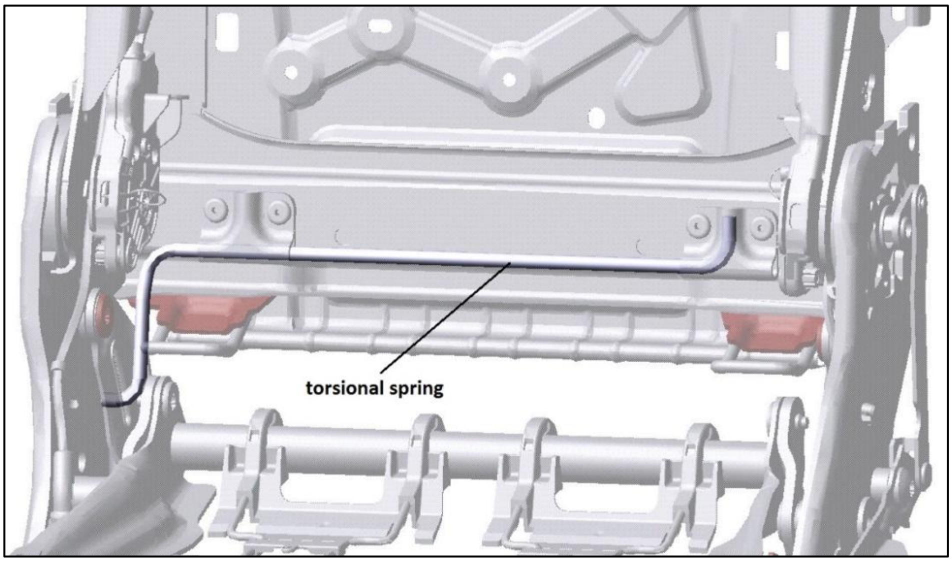

Figure 8. *Torsional spring* highlighted.

2.2. Degrees of freedom

In mechanics, the Degrees Of Freedom (DOF) define the possible movements of a solid body. For an unconstrained solid body in space, the numbers of DOF are six. These are divided into three translational and three rotational degrees of freedom [4]. The body can either be considered as rigid or non-rigid. Rigid meaning that deformation of the body is neglected, whereas for the non-rigid case the deformation is considered.

By applying constraints to the solid body, different DOF can be removed and in turn corresponding movements, meaning that the solid body will not be able to translate and/or rotate in the direction of the removed DOF. In a mechanical system, including several solid bodies, the goal is to neither under- or over- constrain the system. This means that a sufficient amount of DOF should be constrained to obtain the desired movement of the system, while at the same time not constraining the same DOF twice or more.

2.3 Constraints in Adams

A constraint is a type of restriction. By applying a constraint to a body, it is possible to determine and limit a body's movement. There are several types of constraints, each implying a type of restriction on a body. In Adams these constraints are imposed by applying idealized or primitive joints, described below, to one or several bodies[5]. In this section the different types of joints used throughout the project will be explained briefly.

2.3.1 Idealized joints

Idealized joints are mathematical representations of joints that have physical counterparts, for example a revolute joint, seen in figure 9, representing a hinge. The idealized joints are used to connect parts, where the parts can be either rigid- or non-rigid bodies.

2.3.1.1 Revolute joints

A revolute joint connects two parts by allowing rotation of the parts with respect to each other around a common axis, similar to a hinge as can be seen in figure 9. This allows for rotation around one axis, meaning that one part only has one degree of freedom relative the other.

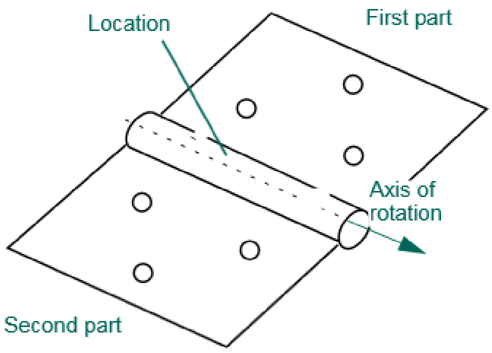

Figure 9. A hinge, which works in the same way as a revolute joint [5].

2.3.1.2 Translational joints

A translational joint allows one part to move with respect to another part along a specified axis, seen in figure 10. This allows for translation in one direction meaning that the parts have one degree of freedom relative one another.

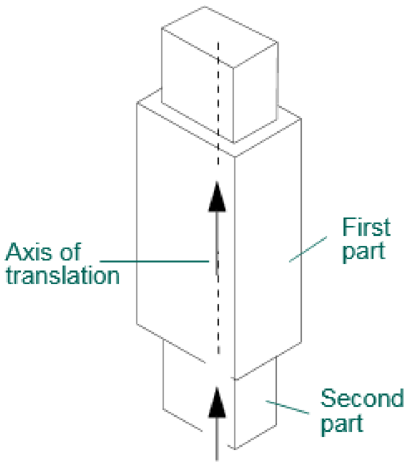

Figure 10. Movement of two parts with respect to one another with a translational joint [**5**]**.**

2.3.1.3 Cylindrical joints

A cylindrical joint allows one part to both translate and rotate with respect to another part, seen in figure 11. The parts have two degrees of freedom with respect to one another.

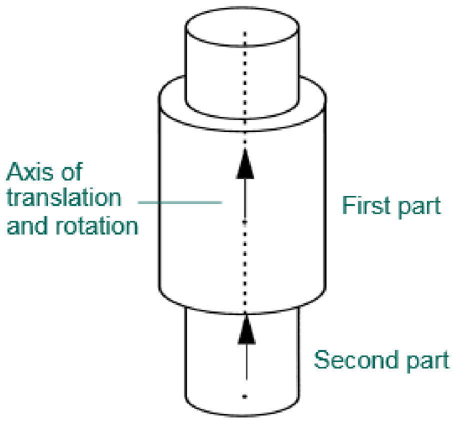

Figure 11. Movement and rotation of two parts with respect to one another with a cylindrical joint [**5**]**.**

2.3.1.4 Spherical joints

A spherical joint allows one part to rotate around all three axes with respect to another part at a common point, as can be seen in figure 12. This joint removes the translational degrees of freedom meaning that one part has only three degrees of freedom relative the other.

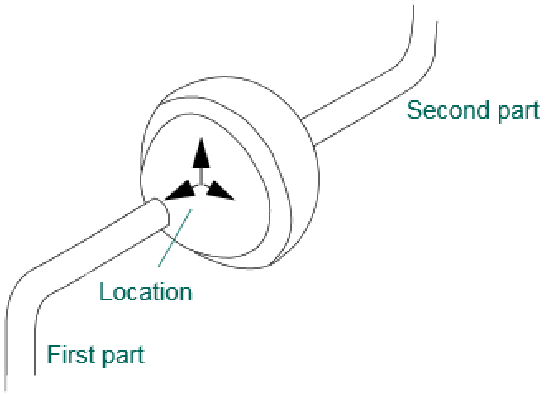

Figure 12. A spherical joint allows one part to rotate around all three axes with respect to another part at a common point [5].

2.3.1.5 Fixed joints

A fixed joint connects two parts together by locking them to each other at a common point, seen in figure 13. The parts are then parts of the same rigid body.

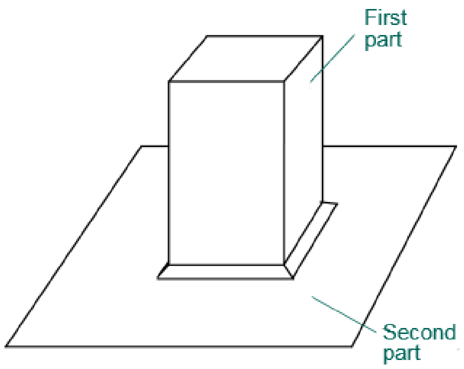

Figure 13. A fixed joint can be used to join two parts [5].

2.3.1.6 Degrees of freedom removed by Idealized joints

A summation of the degrees of freedom removed by applying the joints in Adams is given in table 1.

Table 1. Summation of the DOF removed by applying different joints in Adams [5].

		Rotational DOF Removed			
		0	1	2	3
Translational DOF Removed	0	(Part)			
	1				
	2			Cylindrical	Translational
	3	Spherical		Revolute	Fixed

2.4. Forces in Adams

With forces it is possible to define loads and compliances on parts. Forces do not primarily prescribe motion, meaning that they do not remove degrees of freedom as the joints explained above. Though forces can both resist and induce motion on parts [5]. One of the more frequently used forces throughout the project is bushings. Below a brief explanation of a bushing is described.

