Finite Element and Experimental Analysis of Function of Plastic Clips

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Abstract:
The aim of this work is to investigate the function of plastic clips which are used to join different parts, during mounting and dismounting processes. The clips are made of POM and will be mounted on steel plates. The study is undertaken using experimental and numerical methods. In experiments, the mounting and dismounting forces are measured with respect to vertical displacement of the clips related to the plate. The numerical method is performed using structural implicit non-linear quasi-static Finite Element method. In order to mesh, solve and study the results of FEM, Hyper Works, LS-DYNA and MD-Nastran programs are used. The effect of some parameters, e.g. stiffness, friction, hole diameter, etc. is also investigated and discussed. The project is done at SAAB Automobile AB.

Keywords:
Acknowledgements

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Trollhättan, July 2011

M. Mahdi Honarpardaz
## Contents

**Abstract**

**Acknowledgement**

**Table of contents**

### 1 Notations

### 2 Introduction

- 2.1 Problem Statement
- 2.2 Background
- 2.3 Objective
- 2.4 Framework
- 2.5 Scope

### 3 Material

- 3.1 Acetal/Polyoxymethylene (POM)
- 3.2 Carbon Steel

### 4 Method

- 4.1 Numerical: Finite Element
  - 4.1.1 Non-linearity
    - 4.1.1.1 Material Non-linearity
    - 4.1.1.2 Contact Non-linearity
    - 4.1.1.3 Geometry Non-linearity
  - 4.1.2 Quasi-Static Loading
  - 4.1.3 Contact Theory
    - 4.1.3.1 Contact Position Identification
    - 4.1.3.2 Impenetrability Condition
  - 4.1.4 Direct Integration Method: Implicit algorithm

- 4.2 Experimental Method

### 5 Experiment

- 5.1 Experiment on Clip #1
  - 5.1.1 Setup and conditions

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### Section 1: Notations

### Section 2: Introduction

#### 2.1 Problem Statement

#### 2.2 Background

#### 2.3 Objective

#### 2.4 Framework

#### 2.5 Scope

### Section 3: Material

#### 3.1 Acetal/Polyoxymethylene (POM)

#### 3.2 Carbon Steel

### Section 4: Method

#### 4.1 Numerical: Finite Element

- 4.1.1 Non-linearity
  - 4.1.1.1 Material Non-linearity
  - 4.1.1.2 Contact Non-linearity
  - 4.1.1.3 Geometry Non-linearity

- 4.1.2 Quasi-Static Loading

- 4.1.3 Contact Theory
  - 4.1.3.1 Contact Position Identification
  - 4.1.3.2 Impenetrability Condition

- 4.1.4 Direct Integration Method: Implicit algorithm

#### 4.2 Experimental Method

### Section 5: Experiment

#### 5.1 Experiment on Clip #1

- 5.1.1 Setup and conditions
5.1.2 Performance and results
5.2 Experiment on Clip #2
  5.2.1 Setup and conditions
  5.2.2 Performance and results

6 Simulation
  6.1 Simulation of Clip #1
    6.1.1 Model A: Complete model by LS-DYNA 971-4.2.1
    6.1.2 Model B: Mirror-Symmetric Model by LS-DYNA 971-4.21
      6.1.2.1 Parametric Study
    6.1.3 Model C: Full Model by MD-Nastran and LS-DYNA 971-5.1.1
      6.1.3.1 Parametric Study
  6.2 Simulation of Clip #2
    6.2.1 Parametric Study

7 Results
  7.1 Results of Clip #1
  7.2 Results of Clip #2

8 Discussions and Suggestions
  8.1 Discussions
  8.2 Suggestions

9 Conclusion
  9.1 Future Works

10 References

Appendix I: Material Properties

Appendix II: Human’s Hand Forces
1 Notations

\( \varepsilon \) \hspace{1cm} \text{Strain.}

\( \varepsilon^e \) \hspace{1cm} \text{Elastic strain.}

\( \varepsilon^i \) \hspace{1cm} \text{Inelastic strain.}

\( \sigma \) \hspace{1cm} \text{Stress.}

\( \sigma^e \) \hspace{1cm} \text{Elastic stress.}

\( \sigma^i \) \hspace{1cm} \text{Inelastic stress.}

\( \sigma_y \) \hspace{1cm} \text{Yield stress.}

\( \varepsilon^p \) \hspace{1cm} \text{Plastic strain.}

\( K, K \) \hspace{1cm} \text{Matrix of elastic modulus.}

\( \sigma \) \hspace{1cm} \text{Matrix of stress.}

\( \kappa \) \hspace{1cm} \text{Matrix of kinematic hardening.}

\( \kappa \) \hspace{1cm} \text{Isotropic hardening parameter.}

\( g \) \hspace{1cm} \text{Gap between nodes.}

\( x_{i}^{(s)} \) \hspace{1cm} \text{Position of slave node.}

\( x_{i}^{(m)} \) \hspace{1cm} \text{Position of master node.}

\( \Pi_c \) \hspace{1cm} \text{Penalty factor.}

\( \lambda \) \hspace{1cm} \text{Applied penalty force}

\( \kappa \) \hspace{1cm} \text{Penalty parameter.}
$M$ Matrix of mass.

$D$ Matrix of displacement.

$\dot{D}$ First derivation of matrix of displacement with respect to time.

$\ddot{D}$ Second derivation of matrix of displacement with respect to time.

$C$ Matrix of damping.

$R^{\text{ext}}$ Matrix of external loads and constrains.

$\Delta t_{cr}$ Critical time step size.

SOFSCL Scale factor of surface stiffness in LS-DYNA.

SFS Scale factor on default penalty stiffness in LS-DYNA.

SOFT Parameter of contact algorithm in LS-DYNA.

FS Static friction coefficient.

FD Dynamic friction coefficient.
2 Introduction

2.1 Problem statement

Since the strength per unit weight of engineering plastics is greater than commonly used metals and the material cost is relatively lower, over a few last decades, they are widely used instead of metals (Endo & Marui [15]). One of the industries which are strongly interested in replacement of metallic parts by plastic ones is automotive industry, because they want to decrease energy consumption of their products and also the production cost, as much as possible.

Clips are a large group of parts which are categorized in fasteners. They can be made of metal, rubber or plastic and be designed in very different shapes and for a variety of applications. The tasks of clips are usually sealing, fixing, orientation, making distance, affixing the clamping force, vibration damping, sound reduction, etc. (Jönsson & Måns [12]).

![Figure 2.1. A mounted part using a plastic clip.](image)

Plastic clips are commonly used in mounting the covering or decorative parts to body of automotives. Figure 2.1 shows how clips act as joint
between two parts. The most important reasons of using plastic clips are the low production cost, the high specific strength and the easy mounting and dismounting. However, sometimes manufacturers have some problems with function of these parts. A problem may occur in using the plastic clips is high mounting/dismounting loads. Also, the clips might be damaged during mounting or dismounting of parts. As mentioned before, clips should fix parts to body enough strongly, while, at the same time, be able to easily be mounted and dismounted, but because of some weak designs, they cannot play their role well. This is the motivation that makes SAAB Automobile AB interested in investigation of function of the plastic clips. This investigation requires simulating the clips, performing some experiments on them, and studying their function during mounting and dismounting.

In order to simulate the function of the plastic clips, a CAD model of the real clip should be prepared. Then, the CAD model should be discretized to some finite elements, and the material properties and boundary conditions should be applied. The next steps are solving the model by pushing the clip through a hole on a meshed steel plate and pulling out after complete mounting. The last step is drawing the force-displacement graphs and interpreting them.

Figure 2.2. Clip #1.
Simulation of plastic clips involves some sources of non-linearity which make it very complicated. The gap-contact connection between the clip and the steel plate is one of the non-linearity sources. Another source is the large deformation that occurs in the clip due to contact with the steel plate. The third one is the plastic deformation which might happen during mounting or dismounting. In addition, the 3D and non-regular shape of clips makes the prediction of their behaviour more difficult.