2.4.1 Bushings

Bushings are in Adams a type of flexible connectors used to regulate motion of parts. A bushing is defined by forces and torques between two parts. The forces and torques are calculated from the following mathematical expression, seen below as equation 1 [5].

$$\begin{bmatrix} F_x \\ F_y \\ F_z \\ T_x \\ T_y \\ T_z \end{bmatrix} = -\begin{bmatrix} K_{11} & 0 & 0 & 0 & 0 & 0 \\ 0 & K_{22} & 0 & 0 & 0 & 0 \\ 0 & 0 & K_{33} & 0 & 0 & 0 \\ 0 & 0 & 0 & K_{44} & 0 & 0 \\ 0 & 0 & 0 & 0 & K_{55} & 0 \\ 0 & 0 & 0 & 0 & 0 & K_{66} \end{bmatrix} \begin{bmatrix} x \\ y \\ z \\ a \\ b \\ c \end{bmatrix} - \begin{bmatrix} C_{11} & 0 & 0 & 0 & 0 & 0 \\ 0 & C_{22} & 0 & 0 & 0 & 0 \\ 0 & 0 & C_{33} & 0 & 0 & 0 \\ 0 & 0 & 0 & C_{44} & 0 & 0 \\ 0 & 0 & 0 & 0 & C_{55} & 0 \\ 0 & 0 & 0 & 0 & 0 & C_{66} \end{bmatrix} \begin{bmatrix} V_x \\ V_y \\ V_z \\ w_x \\ w_y \\ w_z \end{bmatrix} + \begin{bmatrix} F_1 \\ F_2 \\ F_3 \\ T_1 \\ T_2 \\ T_3 \end{bmatrix} \quad (1)$$

Viewing the expression from left to right, the forces and torques are represented in the first matrix, stiffness is represented in the second matrix and deformations are represented in the third matrix, where x, y and z are translational deformations and a, b and c are rotational deformations. Damping is represented in the fourth matrix and velocity is represented in the fifth matrix, where V_x, V_y, V_z are the velocities in the x, y and z direction and W_x, W_y, W_z are the angular velocities. Lastly the constant preloads are represented in the sixth matrix. By defining the bushings' stiffness and damping in Adams, the forces and torques between the parts can be calculated. Depending on the magnitude of these, the amount of translation and rotation of the part can be restricted.

2.5 Non-rigid deformation

When working with non-rigid systems, deformation of the parts involved is considered. The deformation is taken care of by adding deformations in the following form, seen below as equation 2.

$$u = \sum_{i=1}^{M} \emptyset_i q_i \quad (2)$$

where M is the number of mode shapes, $\emptyset_i$ is the mode shapes and q_i is the amplitude of the mode shape. Using the Craig-Bampton method, as described in [6], the mode shapes, $\emptyset_i$ can be obtained.

The known load vector together with the defined stiffness, damping and mass matrix are used to obtain the amplitude of the modes, q_i through iteration. Finally the mode shapes and amplitudes are summarized with the equation above to obtain the deformation. A simplified illustration of this modal superposition can be seen in figure 14.

Figure 14. A typical modal superposition [6].

The theory explained above is a brief simplified explanation of non-rigid deformation. The calculations behind the mode shapes and amplitudes are far more advanced. If more profound explanations are desired see [6].

3. Method

This section describes the methods used throughout the project. It covers the modeling and analysis of the seat in Adams, contribution analysis, implementation of geometrical variations and redesign of the most contributing parameter.

3.1 Building the seat model in Adams

In the previous chapter, 2.1 *Seat definitions*, the seat system's mechanical mechanisms were described. When modeling the seat in Adams, the objective was to resemble these mechanical mechanisms as close to reality as possible. To achieve this, some simplifications had to be made. The model was therefore constructed in three different phases, where each phase gradually resembled reality more accurately. In the upcoming sections, these three phases will be carefully explained, but first a brief summary of each phase will be given.

Phase one of the modeling was the most simplified one. Its main purpose was to achieve the correct motions of the seat's different parts when loading the handle with an upward force. In other words, phase one was made to assure that the parts affected by the applied force were moving correctly. The modeling was done with the help of joints, which were explained in the previous chapter 2.3 *Constraints in Adams*.

In phase two of the modeling, the joints that had been used in phase one were replaced with bushings. By using bushings the system will behave in a much more correct way than in the previous phase. This is due to that bushings allow for rotation and translation in all directions between parts and this resembles the actual seat motion in a more accurate way. Joints on the other hand completely remove specific degrees of freedom, which is unlike a real case scenario. In phase two the friction that exists between parts, the friction loss in the *Wire* and the moment needed to open the *Recliner* was also taken into account. This was made with the help of data obtained from subcontractor Brose [12].

In the last phase, phase three, the seat's parts were made flexible i.e. made into non-rigid parts which was explained in the chapter 2.5 *Non-rigid deformation*. By performing this transformation, the final phase represents a model that practically resembles reality.

3.1.1. Phase one of modeling

As explained above, a simplified model was first created to verify the movement of the parts when the *Handle* is pulled upwards. The first step towards creating a simplified model was to import the geometry into Adams. As mentioned in chapter 2. *Seat definitions*, the CAD model consisted of several hundred parts and since it would be very time consuming and computationally demanding to model all the parts in Adams, a simplification had to be made. This was done by identifying which of the parts that were connected to each other and also which of the parts that were moving as a unit. By doing so, the several hundred parts could be grouped together and narrowed down to eleven parts. These parts, seen in figure 15, could then much easier be constrained to each other. The definitions of the different parts are brought up again below to help the reader follow the different phases.

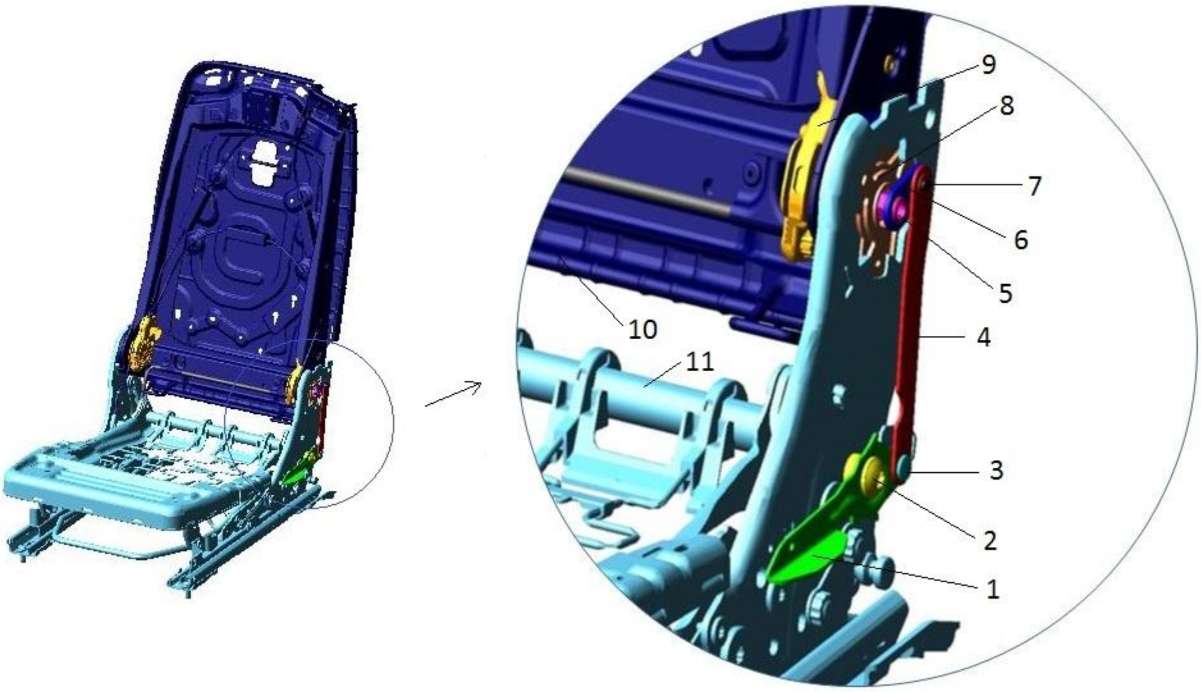

Figure 15. The seat divided in eleven different colors and numbers. Each color represents one of the reduced eleven parts.

Table 2. The name, color and number of every part.

	Name		Name
Part 1	Handle (Green)	Part 7	Link connection (Black)
Part 2	Handle attachment (Yellow)	Part 8	Left roll (Peach)
Part 3	Handle/Long link connection (Cyan)	Part 9	Left wire holder (Maize)
Part 4	Long link (Red)	Part 10	Seat back (Purple)
Part 5	Left cylinder (Magenta)	Part 11	Seat under (Sky blue)
Part 6	Short link (Blue)		

Before proceeding to constrain the model, the different parts had to be assigned with a mass. It was assumed that the whole seat was made of steel and therefore a density of $7800\ \mathrm{kg/m^3}$ was assigned to all parts. By assigning a density, Adams automatically calculates mass and moments of inertia for the different parts and center of gravity location.