In this thesis, the functions of two different plastic clips which are both made of POM are investigated. In order to study the behaviour of these clips, several simulations and experiments in different conditions are performed to give a useful area of information about function of plastic clips. To recognize these two clips, they are named Clip #1 and Clip #2. Their shape and naming of different parts are shown in Figures 2.2 and 2.3, respectively.

![Figure 2.3. Clip #2.](image)
2.2 Background

Over the past few decades, much research have been done on simulation of contact by Finite Element method e.g. [20-23], however, this area of research is very widespread and it needs to be developed much further. Also, many papers have been published about acetals (POMs) and their behaviour, especially in wear and friction e.g. [13-16].

Although there is a short test report about mounting and dismounting forces at SAAB Automobile AB, the test has not followed a scientific method to show the function of plastic clips. Also, the simulation of plastic clips is for the first time in this thesis.

Jönsson and Måns [12] have published a master thesis report which contains some general information about all clips which are used in Volvo Car Corporation. However they have mostly focused on statistics and economical affairs about clips but not on function analysis.

2.3 Objectives

The main purpose is to gain more knowledge about simulations of the behaviour of plastic clips which are used commonly in automotive industries. Since automotive industries are going to simulate parts to predict the behaviour of them individually and as a part of a total body, this thesis is going to investigate the behaviour of plastic clips using simulation and experiment, and compare the results to examine that how accurate and reliable is the simulation. To achieve this aim, both numerical and experimental methods are performed. The numerical method is implemented using Finite Element.

2.4 Framework

The framework of project is shown in Figure 2.4.
Figure 2.4. Framework of project.
2.5 Scope

Due to the limited time and the complexity of project, all parameters which influence on the function of clips cannot be investigated. The main focus is on parameters which have the most important effect on behaviour of clips. From the results point of view, the focus is only on vertical reaction forces which applied to the clip support with respect to vertical displacement. Also, the visible damages on clips during mounting and dismounting processes are investigated.

There are many different types of plastic clips used in a car which can be the subject of investigation but because of limited time (20 weeks) it is not possible to simulate and perform experiments on all of them. Thus, only two plastic clips have been selected randomly to be investigated.

In order to simplifying the problem, the motion of clips during mounting and dismounting is considered that low speed and uniform to can assume it as an quasi-static motion.
3 Material

3.1 Acetal/Polyoxymethylen (POM)

Engineering plastics which have shown that have good mechanical properties are now frequently used in machine parts. Engineering plastics are thermoplastics (Endo & Marui [15]). High specific strength (strength per unit weight), low production cost and satiety of lubrication in sliding (Mergler et al [14]) make designers and manufacturers to use them even instead of metallic parts.

Acetals or polyoxymethylen (POM) are among of high-performance engineering thermoplastics which are similar to nylon in appearance but not in properties. They are strong and rigid (but not brittle) and have good moisture, heat and chemical resistance (Harper [7, 8]). Figure 3.1 shows the chemical structural unit of acetal.

\[ \begin{array}{c}
\text{H} \\
\text{C} \\
\text{H} \\
\end{array} \quad \begin{array}{c}
\quad \\
- \\
\text{n} \\
\end{array} \]

*Figure 3.1. Polyacetal resin*

The most outstanding properties of acetals are high tensile strength and stiffness, resilience, good recovery from deformation under load, good toughness against fatigue and dimensional stability. Their surface is hard, smooth and glossy which makes their static and dynamic friction coefficients very low. They have also very good creep resistance, low moisture absorption and low thermal and electrical conductivity (Harper [7, 8]). High tensile yield strength of acetals is shown in comparison with other thermoplastics in Figure 3.2.
The outstanding properties of acetal result that it finds wide applications in machined rollers, bearings, gears and other wear-resistant parts. Some parts which are made of acetal are automobile instrument clusters, pump impellers, conveyor links, drive sprockets, bearings, etc. Acetal’s spring quality makes it useful for producing clips (Mann [11]).

Polyacetal is supplied in two basic forms: homopolymers and copolymers. Impact strength of acetal resins can be even more improved by addition of elastomers such as polyurethane, polybutadiene, etc (Harper [8]).

In this thesis the acetal is considered as both completely elastic and elasto-plastic material. In both cases it is assumed an isotropic material. The data sheets and properties of acetal are available in Appendix I.

### 3.2 Carbon steel

The carbon steel which is used in experimental tests is a regular structural carbon steel plate that is commonly used in automotive industry. In order to simplifying the problem, in simulations of this thesis, the carbon steel plate is assumed an isotropic rigid plate. However, in tests we can see some plastic deformations on plate near the hole edge after testing. It is may be
due to that the plates we used in tests have not been formed and consequently hardened. Thus, they do not behave as stiff as plates which are used in body of car.
4 Method

Since the geometry and boundary conditions of the problem is complicated, an analytical method seems very difficult to perform. Therefore, the thesis is done by two methods: numerical and experimental.

4.1 Numerical: Finite Element

Finite element method is commonly used to simulate the behaviour of solid structures (Sun et al [17]). Development of this method in different industries has made it the strongest method in analysis of solids. Finite element is now also used in widespread areas of mechanical and electrical engineering. There are many commercial programs which use finite element to analyse problems and by developing the new applications for this method, these programs are improving day by day. Some most famous finite element programs are ANSYS, MSC Nastran, LS-DYNA, RADIOSS, ABAQUS, etc. In this thesis, Hyper Works is used for meshing and post-processing, and LS-DYNA and MD Nastran (MSC) are used as solvers.

The simulation of function of plastic clips in mounting and dismounting processes is a non-linear quasi-static finite element problem. It is non-linear because of the large deformation, the gap-contact conditions, and sometimes, elasto-plastic material, and it is quasi-static because the study of the function of the clips is needed during the process without effect of dynamic loads. In order to solve this non-linear problem, a quasi-static implicit algorithm is used.

4.1.1 Non-linearity

In order to identify a non-linear problem, we need simply recognize if it has linearity conditions. Non-linearity includes a variety of phenomena which may has interaction with another one. For instance, yielding, creep, local buckling, crack propagation, etc. are some reasons of non-linearity in structural mechanics.

The general types of nonlinearity in structural mechanic include the following:

- *Material non-linearity*, in which material properties are functions of the state of stress or strain; e.g. viscoelasticity, plasticity, and creep.
• **Contact non-linearity**, in which gap between adjacent parts may open or close, the contact area between parts changes as the contact force changes, or there is sliding contact with frictional forces.

• **Geometric non-linearity**, in which deformation is that large that equilibrium equations must be written with respect to the deformed structural geometry. Also, loads may change direction as they increase. (Cook et al [1])

These kinds of problems are non-linear because if we consider the structural equation as \([\mathbf{K}]\{\mathbf{D}\} = \{\mathbf{R}\}\), stiffness \([\mathbf{K}]\) or applied forces \(\{\mathbf{R}\}\) are functions of displacement or deformation \(\{\mathbf{D}\}\). We cannot immediately solve for \(\{\mathbf{D}\}\) because information is needed to construct \([\mathbf{K}]\) and \(\{\mathbf{R}\}\) in unknown at first. We need an iterative process to obtain \(\{\mathbf{D}\}\). Then we can construct \([\mathbf{K}]\) and \(\{\mathbf{R}\}\) and solve the equation to determine the new \(\{\mathbf{D}\}\). We should note that when the equation is non-linear, we cannot apply superposition.

Some references, e.g. Zienkiewicz & Taylor [10], consider contact non-linearity as a part of geometric non-linearity.

### 4.1.1.1 Material non-linearity

In order to formulate material non-linearity, it is possible to write for small deformation and inelastic materials as

\[ \varepsilon = \varepsilon^e + \varepsilon^i \]  \hspace{1cm} (4.1)

or

\[ \sigma = \sigma^e + \sigma^i \]  \hspace{1cm} (4.2)

where the elastic part defines as

\[ \varepsilon^e = \mathbf{K}^{-1} \sigma^e \]  \hspace{1cm} (4.3)

in which \(\mathbf{K}\) is the matrix of elastic modulus.