The first step was to fix one of the parts in space, in our case the *Seat under*. As can be seen in figure 16, the *Handle attachment* is connected to the *Seat under* and therefore a fixed joint was placed between these two parts. When the *Handle* is pulled upwards, it rotates around the *Handle attachment*, hence a revolute joint was placed between the *Handle* and the *Handle attachment*, allowing the *Handle* to rotate around the common axis. As explained before, the *Handle* also rotates around the *Handle/Long link connection,* while the *Long link* is fixed to this connection. In the chapter 2.2 *Degrees of freedom*, it was mentioned that in a mechanical system, the goal is to neither under- or over constrain a system. In our case this means that a revolute joint cannot be placed between the *Handle* and the *Handle/Long link connection*, since this would over constrain the system. Therefore a cylindrical joint is placed here instead. The cylindrical joint allows for rotation around a desired axis while also allows for translation in a desired direction. Since the revolute joint, used between the *Handle* and the *Handle attachment* removes all degrees of freedom except one rotational, the parts are not allowed to translate. Therefore the cylindrical joint now only allows for rotation between the *Handle* and the *Handle/Long link connection,* as desired.

Moving on, the *Long link* is attached to the *Link connection* with a fixed joint, while the *Short link* is allowed to rotate around this connection. To prevent over constraining the system a spherical joint is placed between these two parts. Since the other parts have been restricted to no translation and rotation around one specific axis, the spherical joint satisfies our desired movement of the *Short link* around the *Link connection*. At the other side of the *Short link*, a fixed joint is placed between the *Short link* and the *Left cylinder*. In this way the *Left cylinder* rotates when the *Short link* moves as was previously explained. To ensure that the *Left cylinder* stays in the horizontal position as it does in reality, it is locked with a revolute joint. The *Left cylinder* is lastly connected to a cylinder at the right side of the seat with a wire. Since the *Wire* was not included in the CAD model, it was not possible to model it. To model that the *Right cylinder* rotates at the same time as the *Left cylinder*, a coupler was applied in Adams instead. The coupler allows two parts to be connected by specifying how one part moves as the other part moves. In this way it was possible to ensure that the *Right cylinder* rotates when the *Left cylinder* rotates. It should be mentioned that there is a loss in the *Wire* due to friction, which also should be taken into consideration.

Before performing a dynamic simulation, the remaining parts were connected with fixed joints at the positions specified in figure 16. The mechanism when pulling up the *Handle* was now modeled and it was possible to ensure that the movements of the parts were as expected.

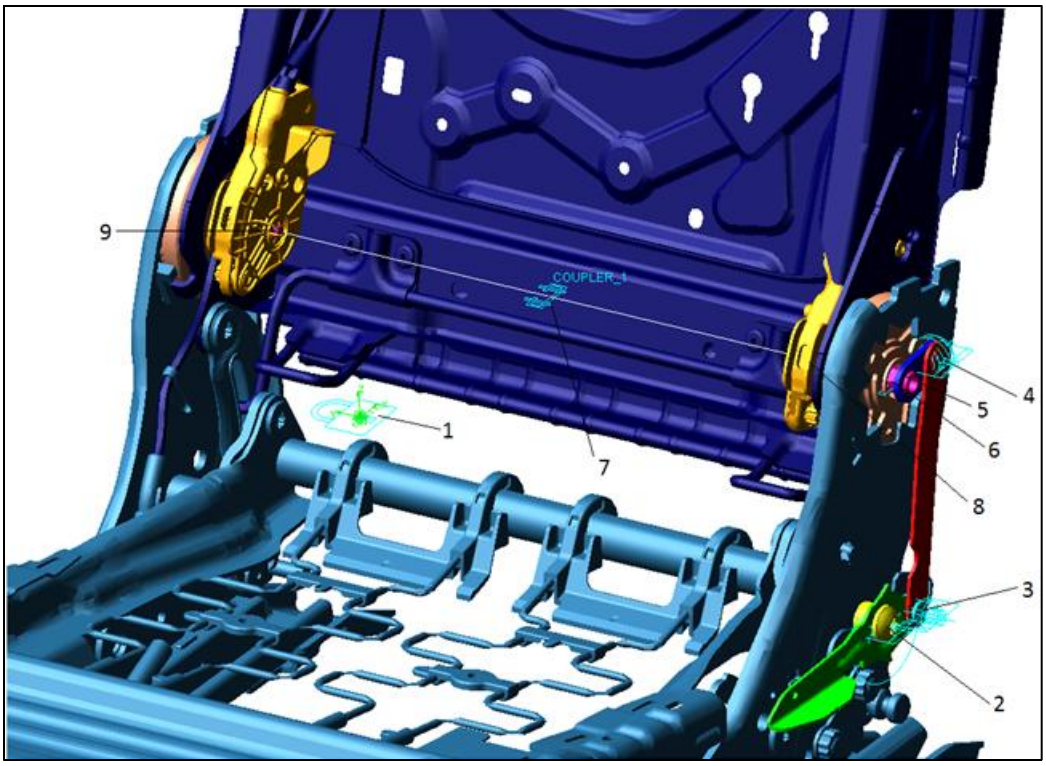

Figure 16. Placement of the introduced joints and what number they have been assigned.

Table 3. The joint number, type of joint and between which parts the joint has been placed. The table also shows which reference number the joints have been given in figure 8.

	Part 1	Part 2	Type of joint	Reference number
Joint 1	Seat under	Ground	fixed joint	1
Joint 2	Handle attachment	Seat under	fixed joint	2
Joint 3	Handle	Handle attachment	revolute joint	2
Joint 4	Long link	Handle/Long link connection	fixed joint	3
Joint 5	Handle	Handle/Long link connection	cylindrical joint	3
Joint 6	Long link	Link connection	fixed joint	4
Joint 7	Short link	Link connection	spherical joint	4
Joint 8	Short link	Left cylinder	fixed joint	5
Joint 9	Left cylinder	Ground	revolute joint	6
Joint 10	Seat under	Left roll	fixed joint	7
Joint 11	Seat back	Left roll	fixed joint	8
Joint 12	Seat back	Left wire holder	fixed joint	8
Joint 13	Seat under	Right roll	fixed joint	8
Joint 14	Seat back	Right wire holder	fixed joint	9
Coupler	Left cylinder	Right cylinder	coupler	9

3.1.2 Phase two of modeling

In phase two of the modeling, the joints that had been used in the simplified model were now replaced with bushings. The previous revolute joint, referred to as joint 3 between the *Handle* and the *Handle attachment*, has been replaced with a bushing, referred to as bushing 1. The previous cylindrical joint, referred to as joint 5 between the handle and the handle/long link connection has been replaced with a bushing, referred to as bushing 2. The previous spherical joint, referred to as joint 7 between the *Short link* and the *Link connection*, has been replaced with a bushing, referred to as bushing 3. The previous revolute joint, referred to as joint 9 between the *Left cylinder* and ground, has been replaced with a bushing, referred to as bushing 4.

The stiffness and damping coefficients for the different bushings can be seen in table 4. These values are not specific but were chosen due to that they accurately resemble the seat motion sufficiently enough. It is the magnitude of these coefficients that decide in which degree of freedom the parts will be able to translate and rotate. This means that if a high stiffness is assigned to for example the translational x direction, the part will not be able to translate much in this direction.

Table 4. The translational stiffness, translational damping, rotational stiffness and rotational damping of the bushings.

	Stiffness [N/mm] (Translational) (x,y,z)	Damping [N-sec/mm] (Translational) (x,y,z)	Stiffness [N-mm/deg] (Rotational) (x,y,z)	Damping [N-mm-sec/deg] (Rotational) (x,y,z)
Bushing 1	$(10^5, 10^5, 10^5)$	(10,10,10)	$(10^5, 10^5, 0)$	(10,10,0)
Bushing 2	$(10^5, 10^5, 10^5)$	(10,10,10)	$(10^5, 10^5, 0)$	(10,10,0)
Bushing 3	$(10^5, 10^5, 10^5)$	(10,10,10)	$(10^5, 10^5, 0)$	(10,10,0)
Bushing 4	$(10^5, 10^5, 10^5)$	(10,10,10)	$(10^5, 10^5, 0)$	(10,10,0)
Bushing 5	$(10^5, 10^5, 10^5)$	(10,10,10)	$(10^5, 10^5, 0)$	(10,10,0)

As was previously mentioned, the friction that exists between the different parts is here included. Instead of applying the friction coefficients between parts in Adams, a simplification with the help of data received from the subcontractor Brose was used [12]. Brose had done measurements of the opening force needed to open a recliner. The measurement was first done by applying a force at the peak of the handle, which can be seen as the blue part in figure 17.