As mentioned above, viscoelasticity is one type of material non-linearity. This phenomenon is characterized by the fact that the rate at which inelastic strains develop depends not only on the current state of stress and strain but
on the full history of their development. Therefore, to determine the increase of inelastic strain over a time step it is necessary to know the state of stress and strain at all preceding time. (Zienkiewicz & Taylor [10])

Classical plastic behaviour of solids is defined by a non-constant relationship between stress and strain which is not dependent on the rate of loading. In fact, one definition of plasticity is the existence of irrecoverable strains on the load removal. Figure 4.1 (a) shows this behaviour for uniaxial loading. Thus, only a non-linear relationship on loading does not determine that non-linear elastic or plasticity occurs.

![Uniaxial behaviour of materials: (a) non-linear elastic and plastic behaviour; (b) ideal plasticity; (c) strain hardening plasticity](Zienkiewicz & Taylor [10]).
Some materials behave approximately as an ideal plastic. Figure 4.1(b) shows that these materials have an elastic behaviour in stresses below yield stress, $\sigma_y$, but in yield stress have indeterminate strains. A modified form of this model is hardening/softening plastic material which in the yield stress depends on the parameter $\kappa$ (such as the accumulated plastic strain $\varepsilon^p$) [Figure 4.1 (c)].

In multiaxial state, the concept of yield should be generalized. We know that, in this state, yielding can occur only if the stress satisfies the general yield criteria

$$F(\sigma, \kappa, \kappa) = 0 \quad (4.4)$$

where $\sigma$ is a matrix form of stress, $\kappa$ represents kinematic hardening parameters and $\kappa$ is isotropic hardening parameter. This yield condition can be shown as an surface in an n-dimensional space of stress (Figure 4.2).

Figure 4.4.2. Yield surface and normality criterion in two-dimentional stress space (Zienkiewicz & Taylor [10]).


4.1.1.2 Contact non-linearity

Contact problem is a kind of geometric non-linearity which arises when different structures, or different surfaces of a single structure, come into contact, separate from each other, or slide on one another with friction. Contact forces should be determined to calculate equilibrium equations. Also, we need to calculate the location and extent of contact because they might be unknown in advance (Cook et al [1]). Usually, the friction between surfaces should be considered. It can follow Coulomb rule or other appropriate theory. These problems can be solved as quasi-static or time-dependent. Further information about contact theory will be explained in section 4.1.3.

4.1.1.3 Geometric non-linearity

Geometric non-linearity occurs when deformation is large enough to change the distribution or direction of applied forces, or internal resisting forces and moments (Cook et al [1]). For example, increase the normal load on a roller in contact with a plate results change in distribution and orientation of reaction forces between roller and plate. The essential difficulty of geometrically non-linear analysis is that equilibrium equations must be written with respect to the deformed geometry which is not known in advance. In other words, the deformed configuration of the body is unknown at the first of an analysis, and therefore, it must be determined as a part of the solution process which is inherently non-linear.

4.1.2 The Quasi-static loading

Since the main aim of this project is the investigation of function of plastic clips during mounting and dismounting processes, it is required to follow the behaviour of clips from start of the contact until complete mounting, and also, from start of pulling until complete dismounting. Therefore, it is needed to use a quasi-static algorithm to achieve the aim of project.

Due to simplification of the problem, the velocity of the clip during mounting and dismounting is assumed constant and the total time (displacement) is divided to several time steps (displacement steps). In every time step, the problem is solved as a steady-state problem.
In both numerical and experimental methods, loads are applied as the vertical displacement of the clip. Change in the displacement from one time step to the next one should be enough small to satisfy the quasi-static loading (movement) condition.

4.1.3 Contact theory

Over the past several decades, the computational modelling and analysis of contact problems have been important subjects of interest, especially using Finite Element methods. Contact problems are inherently non-linear because, before contact, boundary conditions are given by traction conditions (often equal to zero) while during the contact, kinematic constraints must be imposed which prevent penetration of one boundary through other (Zienkiewicz et al. [10]).

In order to solve a contact problem, two actions have to be done. The first action is to identify which points on a boundary interact, and the second one is to impose appropriate conditions to prevent the penetration.

4.1.3.1 Contact position identification

In Finite Element method, there are several methods to identify the contact situation, such as node-node, node-segment, and segment-segment.

In node-node contact, the finite element mesh must be constructed such that boundary nodes on one body (slave nodes) match the location of boundary nodes for the other body (master nodes).

In node-surface, the nodes on the surface of a body (slave) contacts with the surface between nodes (segment) on other body (master). The two-dimensional treatment of this method is shown in Figure 4.3 where the slave node can contact the master segment defined for simplicity in two dimensions by an interpolation. This interpolation may be treated either as the usual interpolation along the edge faces of elements as shown in Figure 4.3 (a) or by an interpolation which smooths the slope discontinuity between adjacent element surface as it is shown in Figure 4.3 (b). To determine the gap, it is necessary to find the point on the master surface which is closest to the slave node. In order to find the nearest point a
numerical method, e.g. Newton-Raphson, might be used (Zienkiewicz et al. [10]).

The segment-segment contact identifying method is based on the mortar method which in, in contrast to node-segment method, the continuity constraints are not enforced at discrete nodal points but they are formulated along the entire coupling boundary in a weak integral sense (Weyler et al. [22]).

![Diagram of node-surface contact](image)

(a)

![Diagram of smoothed interpolation](image)

(b)

Figure 4.3. Node-surface contact (a) element interpolation (b) smoothed interpolation.

In this research, both node-segment and segment-segment methods are used to identify the contact situation.

4.1.3.2 Impenetrability condition: Penalty method

There are several methods to insert *impenetrability* conditions but the most popular ones are Penalty function method and Lagrange multiplier method.

The Lagrange multiplier method of inserting the contact conditions is given simply by multiplying the vertical gap which is formulated below

\[ g = x_2^{(s)} - x_2^{(m)} \]  

(4.5)
by a multiplier. $x_{2}^{(m)}$ and $x_{2}^{(s)}$ are positions of the master node and the slave node, respectively. Accordingly, it can be written for each nodal for each contact

$$\Pi_{c} = \lambda \ g$$  \hspace{1cm} (4.6)$$

where $\lambda$ is identified as a force applied to each node to prevent penetration. When $g \leq 0$, the first derivation of $\Pi_{c}$ will be added to the variational equations being used to solve the problem.

The disadvantage of this method is that the equations in this form introduce a new unknown for each contact pair. Also, the equations are not positive definite and indeed have a zero diagonal for each multiplier term. Thus, especial care is needed in the solution process to avoid divisions by the zero diagonal.

On the other hand, the Penalty method is the most general and the most used interface method to impose impenetrability conditions (Hallquist [5]) which in the contact term is given by

$$\Pi = \kappa \ g$$  \hspace{1cm} (4.7)$$

where $\kappa$ is a penalty parameter (Zienkiewicz et al. [10]). Actually, there is no clear rule to choose the penalty parameter; it depends on particular problems considered. In this method, a spring is considered between master and slave nodes.

The advantage of the penalty method is that this method avoids equation solution difficulties of the Lagrange multiplier method. This method is widely used in complex three dimensional contact-impact problems (Wang et al. [23]). In this method, the final gap will not be zero but becomes a small number depending on the value of the parameter $\kappa$ selected. Therefore, the advantage of Penalty method is somewhat offset by a need to identify the value of the parameter that gives an acceptable answer (Zienkiewicz et al. [10]). In the other words, form an accuracy point of view, the penalty method enforces the constraints only approximately, depending on the value of the penalty parameter chosen, while the
Lagrange multiplier approach enforces the constraints exactly (Pantano & Averill [21], Gu et al. [20]).

Although the penalty method includes remarkable approximations, it is chosen in this thesis to simplify the solution of equations. Also, the friction forces are calculated using coulomb’s theory.

4.1.4 Direct integration method: Implicit algorithm

Dynamic problem can be solved using two categories of methods: modal methods and direct integration methods, but only direct integration methods are suitable for nonlinear problems (Sun et al [17]). Direct integration is based on the calculation of the response history using step-by-step integration in time. Against modal method, this method does not need to change the form of the dynamic equation at first. Responses are evaluated in every separated time step $\Delta t$. At the $n$th time step, the equation of motion is

$$[M]\{\ddot{D}\}_n + [C]\{\dot{D}\}_n + \{R^{\text{int}}\}_n = \{R^{\text{ext}}\}_n$$ \hspace{1cm} (4.8)

or

$$[M]\{\ddot{D}\}_n + [C]\{\dot{D}\}_n + [K]\{D\}_n = \{R^{\text{ext}}\}_n$$ \hspace{1cm} (4.9)

The finite difference approximation can be used to obtain the time derivatives.

In direct integration methods, the conditions at time $n+1$ are calculated using the equation of motion, a difference expression and known conditions at preceding time steps. Algorithms to applying this method can be categorised as explicit or implicit. The explicit algorithm uses a difference expression of

$$\{D\}_{n+1} = f \left( \{D\}_n, \{\dot{D}\}_n, \{\ddot{D}\}_n, \{D\}_{n-1}, \ldots \right)$$ \hspace{1cm} (4.10)

which contains only historical information on its right-hand side. The implicit algorithm uses a difference expression of

$$\{D\}_{n+1} = f \left( \{\dot{D}\}_{n+1}, \{\ddot{D}\}_{n+1}, \{D\}_n, \{\dot{D}\}_n, \{\ddot{D}\}_n, \ldots \right)$$ \hspace{1cm} (4.11)
which is combined with the equation of motion at time \( n+1 \) (Cook et al [1]). The most significant difference between explicit and implicit algorithms is related to stability and economy. Explicit algorithm is \textit{conditionally stable}. It means there is a critical time step size \( \Delta t_{cr} \) which is a maximum limit that must not be exceed. Because \( \Delta t_{cr} \) is very small, many time steps are needed to solve the problem, but calculation for each time step is done quickly. Implicit algorithm is commonly \textit{unconditionally stable}, which means the calculations converge regardless of time step size. Because in explicit method, the coefficient matrix of \( \{D\}_{n+1} \) can be diagonal, the calculation of equations do not cost long time in each time step, but in implicit method, the coefficient matrix cannot be made diagonal and it cause much longer time to solve the equations. As a result, implicit method needs greater CPU which means greater cost (Sun et al, Cook et al [1, 17]).

In direct integration method we have to choose explicit or implicit as the method of solution, while the explicit method need much more time steps with less cost in each step and the implicit method need less time steps with much longer time in each time step. In order to choose the most appropriate method, we can categorize the problem as \textit{wave propagation} type or \textit{structural dynamics} type. A wave propagation problem is created by blast or impact loading, e.g. the car crash analysis, which in high frequency modes must be shown. Structural dynamics problems are created by loads which are applied much more gradually. Lower modes are more significant in these problems. As Yaw [18] has examined, the implicit method gives much more exact results in a non-linear one-dimensional static example (Figure 4.4 (a)). However, the error of explicit method can be reduced by reduction in time step size (Figure 4.4 (b)).
Figure 4.4. Single bar in tension: (a) Comparison of explicit, implicit and exact results for 3 increments (b) explicit results using 20 increments (Yaw [18]).
After this definition, we can have a criterion to choose the most appropriate method (Table 4.1).

<table>
<thead>
<tr>
<th>Types of problem</th>
<th>Methods</th>
<th>Reasons</th>
</tr>
</thead>
<tbody>
<tr>
<td>Structural dynamics</td>
<td>Modal method</td>
<td>• Few needed modes</td>
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<tr>
<td></td>
<td></td>
<td>• Low cost per time step</td>
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<tr>
<td></td>
<td>Implicit method</td>
<td>• Few needed modes</td>
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<tr>
<td></td>
<td></td>
<td>• Numerical stability</td>
</tr>
<tr>
<td></td>
<td></td>
<td>• Easy in non-linearity</td>
</tr>
<tr>
<td>Wave Propagation</td>
<td>Explicit method</td>
<td>• Importance of higher frequencies</td>
</tr>
<tr>
<td></td>
<td></td>
<td>• More modes are needed</td>
</tr>
<tr>
<td></td>
<td></td>
<td>• Easy in non-linearity</td>
</tr>
</tbody>
</table>

*Table 4.1. Suitable methods for different types of problem.*

Because the problem which is investigated in this thesis is a non-linear structural problem and it is analyzed in quasi-static condition, the appropriate method to use is implicit method.

### 4.2 Experimental method

Experiments are used to investigate the real behaviour of clips in mounting and dismounting processes. Also, the results obtained from tests will be compared with numerical results to investigate their accuracy.

A digital Tension-Compression test machine which is programmable for different speeds and loadings is used to execute the tests. Using this machine and a prepared fixture, the clip is pushed into a hole on a fixed steel plate as gradually as we can consider that the motion is quasi-static. The hole on the steel plate has the diameter which the designer has recommended.

The load which is applied on clip is displacement that is discretized in time. Because it is difficult to start the motion from the same point for all tests, it is not possible to apply an exact distance limitation to stop pushing process. Therefore, a 100 N maximum force limitation is applied to stop pushing load and start pulling out the clip.
5 Experiment

5.1 Experiment on Clip #1

In order to study the real function of clips and have a criterion to examine the accuracy of simulations, some experiments were performed on clip #1. These experiments contain mounting and dismounting processes which were done in quasi-static situation and in different conditions, e.g. lubricated clip, dry clip, 0° position, 90° position, etc.

5.1.1 Setup and conditions

In experiment method, a programmable tension-compression testing device is used. The testing machine has a fixed part which the steel plate is fixed on, using two clamps and a thick plate (plate’s fixture) with a hole in center. It has, also, a mobile part which applies load on specimen. Clip is fixed on a fixture which is held by a clamp on mobile part of machine (Figures 5.1 and 5.2).

![Figure 5.1. Setup of experiment on clip #1.](image-url)
The mobile part of the testing machine moves the clip downward with a low and constant speed of 10 mm/min and measures the reaction force from the fixture during the motion. The clip starts to contact with the inner surface of the hole and to enter into the hole. The mobile part continues pushing downward until the measured load reaches the maximum limit of 100 N. When the reaction force reaches 100 N, the machine changes the direction of motion to upward. The mobile part pulls the clip up until the minimum pulling load becomes zero that means the clip is either completely out of the hole or broken.

The clip is fixed in the fixture in two positions which are shown in Figure 5.3, because the strength of the clip’s head is different in these two positions. The clip, also, is experimented both dry and lubricated to study the difference in behaviour. In one of the experiments, I used some fillers to make a gap between the hat of the clip and the steel plate in order to investigate the possibility of use of washers to improve the function of clips.

The steel plates are plates made of carbon steel with dimensions 50x50x0.75 mm. Each plate has a hole at the center with diameter of 8 mm, recommended by the designer. Some tests, also, were done with hole
diameters of 8.2mm, 8.4mm, 8.6mm, 8.8mm and 9.0mm to study the effect of hole diameter on the function of clips.

Figure 5.3. Relative positions of the clip to the fixture 
(a) 0° and (b) 90°.
5.1.2 Performance and results

The first set of experiments on the clips was done in order to study the influence of clip position in 0° and 90°, and using lubrication in contact between the clip and the steel plate. The results are shown in Figure 5.4. The figure shows that the clips in position 90° are stronger than which in position 0° but the lubrication does not have a significant effect on maximum forces. The results from tests in the same conditions are more or less similar and there is no significant difference between them. Therefore the test seems enough repeatable.

![Figure 5.4. Force-displacement curve for 0° and 90° positions of dry and lubricated clips.](image)

The second set of experiments was done using two steel washers with thicknesses 0.22 mm and 0.44. This experiment was performed to see if using washers helps the dismounting of the clip. Figure 5.5 shows how using washers reduces the maximum pulling force.
The third set of experiments on clip #1 was performed with different diameters of the holes on the steel plates. This experiment was done with 6 different hole diameters and the results are shown in Figure 5.6. The Figures 5.6 and 5.7 demonstrate that the diameter of hole has an important role in decrease of maximum forces.