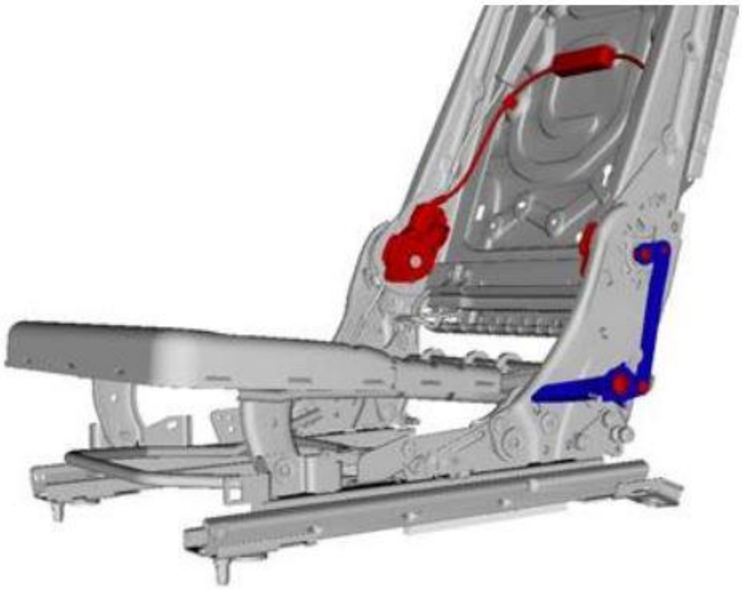

Figure 17. The seat where the handle system has been highlighted in blue.

In the second measurement the handle system is disassembled and a force is applied using the same lever as previously. By comparing the two forces needed to open a recliner it could be noted that a 80% higher force was needed when taking the handle system into consideration. This means that the friction and possible geometrical variation increases the handle force with 80%. By applying a torque in Adams which increases the required handle force with 80%, the corresponding forces introduced by the friction coefficients and the geometrical variation are included. These forces reduce the motion between parts and are therefore applied as an opposing torque relative the motion. The magnitude of the fully developed torque was 1840 Nmm.

As was described in the chapter, 2.1 *Seat definitions*, to open the *Left recliner* three *Sliders* have to be released and the friction between the parts has to be overcome. Since the CAD geometry did not contain a detailed model of the recliner system a simplification had to be made where the *Sliders* and the friction was taken into consideration. Brose had made measurements of the moment needed to open the recliner system and provided these. This value was between $1500 - 2000$ Nmm and this could be implemented in the model as a similar torque explained for the friction earlier. When simulating the friction between the parts it had to be done in a way where the friction is not acting between parts when the system is at rest. This reflects a realistic system. This was made in Adams by making the torque, representing the friction, dependent on the angle of the parts where the friction is acting. When the angle between the parts reaches 0.1 degrees the torque is activated. A similar implementation was used in the torque needed to open the recliner. In Adams, the torque in the recliner was created in such a way that the maximum torque was activated at 0.1 degrees and then linearly decreased to 0 Nm at 14 degrees. This was due to that the recliner is fully opened at 14

degrees. As was previously mentioned, there are some losses in the wire due to friction which is here accounted for. Brose provided the value of the friction loss which was 20 % and by multiplying the recliner moment with 2.2 this was taken into consideration. Multiplying with 2 would only take into account the forces created in both the left and right cylinder but not the losses in the wire due to friction. To measure the magnitude of the force needed in the *Handle*, a force was applied on the *Handle* in Adams. This force was gradually increased with time in order to obtain the needed Handle force. When the force is large enough to make the cylinders rotate 14 degrees, meaning that the recliners are opened, the simulation is stopped and a force magnitude can be obtained.

3.1.3 Phase three of modeling

To get a more accurate model in Adams some parts had to be made flexible i.e. into non-rigid parts, meaning that these parts are allowed to deform. To do this a Finite Element Analysis (FEA) had to be made. The parts that were made flexible were the ones that were expected to deform due to their geometries and the way the applied load affected them. These parts were the *Handle* and the *Long link* which can be seen in figure 15. The *Handle* because the force is directly applied on it and the *Long link* because of its long and slender shape.

The first step when making these parts flexible was to import them into the preproccesor SimXpert [7] one by one.

A mesh was created with solid elements with size 2.5 mm. The program allows the user to check the quality of the created mesh meaning that the user can verify that the mesh is created properly with the selected element size. A proper mesh is characterized by among other things, that the angles in the elements are not too sharp or to blunt and that the mesh elements that are in contact which each other are about the same size. The program easily lets the user know if these criteria are not fulfilled by assigning the divergent elements with a different color, in this case purple. This can be seen for the *Long link* in figure 18. Without making the element size unnecessarily small, which would be more computationally demanding, the element size 2.5 mm was sufficient enough. The next step was to define the parts' material properties. Since the parts are made of steel, the Young's modulus was defined as 210 GPa , the Poisson's ratio as 0.3 and the density as $7800\ \mathrm{kg/m^3}$.

Figure 18. The software SimXpert lets the user know if a proper mesh has not been created.

Two nodes on each part, corresponding to the part's attachment points in Adams, had to be added. This is because when working with flexible parts in Adams the user has to work with nodes instead of points and markers which were used in the previous rigid case. These nodes i.e. attachment points for respective part, can be seen in figure 19. An extra node was added on the *Handle* were the applied load is positioned.

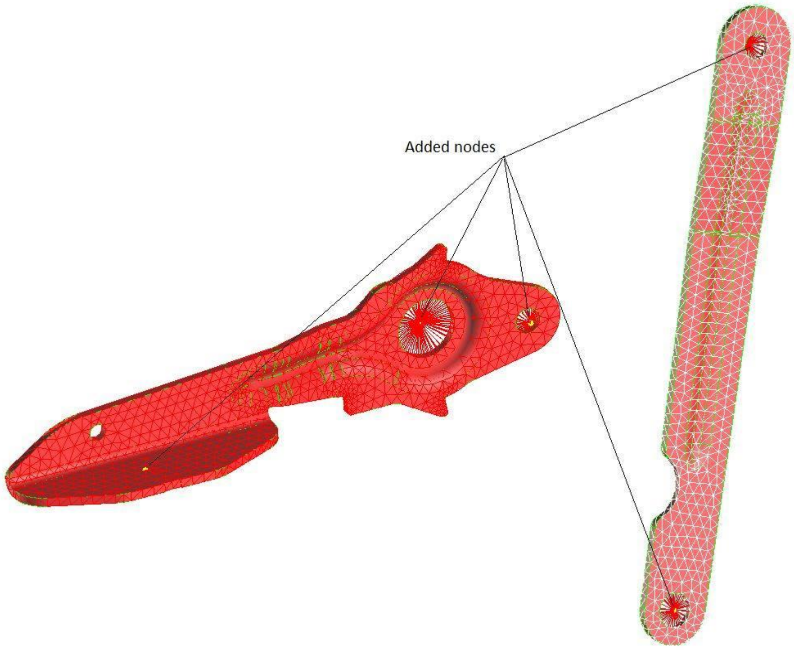

Figure 19. Placement of the added nodes on the non-rigid parts.

Once the parts were meshed, MSC Nastran, an FEA program [8], was used to perform the FEA and to calculate the mode shapes. Finally the now created flexible parts could be imported into Adams. Once the parts had been imported into Adams it was possible to verify that the parts were now flexible. When performing a simulation of the model, the deformations of these two parts are now taken into account, which can be seen in figure 20.

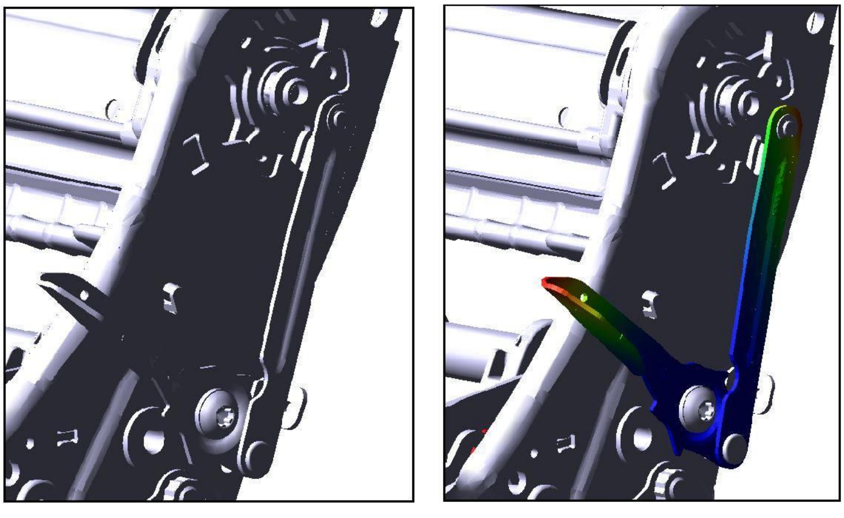

Figure 20. The rigid parts (left) and the non-rigid parts (right) are seen in Adams during a simulation.

3.2 Contribution analysis

A contribution analysis was made to determine which parameters that contributed most to the magnitude of the handling force. To make such an analysis, Adams/Insight was used which is a program that allows the user to run a series of investigations for measuring the performance and optimizing a mechanical system [9]. Parameters, which in Adams are known as design variables, were chosen and investigated. These were the recliner moment, the friction between the parts and the friction loss in the *Wire*. These three parameters primarily determine the magnitude of the handling force and by varying these it could be determined how much they affect the force. By determining the most contributing parameter(s) it is possible to identify the major cause of the obtained result.

As mentioned above the friction loss in the *Wire* was $20\ \%$. This value was now defined to vary between $10\% - 30\ \%$. As explained in chapter 3.1.2 *Phase two of the modeling*, for friction loss in the *Wire* as $20\ \%$, the recliner moment was multiplied with 2.2. Now, when the friction loss in the *Wire* varies between $10\% - 30\ \%$, the recliner moment is instead multiplied with $2.1 - 2.3$. For the recliner moment the value was defined to vary in the confirmed range $1500 - 2000$ Nmm that was given by Brose. Lastly, the increased handling force generated by the friction between the parts was allowed to be 20% larger/smaller than the given value from Brose. This gave the range $1480 - 2200$ Nmm for the moment representing the friction mentioned in chapter 3.1.2 *Phase two of the modeling*. With the help of Adams/Insight, 64 different combinations of these design variables were created which can be seen in table 5. These combinations were then used to evaluate the impact of each parameter and the magnitude of the handling force.

Table 5. The variation of parameters seen in Adams/Insight.

	Trial	Recliner moment [Nmm]	Moment due to friction [Nmm]	Wireloss
1	Trial 1	1500	1480	2.1
2	Trial 2	1500	1480	2.16667
3	Trial 3	1500	1480	2.23333
4	Trial 4	1500	1480	2.3
5	Trial 5	1500	1720	2.1
6	Trial 6	1500	1720	2.16667
7	Trial 7	1500	1720	2.23333
8	Trial 8	1500	1720	2.3
9	Trial 9	1500	1960	2.1
10	Trial 10	1500	1960	2.16667
11	Trial 11	1500	1960	2.23333
12	Trial 12	1500	1960	2.3
13	Trial 13	1500	2200	2.1
14	Trial 14	1500	2200	2.16667
15	Trial 15	1500	2200	2.23333
16	Trial 16	1500	2200	2.3
17	Trial 17	1666.67	1480	2.1
18	Trial 18	1666.67	1480	2.16667
19	Trial 19	1666.67	1480	2.23333
20	Trial 20	1666.67	1480	2.3
21	Trial 21	1666.67	1720	2.1
22	Trial 22	1666.67	1720	2.16667
23	Trial 23	1666.67	1720	2.23333
24	Trial 24	1666.67	1720	2.3
25	Trial 25	1666.67	1960	2.1
26	Trial 26	1666.67	1960	2.16667
27	Trial 27	1666.67	1960	2.23333
28	Trial 28	1666.67	1960	2.3
29	Trial 29	1666.67	2200	2.1
30	Trial 30	1666.67	2200	2.16667
31	Trial 31	1666.67	2200	2.23333
32	Trial 32	1666.67	2200	2.3
33	Trial 33	1833.33	1480	2.1
34	Trial 34	1833.33	1480	2.16667
35	Trial 35	1833.33	1480	2.23333
36	Trial 36	1833.33	1480	2.3
37	Trial 37	1833.33	1720	2.1
38	Trial 38	1833.33	1720	2.16667
39	Trial 39	1833.33	1720	2.23333
40	Trial 40	1833.33	1720	2.3
41	Trial 41	1833.33	1960	2.1
42	Trial 42	1833.33	1960	2.16667
43	Trial 43	1833.33	1960	2.23333
44	Trial 44	1833.33	1960	2.3
45	Trial 45	1833.33	2200	2.1
46	Trial 46	1833.33	2200	2.16667
47	Trial 47	1833.33	2200	2.23333
48	Trial 48	1833.33	2200	2.3
49	Trial 49	2000	1480	2.1
50	Trial 50	2000	1480	2.16667
51	Trial 51	2000	1480	2.23333
52	Trial 52	2000	1480	2.3
53	Trial 53	2000	1720	2.1
54	Trial 54	2000	1720	2.16667
55	Trial 55	2000	1720	2.23333
56	Trial 56	2000	1720	2.3
57	Trial 57	2000	1960	2.1
58	Trial 58	2000	1960	2.16667
59	Trial 59	2000	1960	2.23333
60	Trial 60	2000	1960	2.3
61	Trial 61	2000	2200	2.1
62	Trial 62	2000	2200	2.16667
63	Trial 63	2000	2200	2.23333
64	Trial 64	2000	2200	2.3

3.3 Geometrical variations

During production and assembly of the seat, different parts of the seat are subjected to forces. Due to these forces, deformations will occur in different parts, which will remain in the seat when it is assembled. The magnitude of these forces can vary and will therefore affect the geometry of the parts with different amounts. A geometrical variation was therefore performed were some chosen parts were varied in a worst case scenario i.e. the most likely maximum amount of variation that can occur in the parts were implemented.

After discussions with the supervisor at VCC and a technical expert which had earlier worked within this area of the seat [11], it was decided to examine two different kinds of geometrical variations. The first variation was in the peak of the *Handle*, which was now translated 3 mm in the lateral direction. The second variation was in the *Long link*, which was now made distorted and rotated 1.5 degrees in its lower and upper part. The rotations were made in a way that the rotation in the upper part of the *Long link* was in the opposite direction of the rotation in the lower part of the *Long link*. This was made to get a "worst case" scenario. These two variations were considered the most occurring and were therefore chosen. Simulations were made in Adams with these variations implemented and the results were compared to the results were there was no geometrical variation at all. By comparing these two results, the force in the *Handle* which is needed to release the recliner with and without geometrical variation, could be obtained.

3.4 Redesign based on the most contributing parameter

From the contribution analysis, it was possible to identify which design variable that contributed the most to the magnitude of the handle force. As will be seen in the results chapter below, the recliner moment affected the force magnitude the most. A redesign of the recliner was therefore seen as an area for potential improvement of the handle force. The redesign would primarily cover a change of spring stiffness and also a change in geometry of the *Steering ring* in figure 5.

Since the recliner system was not included in the CAD model, a physical seat had to be disassembled to obtain more accurate information regarding the part. Once the recliner had been removed from the seat it was possible to scan the desired parts, seen in figure 21, and in turn obtain their CAD geometries. The CAD geometries were then imported into CATIA where these could be used to model the recliner. Once the modeling was done in CATIA the geometry could be imported into Adams, seen in figure 22. The blue part, known as the *Steering ring*, is modeled with a revolute joint to ground, allowing the part to rotate around one axis in space. The three red *Sliders* are modeled with translational joints to the *Steering ring*, allowing them to translate in one direction. The *Steering ring* and the *Sliders* are interacting as explained in chapter 2.1 *Seat definitions*, meaning that a connection between the two parts had to be applied in Adams. From the points used to construct the model in CATIA, it was possible to create a spline in Adams, seen in figure 23. This spline was used together with the point, seen in figure 23, to model a force connection. This connection allowed the parts to interact with each other, meaning that the point was allowed to move along the created spline. Lastly three springs were added to the *Sliders*, to resemble the *Closer springs* in the real recliner model. Now a simulation of the recliner movement could be performed. A moment of 2000 Nmm was applied to the *Steering ring* and the springs' stiffness and damping were adjusted until the *Steering ring* rotated 14 degrees. This means that a recliner model was constructed where the recliner is opened after 14 degrees when applying a moment of 2000 Nmm.

Figure 21. To the left, one of the disassembled *Sliders* can be seen. To the right, a third of the *Steering ring* can be seen.

The next step was to redesign the *Steering ring* in order to achieve a design that allowed for easier rotation. The previously created spline was now modified, as seen in figure 23. By using the new spline design, while keeping the same stiffness and damping in the springs, the magnitude of the applied moment was decreased until the *Steering ring* could no longer rotate 14 degrees. By performing the redesign it was possible to examine if a slight modification of the spline could lower the magnitude of the applied torque needed to open the recliner.