**Figure 5.5.** Force-displacement curves from experiments with washers with thickness 0.22 mm and 0.44 mm.

**Figure 5.6.** Force-displacement curves for different diameters of the hole.
5.2 Experiment on Clip #2

In order to have a criterion to investigate the accuracy of simulations and study the real function of clips, some experiments was performed on clip #2. These experiments contain mounting and dismounting processes which were done in the quasi-static condition.

5.2.1 Setup and conditions

The setup of experiment on clip #2 is almost the same as what we had for clip #1 (Section 5.1). However the fixture was changed to another one which was more suitable for dimensions of clip #2. The steel plate which is used in this experiment had dimensions of 50x50x1.6 and a hole with diameter of 7.4 at the center (thickness of the plate and diameter of the hole are recommended by the designer). All experiments on clip #2 were done in

Figure 5.7. Maximum force in pushing and pulling for different diameters of hole.
the same natural (dry) condition. The speed of motion in this experiment was 20 mm/min which was enough low to consider that the loading was quasi-static.

5.2.2 Performance and results

The experiment on clip #2 contains two parts. In first part, 6 clips was mounted in and dismounted from 6 similar steel plates, one-by-one, and the reaction force was recorded by testing machine in every displacement. Then, the force-displacement curves was extracted which are seen in Figure 5.8. The figure shows that the results are more or less similar and there is no significant difference between different tests. Therefore the test seems enough repeatable.

![Force-displacement curves from experiment on clip #2.](image)

One of the remarkable phenomena happens to clip in mounting and dismounting process is wearing. Wearing occurs on the outer surfaces of wings and especially on two steps which are created on top of the wings. Figure 5.9 (a) shows how the clips are damaged during the experiment. Also, the POM chips which remain after experiment is shown in Figure 5.9 (b).
In the second part, the behaviour of one clip was studied while it was mounted and dismounted three times, consecutively. Figure 5.10 demonstrates the force-displacement curves extracted from this part of the experiment. This figure shows the plastic deformation which occurs during the first mounting and dismounting of clip which effects significantly on both pushing and pulling forces. The maximum pulling force in second
dismounting falls by approximately 60%. There is also a similar reduction in second mounting force.

Figure 5.10. Force-displacement curves from experiment for repeated mounting and dismounting.
6 Simulation

6.1 Simulation of Clip #1

In order to simulate clip #1, first, the mirror-symmetric model was simulated using LS-DYNA 971-4.2.1 but because of some problems, which it will be explained in coming subsections, the simulation was repeated with new meshing and by two other solvers: LS-DYNA 971-5.1.1 and MD-Nastran.

Figure 6.1. CAD model of clip #1.

6.1.1 Model A: Complete Model by LS-DYNA 971-4.2.1

In this part, first of all, the 3D CAD model of clip #1 (Figure 6.1) was meshed automatically by 4-node linear tetrahedral elements. The steel plate was meshed by linear hexahedral made by dragging the shell elements, which were created on a circular surface including a hole with 8 mm diameter. Then, a ring meshed by 4-node linear quadrilateral element around the neck of clip which is called rigid plate was created (Figure 6.2). Also, a 2D beam element was created normal to the rigid plate and with one node shared with rigid plate. The load (displacement) was applied on the free end of the beam element. The axial force applied on element beam was measured in every displacement step. The rigid plate transfers the point load from beam element to clip as a distributed load. In this model, the rigid
plate can only move vertically along the neck of clip. The material of steel plate was defined as carbon steel first but because of some difficulties and in order to simplifying the problem, in all models, the material of the steel plate is considered a rigid material. This model is called model A.

Figure 6.2. Clip #1 meshed by tetrahedral solid element and steel plate meshed by hexahedralsolid elements(Model A).

<table>
<thead>
<tr>
<th>Part</th>
<th>Type of Elements</th>
<th>Quantity of elements</th>
</tr>
</thead>
<tbody>
<tr>
<td>Clip</td>
<td>4-Node (Linear) Tetrahedral Solid</td>
<td>18944</td>
</tr>
<tr>
<td>Steel plate</td>
<td>8-Node (Linear) Hexahedral Solid</td>
<td>5211</td>
</tr>
<tr>
<td>Rigid plate</td>
<td>4-Node (Linear) Quadrilateral Shell</td>
<td>92</td>
</tr>
<tr>
<td>Beam Element</td>
<td>2-Node 2D Beam</td>
<td>1</td>
</tr>
<tr>
<td><strong>Total</strong></td>
<td></td>
<td><strong>24248</strong></td>
</tr>
</tbody>
</table>

Table 6.1. Quantities and types of elements used in model A.
Table 6.1 shows the quantities and types of elements which are used in this model. Figure 6.3 shows the loading curve applied on the beam element in this simulation. We should notice that time in this curve is not the real physical time but also, it is a duration which in clip moves inside and outside of the hole and the solver considers it as time. Therefore it cannot have a physical unit.

The results obtained from the simulation shows some significant penetrations which occur between wings of the clip and the steel plate, and also, between hat of the clip and the steel plate (Figure 6.4). These penetrations resist against releasing or sliding between two contact surfaces and cause a large concentrated stress at penetration points which results non-supposed elastic/plastic deformations. Penetrations, also, increase releasing (disconnecting) and friction forces between connected parts.

![Figure 6.3. Loading curve applied on model A.](image)

6.1.2 Model B: Mirror-Symmetric Model by LS-DYNA 971-4.2.1

Because there was some significant penetration in model A which affect seriously on the results, some changes was made in the finite element model. As it is shown in Figure 6.5, clip #1 has two symmetry planes which are named symmetry plane xz and symmetry plane yz. The symmetry plane xz was not used because it prevents bending of body of clip in yz plane, while in reality this bending is not negligible.
Figure 6.4. Penetrations occur in model A (a) between clip’s wings and steel plate (b), and between clip’s hat and steel plate.

Therefore, in the first step, the CAD model was cut by symmetry plane xy (Figure 6.6).
The second change in model was change in element type. The tetrahedral elements were replaced by hexahedral linear elements. The reason was that the linear tetrahedral elements do not work well and they are undesirably stiff (Cook et al [1]). However, the type of the element may not affect on penetration.

Figure 6.5. CAD model of clip #1 with symmetry planes.

Figure 6.6. CAD model of clip #1 cut by symmetry plane yz.
The third change which was applied at the same time with meshing was refining the elements. The element size was decreased to increase the accuracy of simulation and decrease the undesirable sticking.

![Figure 6.7. Model B.](image)

<table>
<thead>
<tr>
<th>Part</th>
<th>Type of Elements</th>
<th>Quantity of elements</th>
</tr>
</thead>
<tbody>
<tr>
<td>Clip</td>
<td>8-Node (Linear) Hexahedral Solid</td>
<td>20735</td>
</tr>
<tr>
<td>Steel plate</td>
<td>4-Node (Linear) Quadrilateral Shell</td>
<td>3800</td>
</tr>
<tr>
<td>Rigid plate</td>
<td>4-Node (Linear) Quadrilateral Shell</td>
<td>4103</td>
</tr>
<tr>
<td>Beam Element</td>
<td>2-Node 2D Beam</td>
<td>1</td>
</tr>
<tr>
<td><strong>Total</strong></td>
<td></td>
<td><strong>28639</strong></td>
</tr>
</tbody>
</table>

*Table 6.2. Quantities and types of elements used in model B.*
The next step in modifying the model was use of rigid linear quadrilateral shell elements with thickness instead of solid elements to mesh the steel plate. This change decreases the total number of elements and reduces the time of solving equations in every time step. Therefore, it reduces the total time of solution. It also makes the contact situation much simpler. The clip and plates are constrained as a plane symmetric model. Figure 6.7 shows the new model which is named model B. Quantities and types of elements are demonstrated in Table 6.2.