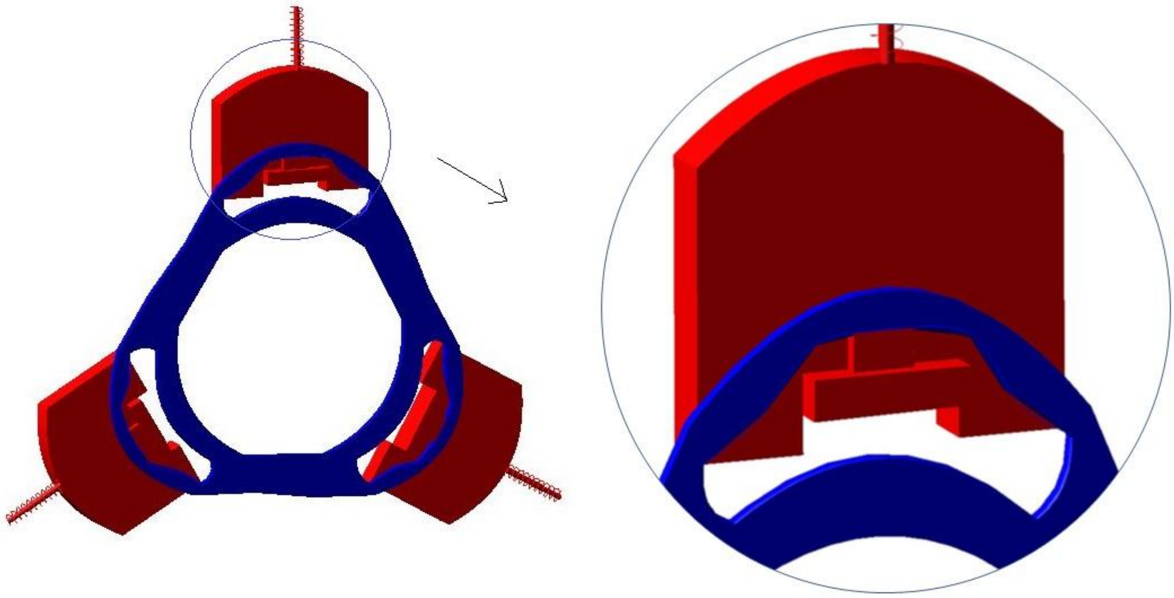

Figure 22. The *Steering ring* with *Sliders* in Adams.

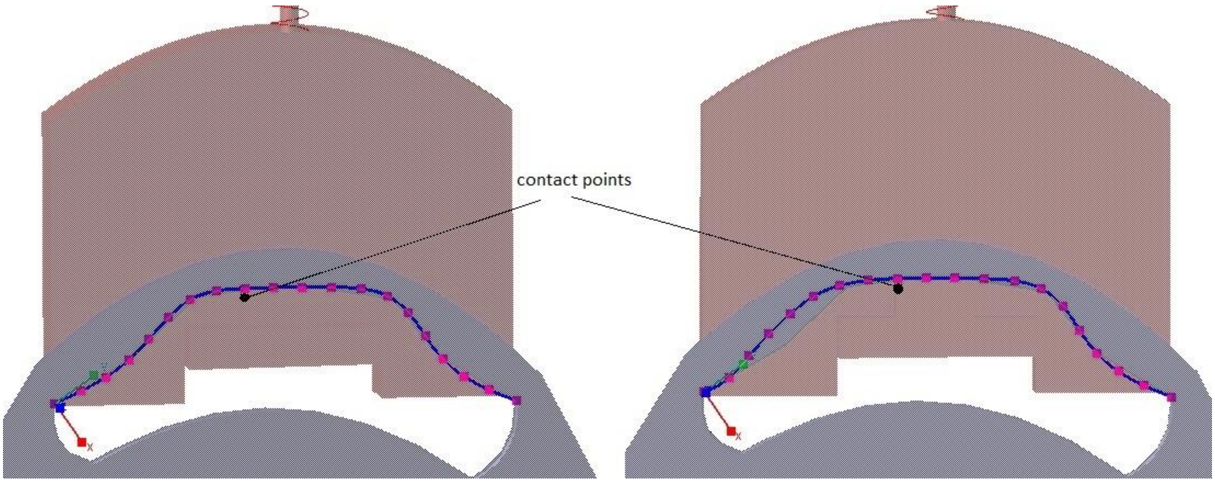

Figure 23. To the left, the original spline i.e. the original shape of the *Steering ring* can be seen. To the right, the new spline i.e. the redesigned shape of the *Steering ring* can be seen.

Once the redesign was performed, a new model was built in Adams, as can be seen in figure 24. The new Adams model, including the new recliners, was now constructed to examine if there existed a possible combination of redesigned spline and change in spring stiffness that would lower the handle force to 40 N which was a request from VCC. A coupler is used between the *Left* and *Right cylinder* to assure that they rotate simultaneously. To account for the loss in the *Wire*, the *Right cylinder* has springs with slighty higher stiffness which will make it harder for the cylinder to rotate.

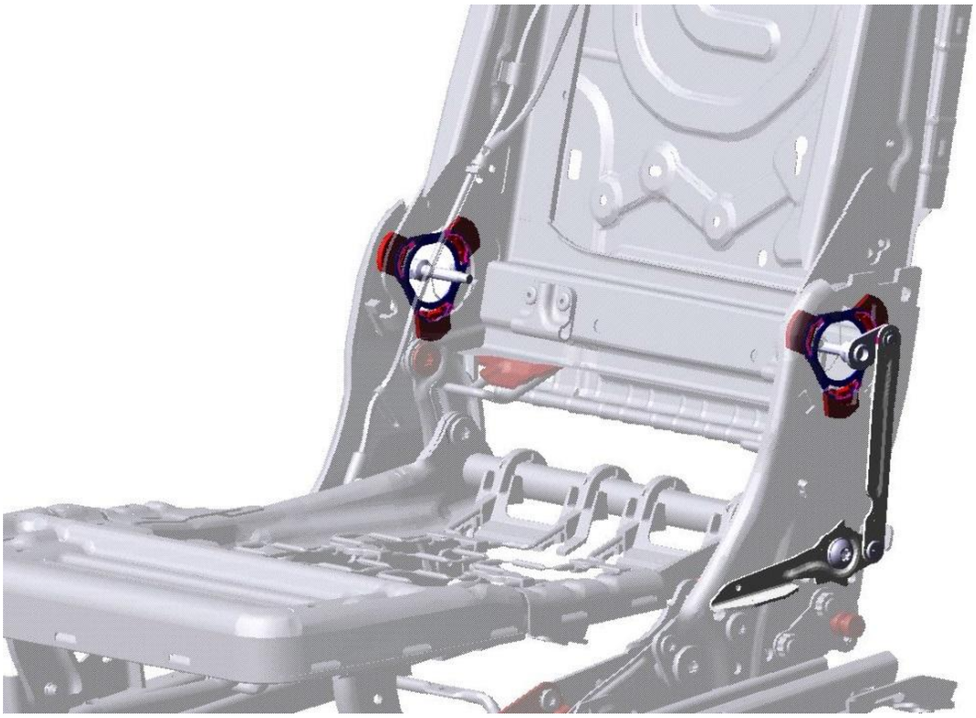

Figure 24. In the new model the redesigned recliners have been implemented to see what handle force is needed to release the recliners.

4. Results

Below the results from the handle force simulation, contribution analysis, geometrical variations and the redesign of the recliner are presented.

4.1 Handle force

Once the modeling of the seat in Adams was performed, it was possible to examine the magnitude of the force needed to pull up the *Handle*. Without the implementation of flexible parts, the handle force needed was $70 - 93$ N, seen in figure 25 and figure 26. The span here is obtained due to that the recliner moment was known to vary between $1500 - 2000$ Nmm. These results can be compared to VCC physical testing which are $70 - 120$ N. After the implementation of flexible parts, the needed handle force did not change, meaning that no deformation in the parts is obtained.

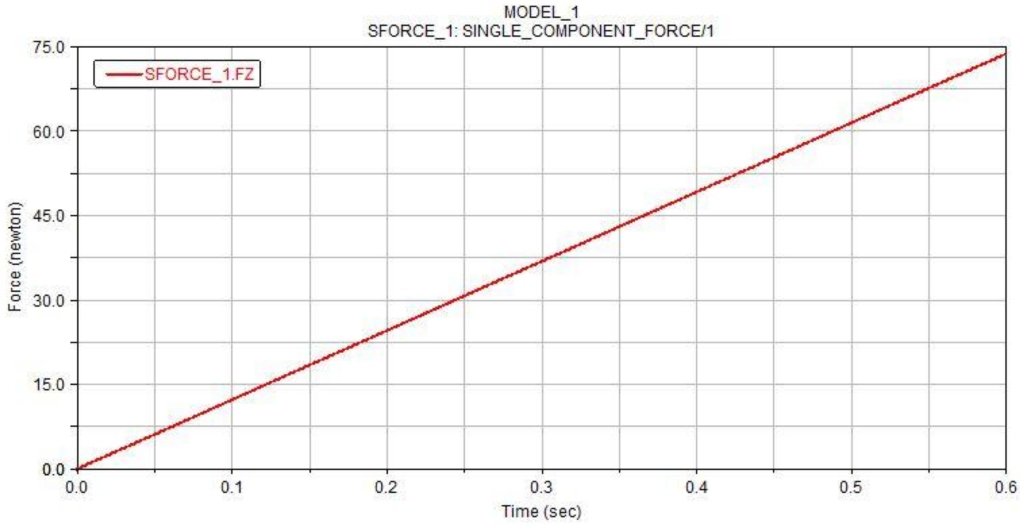

Figure 25. The development of the handle force with time for a recliner moment of 1500 Nmm.

Figure 26. The development of the handle force with time for a recliner moment of 2000 Nmm.