This model was solved using the loading curve which is shown in Figure 6.8. This curve was applied to reduce the change of displacement in critical step sizes in order to decrease the undesirable sticking between the clip and the steel plate. The maximum displacement was also reduced to decrease the penetration between hat of the clip and the steel plate. The material of the clip in this solution was assumed the pure elastic POM.

Some important contact factors of LS-DYNA used in this solution are shown in Table 6.2.
<table>
<thead>
<tr>
<th>Factor</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Clip - Rigid plate Contact</td>
<td>Contact_Automatic_Surface_To_Surface</td>
</tr>
<tr>
<td>Clip – Steel plate Contact</td>
<td>Contact_Surface_To_Surface_Smooth</td>
</tr>
<tr>
<td>SOFT</td>
<td>1</td>
</tr>
<tr>
<td>SFS</td>
<td>0.1</td>
</tr>
<tr>
<td>SOFSCL</td>
<td>0.05</td>
</tr>
<tr>
<td>FS</td>
<td>0.0001</td>
</tr>
<tr>
<td>FD</td>
<td>0.0001</td>
</tr>
</tbody>
</table>

Table 6.2. Contact factors of LS-DYNA for initial solution of model B.

The results that are obtained from this solution are the forces which are measured in beam element with respect to vertical displacement. The force-displacement curve is shown in Figure 6.9. The maximum pushing and pulling forces in this curve are about 300 N and 380 N, respectively, which are not reasonable because we supposed that the clip should be mounted and dismounted by human’s hand force easily without any damage in other parts but these forces are too high (Appendix II). The pushing force is also much higher than what we obtained from experiments (Sections 5.1). The curve shows that dismounting was not completed because the equations could not converge at stop point.

![Figure 6.9. The force-displacement curve obtained from initial solution of model B.](image-url)
6.1.2.1 Parametric study

Due to solve the significant contradiction between results, it was necessary to find which factors affect on measured forces. In this way, the contact factors were changed one-by-one and the obtained results were studied.

First, both static and dynamic coefficients of friction in both connections rigid plate-clip and steel plate-clip were increased from 0.0001 to 0.2 but no significant difference between the two force-displacement curves was seen (Figure 6.10).

Figure 6.10. The force-displacement curve with and without friction.

In the next step, the effect of change of scale factor of surface stiffness (SOFSCL) in contact between the clip and the steel plate is investigated. Figure 6.11 shows no change in the force-displacement curve.
Figure 6.11. The force-displacement curve with different SOFSCL in the clip-steel plate contact.

Also, the model B was solved with different scale factor on default penalty stiffness (SFS) for contact between clip and steel plate. The force-displacement curves of these simulations are also the same (Figure 6.12).

Figure 6.12. The force-displacement curve with different SFS in the clip-steel plate contact.
Then, the model was solved with change of SOFSCL for the contact between the clip and the rigid plate. Figure 6.13 demonstrates that both pushing and pulling loads rise when SOFSCL increase. However, in some simulations, equations did not converge and solution did not become completed. Figure 6.14 shows that how pushing force changes by SOFSCL in the sample displacement z=-6mm.

Figure 6.13. The force-displacement curve with different SOFSCL in the clip-rigid plate contact.

Figure 6.14. Change in force with respect to SOFSCL in the clip - rigid plate contact.
After all, the changes in force were studied by change in SFS for the contact between the clip and the rigid plate. The results are shown in Figure 6.15. The curve does not change by the change in SFS.

![Figure 6.15. The force-displacement curve with different SFS in the clip-rigid plate contact.](image)

The investigations gave the result that inappropriate factors of the contact between the rigid plate and the clip is one of the sources of error in the force-displacement curve obtained from simulation. Therefore, some factors of this contact were changed to achieve a reasonable condition for simulation. Table 6.3 contains the appropriate factors which were obtained to have reasonable results from simulation.

<table>
<thead>
<tr>
<th>Contact</th>
<th>Factor</th>
<th>value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Clip with Steel Plate</td>
<td>SOFT</td>
<td>1</td>
</tr>
<tr>
<td></td>
<td>SOFSCL</td>
<td>0.05</td>
</tr>
<tr>
<td></td>
<td>SFS</td>
<td>1</td>
</tr>
<tr>
<td>Clip with Rigid Plate</td>
<td>SOFT</td>
<td>0</td>
</tr>
<tr>
<td></td>
<td>SOFSCL</td>
<td>0.02</td>
</tr>
<tr>
<td></td>
<td>SFS</td>
<td>0.15</td>
</tr>
</tbody>
</table>

*Table 6.3. The appropriate contact factors for model B.*
We can see the force-displacement curve obtained by these factors in Figure 6.16.

\[\text{Figure 6.16. The final force-displacement curve of model B using the elastic material.}\]

In order to simplify the problem, it was assumed that the material is purely elastic, but now it is time to solve the model by more realistic material that follows an elasto-plastic curve. The force-displacement curve for the elasto-plastic material is shown in Figure 6.17.

\[\text{Figure 6.17. The final force-displacement curve of model B using the elasto-plastic material.}\]
Now, we have a solution which is reasonable in comparison with experiment (Sections 5.1 and 7). However, in most of simulations which in the clip could not go out of the hole, the equations did not converge and simulations stopped because of much iteration to achieve a converged result. Also, due to many elements and the small time step size, solutions took long time and it was not very affordable if we want to solve many models. Therefore, a model with coarser meshes and less time consuming was needed.

6.1.3 Model C: Full Model by MD-Nastra and LS-DYNA 971-5.1.1

Because model B was very expensive and had some problem in the contact between the rigid plate and the clip, a new complete model was made with less and coarser elements. In this model, which is called model C, the complete CAD model of clip is meshed using linear hexahedral solid elements and the rigid plate is not free but attached to the top part of the hat of the clip to remove problems occurred in the contact between the rigid plate and the clip. The steel plate, also, is meshed by hexahedral elements. As well as models A and B, this model contains a beam element to apply loads (displacement) and measure the vertical reflecting forces. Figure 6.18 demonstrates this model and Table 6.4 shows the quantities and types of elements used in it.

![Figure 6.18. Model C of clip #1.](image-url)
<table>
<thead>
<tr>
<th>Part</th>
<th>Type of Elements</th>
<th>Quantity of elements</th>
</tr>
</thead>
<tbody>
<tr>
<td>Clip</td>
<td>8-Node (Linear) Hexahedral Solid</td>
<td>6060</td>
</tr>
<tr>
<td>Steel plate</td>
<td>8-Node (Linear) Hexahedral Solid</td>
<td>1920</td>
</tr>
<tr>
<td>Rigid plate</td>
<td>4-Node (Linear) Quadrilateral Shell</td>
<td>108</td>
</tr>
<tr>
<td>Beam Element</td>
<td>2-Node 2D Beam</td>
<td>1</td>
</tr>
<tr>
<td>Total</td>
<td></td>
<td>8089</td>
</tr>
</tbody>
</table>

*Table 6.4. Types and quantities of elements used in model C.*

In order to solve this model, MD-Nastran and LS-DYNA 971-5.1.1 were used as solvers. In LS-DYNA 971-5.1.1 a new contact algorithm called Mortar is used which uses the segment-segment method to identify the contact. This type of contact simulation is very good in preventing penetration and undesirable sticking.

We can see the results of these two solutions in Figure 6.19.

![Figure 6.19. The force-displacement curves obtained from simulation of model C by LS-DYNA 5.1.1 and MD-Nastran.](image-url)
In these simulations, the clip is dismounted without any problem. The forces obtained from these two simulations are approximately similar but they are almost half of which are obtained from simulation by LS-DYNA 4.2.1 and experiments (Sections 6.1.2 and 5.1). Therefore, because these solutions are only used to compare the different properties of the problem, the values of force which are obtained from these two solvers are multiplied by a correction factor of 2.

6.1.3.1 Parametric study

As it was mentioned in section 6.1.2 and 5.1, clip #1 needs a high force to be dismounted and in the most of times either it cannot dismounted or dismounted with damages. To find that which properties of the clip and the steel plate affect the function of the clip and how much is the value of effects, the Young’s modulus (E) of the clip, the friction coefficients of the contact between the clip and the steel plate (FS and FD) for both purely elastic and elasto-plastic materials, and diameter of the hole on the steel plate for elasto-plastic material was changed. Then, the model was solved using LS-DYNA 5.1.1 and after that, the results were shown in some graphs.

![Graph showing maximum forces vs. Young's modulus for the elasto-plastic material.](image)

*Figure 6.20. The maximum forces vs. Young’s modulus for the elasto-plastic material.*
First, the Young’s modulus of the clip with the elasto-plastic material was changed and the maximum forces were recorded in mounting and dismounting. Then, the curve of maximum forces with respect to Young’s modulus was drawn. Figure 6.20 shows that the pushing force rises by increase of Young’s modulus while the pulling force falls.

As the next step, the maximum forces for different friction coefficients were obtained. It is assumed that the static and the dynamic friction coefficients are identical. Figure 6.21 shows the maximum forces vs. the friction coefficient for the elasto-plastic material. It is seen that the maximum pulling force falls by reduction of the friction coefficient but surprisingly, it rises sharply when the friction coefficient becomes very low.

![Figure 6.21. The maximum forces vs. friction coefficient for the elasto-plastic material.](image)

These two investigations were repeated for the purely elastic material. The obtained curves are demonstrated in Figures 6.22 and 6.23. In Figure 6.22 it is seen that both pulling and pushing forces increase by the increase of stiffness. Figure 6.23, also, shows that the maximum forces for the elastic material follows a curve which is more or less similar to what was obtained for the elasto-plastic material.
Figure 6.22. The maximum forces vs. Young modulus for elastic material.

Figure 6.23. The maximum forces vs. friction coefficient for elastic material.

Also, the effect of increase in diameter of hole is investigated for elasto-plastic material. The related curves are shown in Figure 6.24. These curves show that both pulling and pushing forces decline by the increase of
diameter of the hole. The decrease in pulling forces is much more impressive.

![Figure 6.24. The maximum forces vs. friction coefficient for the elasto-plastic material.](image)

These graphs will be interpreted and compared with experimental results in coming sections.
6.2 Simulation of Clip #2

![CAD model of clip #2.](image)

The second clip which is studied in this thesis is named clip #2. The CAD model of this clip (Figure 6.25) was meshed with linear hexahedral solid elements. The steel plate was, also, meshed using dragging the shell elements created on upper surface of it in vertical direction. This model, which is named \textit{model D}, contains a rigid plate meshed by linear quadrilateral shell elements and a 2D beam element to apply the load and measure the vertical reaction force, as well as model C.

![Model D.](image)
Model D is shown in Figure 6.26, and quantities and types of elements used in this model are listed in Table 6.5.

<table>
<thead>
<tr>
<th>Part</th>
<th>Type of Elements</th>
<th>Quantity of elements</th>
</tr>
</thead>
<tbody>
<tr>
<td>Clip</td>
<td>8-Node (Linear) Hexahedral Solid</td>
<td>13016</td>
</tr>
<tr>
<td>Steel plate</td>
<td>8-Node (Linear) Hexahedral Solid</td>
<td>1500</td>
</tr>
<tr>
<td>Rigid plate</td>
<td>4-Node (Linear) Quadrilateral Shell</td>
<td>175</td>
</tr>
<tr>
<td>Beam Element</td>
<td>2-Node 2D Beam</td>
<td>1</td>
</tr>
<tr>
<td><strong>Total</strong></td>
<td></td>
<td><strong>14692</strong></td>
</tr>
</tbody>
</table>

*Table 6.5. Quantities and types of elements used in model D.*

This model was solved using LS-DYNA 971-5.1.1. The same contact factors as for model C are used (Section 6.1.3). In the first simulation, the model was solved with the elasto-plastic material and its force-displacement is seen Figure 6.27.

*Figure 6.27. Force-displacement curve of model D.*
In this simulation, the clip could be completely dismounted but some plastic deformations can be recognized on the steps which are created on top of the wings. These deformations are shown in Figure 6.28 (b).

Figure 6.28. Plastic deformation in steps: (a) mounting, (b) dismounting

Also, a general plastic deformation occurs in the entire wings which can be seen in Figure 6.29. These plastic deformations are the reasons of reduction of pulling and pushing force in second use of the clip, as it was seen in experiments (Section 5.2).

Figure 6.29. Plastic deformation of wings.
6.2.1 Parametric study

The second step in simulation of clip #2 is solving model D with different friction coefficients in the contact between the clip and the steel plate, and the comparison between the obtained results. Figure 6.30 shows how the maximum forces in pushing and pulling, change by friction coefficient. The unexpected phenomenon in this figure is again the raise of forces where the friction coefficient has values lower than 0.1 and 0.05 in pulling and pushing, respectively.

![Figure 6.30. Maximum force in pushing and pulling vs. friction coefficient](image)

As the next step, the effect of the changes in Young’s modulus (E) on the maximum forces in mounting and dismounting processes was investigated. The changes in the maximum force against Young’s modulus are seen in Figure 6.31.
Finally, the material properties of clip #2 were changed from elasto-plastic to purely elastic. In this case, the clip could not go out of the hole because no plastic deformation occurs and the non-deformed steps do not allow the clip to be dismounted.

*Figure 6.31. Maximum force in pushing and pulling vs. Young’s modulus.*
7 Results

7.1 Results of Clip #1

All simulations of model B show that the clip cannot be dismounted (Section 6.1.2). The visual investigation of experiments on clip #1 accepts these results, too. Table 7.1 gives some statistic information from visual investigation of clip #1. It shows that the most of clips have been completely broken during dismounting. Also, it demonstrates that and only 12% of clips could be dismounted completely. However, all of them have been damaged.

<table>
<thead>
<tr>
<th>Description</th>
<th>Quantity</th>
<th>Percent</th>
<th>Remark</th>
</tr>
</thead>
<tbody>
<tr>
<td>Performed tests</td>
<td>18</td>
<td>---</td>
<td>---</td>
</tr>
<tr>
<td>Acceptable tests</td>
<td>16</td>
<td>---</td>
<td>---</td>
</tr>
<tr>
<td>Setup with angle $\theta=0^\circ$</td>
<td>8</td>
<td>50%</td>
<td>---</td>
</tr>
<tr>
<td>Setup with angle $\theta=90^\circ$</td>
<td>8</td>
<td>50%</td>
<td>---</td>
</tr>
<tr>
<td>Complete fracture in clip</td>
<td>14</td>
<td>88%</td>
<td>In pulling off</td>
</tr>
<tr>
<td>Clips completely dismounted</td>
<td>2</td>
<td>12%</td>
<td>Only in $\theta=90^\circ$</td>
</tr>
<tr>
<td>Damaged clips</td>
<td>16</td>
<td>100%</td>
<td>---</td>
</tr>
</tbody>
</table>

*Table 7.1. Statistic results from first experiments on clip #1 using visual investigation.*

Table 7.2 contains average maximum pushing and pulling forces obtained from both simulation and experiment. It shows how much is the error between values from simulation and experiment in each part. It is seen that there is no significant difference between averages of maximum forces in experiments and simulations. However, very small error in pulling force has happened incidentally. Of course, the error between pushing forces is 26% that is remarkable.
<table>
<thead>
<tr>
<th>Average of maximum pushing force</th>
<th>Simulation</th>
<th>34 N</th>
<th>Error=26%</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Experiment</td>
<td>46 N</td>
<td></td>
</tr>
<tr>
<td>Average of maximum pulling force</td>
<td>Simulation</td>
<td>554 N</td>
<td>Error=0.002%</td>
</tr>
<tr>
<td></td>
<td>Experiment</td>
<td>551 N</td>
<td></td>
</tr>
</tbody>
</table>

Table 7.2. Average of maximum forces in pulling and pushing from the experiment on clip #1.