4.2 Adams/Insight

As explained in 3.2 *Contribution analysis*, three design variables were varied according to table 5. The resulting force from the 64 different combinations of design variables can be seen in table 6. Here it can be noted that the smallest combination of these three variables gave the handle force 66.5 N, while the largest combination of these gave the value 97.4 N. This means that with the selected variation for the parameters, the handle force will either be approximately 7 N smaller or larger than for the handle force in chapter 4.1 *Handle force*.

Table 6. The variation of parameters and the resulting force seen in Adams/Insight.

	Trial	Recliner momer	Moment due to	Wireloss	Force
1	Trial 1	1500	1480	2.1	66.4813
2	Trial 2	1500	1480	2.16667	67.9819
3	Trial 3	1500	1480	2.23333	69.4812
4	Trial 4	1500	1480	2.3	70.9815
5	Trial 5	1500	1720	2.1	69.5368
6	Trial 6	1500	1720	2.16667	71.0381
7	Trial 7	1500	1720	2.23333	72.5391
8	Trial 8	1500	1720	2.3	74.0358
9	Trial 9	1500	1960	2.1	72.5925
10	Trial 10	1500	1960	2.16667	74.093
11	Trial 11	1500	1960	2.23333	75.5956
12	Trial 12	1500	1960	2.3	77.0915
13	Trial 13	1500	2200	2.1	75.6482
14	Trial 14	1500	2200	2.16667	77.1489
15	Trial 15	1500	2200	2.23333	78.6492
16	Trial 16	1500	2200	2.3	80.1491
17	Trial 17	1666.67	1480	2.1	71.7283
18	Trial 18	1666.67	1480	2.16667	73.4008
19	Trial 19	1666.67	1480	2.23333	75.068
20	Trial 20	1666.67	1480	2.3	76.7345
21	Trial 21	1666.67	1720	2.1	74.7864
22	Trial 22	1666.67	1720	2.16667	76.4552
23	Trial 23	1666.67	1720	2.23333	78.1204
24	Trial 24	1666.67	1720	2.3	79.7889
25	Trial 25	1666.67	1960	2.1	77.8445
26	Trial 26	1666.67	1960	2.16667	79.5113
27	Trial 27	1666.67	1960	2.23333	81.1785
28	Trial 28	1666.67	1960	2.3	82.8432
29	Trial 29	1666.67	2200	2.1	80.901
30	Trial 30	1666.67	2200	2.16667	82.5683
31	Trial 31	1666.67	2200	2.23333	84.2376
32	Trial 32	1666.67	2200	2.3	85.9021
33	Trial 33	1833.33	1480	2.1	76.9859
34	Trial 34	1833.33	1480	2.16667	78.8174
35	Trial 35	1833.33	1480	2.23333	80.6535
36	Trial 36	1833.33	1480	2.3	82.4872
37	Trial 37	1833.33	1720	2.1	80.0384
38	Trial 38	1833.33	1720	2.16667	81.8723
39	Trial 39	1833.33	1720	2.23333	83.7081
40	Trial 40	1833.33	1720	2.3	85.5415
41	Trial 41	1833.33	1960	2.1	83.0968
42	Trial 42	1833.33	1960	2.16667	84.928
43	Trial 43	1833.33	1960	2.23333	86.7654
44	Trial 44	1833.33	1960	2.3	88.5992
45	Trial 45	1833.33	2200	2.1	86.1529
46	Trial 46	1833.33	2200	2.16667	87.9875
47	Trial 47	1833.33	2200	2.23333	89.8195
48	Trial 48	1833.33	2200	2.3	91.6558
49	Trial 49	2000	1480	2.1	82.2384
50	Trial 50	2000	1480	2.16667	84.2398
51	Trial 51	2000	1480	2.23333	86.2402
52	Trial 52	2000	1480	2.3	88.2396
53	Trial 53	2000	1720	2.1	85.2945
54	Trial 54	2000	1720	2.16667	87.2948
55	Trial 55	2000	1720	2.23333	89.298
56	Trial 56	2000	1720	2.3	91.2975
57	Trial 57	2000	1960	2.1	88.3504
58	Trial 58	2000	1960	2.16667	90.3513
59	Trial 59	2000	1960	2.23333	92.3524
60	Trial 60	2000	1960	2.3	94.3513
61	Trial 61	2000	2200	2.1	91.4091
62	Trial 62	2000	2200	2.16667	93.4057
63	Trial 63	2000	2200	2.23333	95.4077
64	Trial 64	2000	2200	2.3	97.4097

In Insight it was possible to examine which of the three variables that contributed the most to the magnitude of the handle force. By creating three scatter plots with the force versus the design variables, the impact of these variations could be studied, as seen in figures 27-29. In the figures, the results for the 64 different trials can be viewed. If one would for example examine the difference in the handle force, for the marked trials, it is possible to evaluate the impact of the variation. This is due to that for the marked trials, two design variables have the same value, while varying the studied parameter. In figure 27, the scatter plot for the wire loss is shown. Here it can be seen that when varying the value for the wire loss, between its smallest and largest value, while holding the other two parameters constant, the magnitude of the force varies with 3.5 N. For the moment due to friction, this variation affects the handle force with 9 N, which can be seen in figure 28. Lastly studying the recliner moment, the variation affects the handle force with 16 N, which can be seen in figure 29. This means that the recliner moment is the parameter that affects the magnitude of the handle force the most.

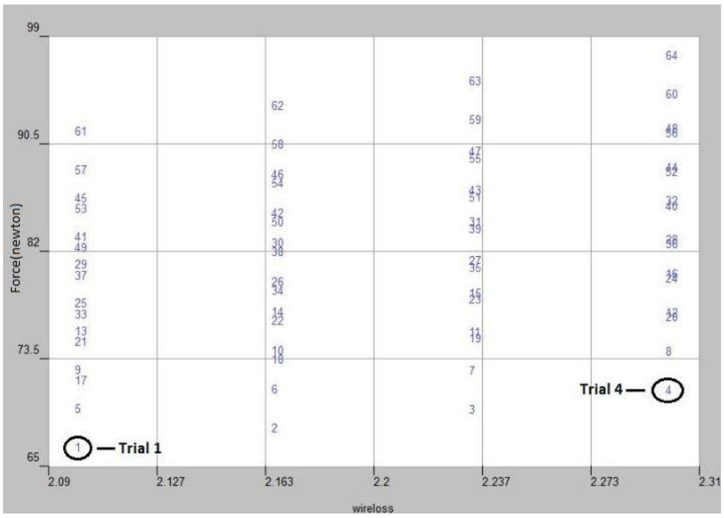

Figure 27. The magnitude of the handle force varies with 3.5 N between the smallest and largest value of the wire loss.

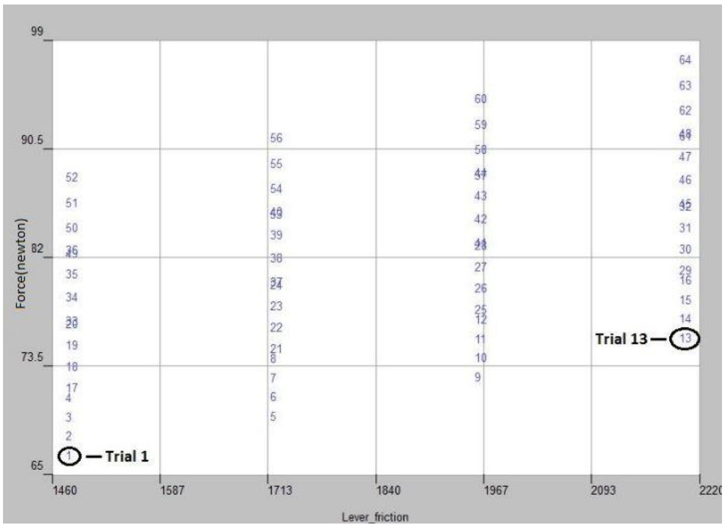

Figure 28. The magnitude of the handle force varies with 9 N between the smallest and largest value of the lever friction.

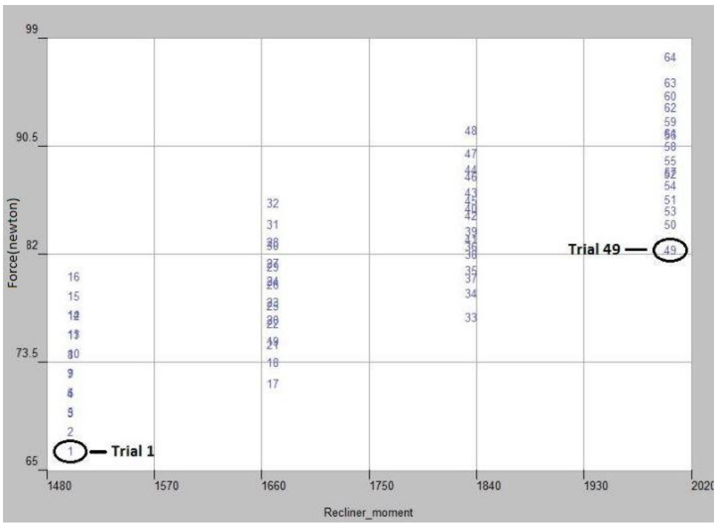

Figure 29. The magnitude of the handle force varies with 16 N between the smallest and largest value of the recliner moment.

4.3 Geometrical variations

One of the questions in the problem definition was to evaluate whether geometrical variations should be incorporated as a parameter in the simulation if output variation cannot be motivated by existing contributing parameters. By applying a slight deformation to the parts as explained in 3.3 *Geometrical variations*, the increase of the handle force was studied. With the applied rotation on the *Long link* and the applied lateral deflection on the *Handle*, the magnitude of the handle force increased with 6.5 %.

4.4 Handle force after redesign

As explained in *3.4 Redesign based on the most contributing parameter*, a new Adams model was created to examine what combination of spline and spring stiffness that would lower the handle force to 40 N. The original stiffness and damping in the springs were 9 N/mm and 0.09 N-sec/mm. By using the new spline and lowering the springs' stiffness with 58 %, it was possible to obtain a handle force of 40 N, seen in figure 30. As will be discussed in chapter 5 *Discussion*, the translation distance of the *Sliders* differs between the original spline and the redesigned spline. For the original recliner system the translation distance was 3 mm and for the redesigned recliner system the translation distance was now decreased to 2 mm.

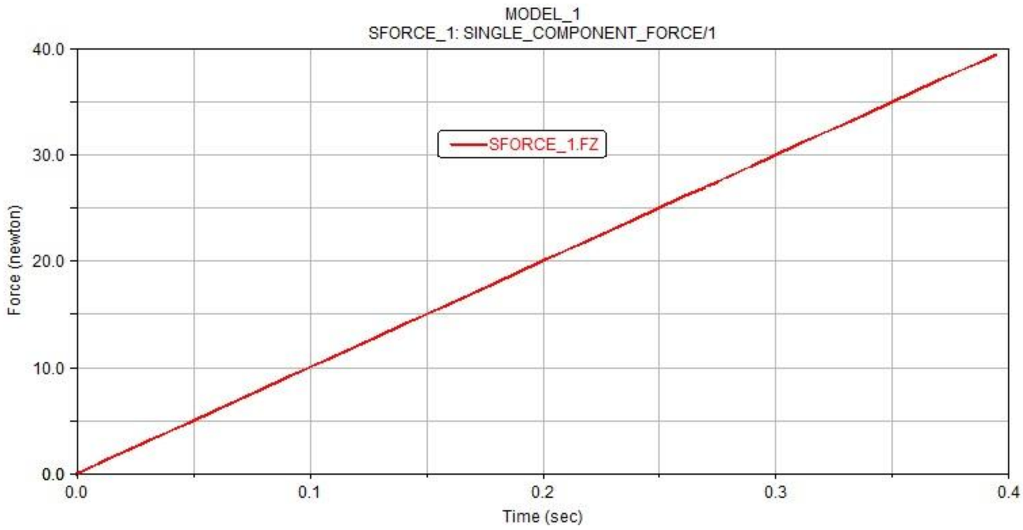

Figure 30. The development of the handle force with time for the redesigned recliner.

5. Discussion and conclusions

In this chapter the discussion of the different results is presented. Also the conclusions and recommendations for future work based on the results are stated.

5.1 Discussion

The goal of the master thesis was to develop a method for virtual verification of handling forces with the help of Adams. As seen in chapter 4.1 *Handle force*, the results from the simulation corresponded well to the physical test data. Even though the results are quite similar, there are some areas that are worth discussing.

Regarding the seat model used in Adams, some simplifications had to be made. As explained in chapter 3.4 *Redesign based on the most contributing parameter*, a geometry of the recliner system was not included in the CAD model. Therefore an opposing torque representing the moment needed to open the recliner had to be implemented. Instead of modelling the interaction between all the parts in the recliner system, an assumption of the characteristic of the opening moment had to be made as explained in 3.1.2 *Phase two of the modelling*. This assumption was made through discussions with a supervisor with the goal to resemble the moment behavior as accurately as possible. Even though it is a qualified assumption, it is worth mentioning that it is an approximation.

The friction between the parts in the seat was also implemented through some simplifications. Since no measured friction data existed, it was not possible to model the exact friction between the parts. Therefore the friction was modeled as explained in 3.1.2 *Phase two of the modelling*, where physical test data was used that stated that the handle force increased with 80 % when measuring with the handle system attached. It should also be mentioned that these 80 % do not entirely consist of friction. When measuring with the handle system attached, it is possible that some parts are canted, which in turn will affect these 80 % and therefore the implemented friction also includes some geometrical variations. This means that geometrical variations that exist in the parts is also taken into account when modeling the friction, where the friction is implemented as an opposing torque as explained above for the recliner system.

For the studied handling force, the parts were not expected to deform. Therefore an implementation of non-rigid bodies is unnecessary for the seat system. Though it should not be excluded from future simulation of other handling forces, where there is a possibility of parts deforming.

When performing the contribution analysis in Adams/Insight, the design variables and their variations were selected through qualified assumptions. As explained in 3.2 *Contribution analysis*, the chosen design variables were identified to primarily determine the magnitude of the handle force. It is worth mentioning that the variation for the friction and the wire loss were foremost selected by the authors. This means that the variations were qualified guesses and that a change in these will affect the contribution to the handle force.

The implementation of geometrical variations was primarily performed to evaluate its effect. As mentioned above for the lever friction, the 80 % that were accounted for also included geometrical variation. This means that when implementing geometrical variations, these are accounted for twice. Therefore the obtained handle force is not correct but the percentage of the geometrical variations' effect is.

Lastly regarding the recliner system, the performed redesign was not optimized, but it was made to examine possible areas that could decrease the handle force. When changing the spline geometry, as explained in 3.4 *Redesign based on the most contributing parameter*, the translation distance of the sliders are affected, as mentioned in 4.4 *Handle force after redesign*. Since the *Sliders* have to translate a certain distance to separate from the grooves, it is important that this distance is taken into account when creating a new design. It should also be mentioned that it is not possible to guarantee that with the redesigned recliner, a handle force of 40 N will be obtained. To get a more accurate value, all the parts in the recliner and their interactions have to be taken into account. Since the only two parameters, in the recliner system in Adams, that could be changed were the spline and the spring stiffness, the spring stiffness had to be decreased to an unrealistic value of 60 %. But the results show that there is potential in decreasing the handle force by changing the design of the recliner system.

5.2 Conclusions

As was presented in 3.1 *Building the seat model in Adams*, the simulated handle force was 73-90 N compared to VCC physical tests which were 70-120 N. From these results it can be concluded that the two results do not differentiate considerably.

From the contribution analysis it was possible to determine that the recliner moment affected the handle force the most. As explained in 5.1 *Discussion*, the design variables in the contribution analysis were chosen by us. Therefore more variables should be evaluated and in this case more areas for potential improvements can be obtained.

A subtarget of the master thesis was to understand whether geometrical variations should be considered for the evaluated system. Since the simulated results and physical measurements do not conform exactly, a more advanced implementation of geometrical variations should be performed, although geometrical variations are expected to increase the handle force magnitude even more.

For the studied handling force, the parts were not expected to deform. Therefore an implementation of non-rigid bodies is unnecessary. However, it should not be excluded from future simulation of other handling forces.

The purpose of the master thesis was to develop a method for virtual verification of handling forces for a rear seat system using primarily Adams. Since VCC has over 100 demands on different handling forces and the modeled handling force in this master thesis is one of the more advanced, it can be concluded that it should be possible to model handling forces virtually and obtain good results. This means that there is potential in eliminating the experimental need for measuring handling forces and in turn decrease dependency of physical product testing.

5.3 Future work

To get a more accurate result for the handle force, the complete recliner system should be included in the Adams model. Instead of modelling the friction as explained in chapter 3.1.2 *Phase two of the modelling*, more accurate investigations have to be made to determine exact friction data that can be implemented in the Adams model. Using this approach it is possible to eliminate the opposing torques that represents the friction and the moment needed to open the recliner and thus obtain a model that resembles reality more accurately.

Once a more advanced model is built in Adams, a more extensive contribution analysis should be made. This means that more design variables and a more thorough examination of the amount of variation for the design variables should be performed. With this more accurate results will be obtained and it will be easier to identify the most contributing parameter(s).

Regarding the geometrical variations, a more comprehensive analysis should be made where more potentially canted parts are included. With this it will be possible to identify what effect geometrical variation has on the handle force and also to distinguish the most critical parts. Finally, for the redesigned recliner system, all the parts and their interactions should implemented in the Adams model. With this it will be possible to obtain more reliable results. Since the redesign of the recliner system in this master thesis only examined if there was potential in lowering the handle force with a redesign, an optimization should be performed. The optimization should include the spline geometry, spring stiffness, interaction between the other parts and how these changes will affect the recliner function.

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