The force-displacement curves extracted from simulations of model B with the purely elastic and the elasto-plastic materials using LS-DYNA and experiments are compared in Figure 7.1. The figure shows that the curve related to elasto-plastic material is more similar to curve extracted from experiment. It means approximation of considering POM as an elastic material seems not to be reasonable.

![Figure 7.1. Force-displacement curve from simulations of model B and experiment.](image)

Figure 7.1. Force-displacement curve from simulations of model B and experiment.

Figure 7.2 compares the force-displacement curves obtained from simulations of model C using LS-DYNA 5.1.1 and MD-Nastran (Section 6.1.3) with the curve obtained from experiment of clip#1 (Section 5.1). These curves demonstrate that the results of two solvers are more or less similar. However, they are not completely correlated with experimental results. The reason is that in experiment, clips mostly could not be
dismounted and become broken but in the simulation of model C, they are dismounted completely.

![Force-displacement curve from simulations of model C using MD-Nastran and LS-DYNA 5.1.1.](image)

**Figure 7.2. Force-displacement curve from simulations of model C using MD-Nastran and LS-DYNA 5.1.1.**

In Figure 7.3 we can see the changes in the maximum forces in mounting and dismounting for elastic and elasto-plastic materials with respect to the modulus of elasticity. The figure shows that the rate of changes in maximum forces for the elastic material and the elasto-plastic one are completely different. Maximum pulling force rises by increase of Young modulus when we assume POM is an elastic material while it falls when consider the plasticity for material. Although the maximum forces rise in pushing for both materials, the rate of the increase in the elastic material is more than what is in elasto-plastic, remarkably.
Figure 7.3. Maximum forces vs. Young modulus in pulling and pushing for both elasto-plastic and elastic materials.

Figure 7.4. Maximum forces vs. friction coefficient in pulling and pushing for both elasto-plastic and elastic materials.

Figure 7.4 compares the changes in maximum reaction force in pulling and pushing processes vs. the friction coefficient in the purely elastic and the
elasto-plastic materials. The natural friction coefficient in the contact between POM and the carbon steel is about 0.2. As we suppose, the maximum force reduces in all four cases. However, the maximum forces in pulling processes fall much more rapidly. The surprising phenomenon in this figure is that the maximum pulling forces increase suddenly when the friction coefficient becomes less than 0.01 so that the maximum forces for the frictionless contact become even more than which for contact with actual friction.

Figure 7.5 demonstrates that if we increase the diameter of the hole, what happens to maximum forces in pulling and pushing, and also, by simulation and experiment. Although the curve of maximum pulling force obtained from simulation has a little fluctuation, in both experiment and simulation, the maximum pulling force reduce by the same rate. This is while, we do not see any significant change in pushing process, neither in simulation nor in experiment.

![Figure 7.5. Maximum forces vs. diameter of hole from simulation and experiment.](image)

The visual investigation of clips in this experiment has come in Table 7.3. The table shows that for all diameters of hole equal or more than 8.0 mm,
the clip can dismounted completely but only for holes with diameter 8.8 mm and more the clips are not damaged. For holes with diameters up to 8.2 mm, some fracture occurs, especially in connection between the neck and the head of clips.

<table>
<thead>
<tr>
<th>Dia. (mm)</th>
<th>Qty. of tests</th>
<th>Angle of clip</th>
<th>Gone out of hole</th>
<th>Damaged clips</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>8.0</td>
<td>8</td>
<td>90°</td>
<td>25%</td>
<td>100%</td>
<td>Fracture and wear in clip, plastic deformation in plate</td>
</tr>
<tr>
<td>8.2</td>
<td>2</td>
<td>90°</td>
<td>100%</td>
<td>100%</td>
<td>Fracture and wear in clip, plastic deformation in plate</td>
</tr>
<tr>
<td>8.4</td>
<td>2</td>
<td>90°</td>
<td>100%</td>
<td>100%</td>
<td>Wear on clip, plastic deformation in plate</td>
</tr>
<tr>
<td>8.6</td>
<td>2</td>
<td>90°</td>
<td>100%</td>
<td>100%</td>
<td>Wear on clip, plastic deformation in plate</td>
</tr>
<tr>
<td>8.8</td>
<td>2</td>
<td>90°</td>
<td>100%</td>
<td>0%</td>
<td>Little wear on clip, little plastic deformation in plate</td>
</tr>
<tr>
<td>9.0</td>
<td>2</td>
<td>90°</td>
<td>100%</td>
<td>0%</td>
<td>No damage in clip and plate</td>
</tr>
</tbody>
</table>

*Table 7.3. Statistic results from visual investigation of clip #1 in experiments with different diameter of hole.*

In Figure 7.6 we can see the effect of using washers between the hat of clip and the steel plate on maximum pulling and pushing forces. As it was expected, adding washers does not affect on pushing force but declines the pulling force.
Figure 7.6. Maximum forces vs. thickness of washer using between hat of clip and steel plate, from experiment.

7.2 Results of Clip #2

The visual investigation of clips #2 which has done during experiments shows that although all clips can come out of the hole, all of them have been damaged during testing; specially, in dismounting (Section 5.2.2).

<table>
<thead>
<tr>
<th></th>
<th>Quantity</th>
<th>Percent</th>
<th>Remark</th>
</tr>
</thead>
<tbody>
<tr>
<td>Performed tests</td>
<td>6</td>
<td>---</td>
<td>---</td>
</tr>
<tr>
<td>Acceptable tests</td>
<td>6</td>
<td>---</td>
<td>---</td>
</tr>
<tr>
<td>Complete fracture in clip</td>
<td>0</td>
<td>0%</td>
<td>---</td>
</tr>
<tr>
<td>Clips came completely out</td>
<td>6</td>
<td>100%</td>
<td>---</td>
</tr>
<tr>
<td>Damaged clips</td>
<td>6</td>
<td>100%</td>
<td>Wear on wings and steps</td>
</tr>
</tbody>
</table>

Table 7.4. Statistic results from first experiments on clip #2 using visual investigation.
Table 7.4 contains the statistic information which are recorded in the first experiment on clip #2.

<table>
<thead>
<tr>
<th>Average of maximum pushing force</th>
<th>Simulation</th>
<th>51 N</th>
<th>Error=28%</th>
</tr>
</thead>
<tbody>
<tr>
<td>Experiment</td>
<td>70 N</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Average of maximum pulling force</th>
<th>Simulation</th>
<th>323 N</th>
<th>Error=10%</th>
</tr>
</thead>
<tbody>
<tr>
<td>Experiment</td>
<td>360 N</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

*Table 7.5. Average of maximum forces in pulling and pushing from experiment on clip #2.*

Table 7.5 contains average maximum pushing and pulling forces obtained from both simulation and experiment. It shows how much is the error between values from simulation and experiment in each part. We can see that there is a reasonable difference between averages of maximum forces in experiments and simulations but the error between pushing forces is remarkable.

![Force-displacement curve from simulation and Experiment](image)

*Figure 7.6. Force-displacement curve from simulation and Experiment.*
In Figure 7.6 two curves are seen which are force-displacement curves related to results obtained from the simulation and the experiment. The figure shows that the shapes of curves are more or less similar. However, they are not completely coincident. It means that we can say that simulation has enough accuracy.

The changes in maximum pushing and pulling forces are shown in Figure 7.7. The maximum pulling force declines by the decrease in the friction coefficient, and as well as we saw for clip #1, it increases again in friction coefficients in neighbourhood of zero. However, for this clip, increase in force starts when the friction coefficient is approximately equal to 0.1. Almost the same thing happens for pushing force but reduction has smaller rate and the raise starts from friction coefficient 0.05.

![Figure 7.7. Maximum force in pushing and pulling vs. friction coefficient.](image)

Finally, in Figure 7.8, it is seen that the maximum pulling force falls by the raise of Young modulus with a declining rate while pushing force increases by an approximately linear rate.
Figure 7.8. Maximum force in pushing and pulling vs. Young modulus.
8 Discussions and Suggestions

8.1 Discussions

- Although experiments show plastic deformations around the hole on steel plates, to simplify the problem in the simulation, the steel plate has been considered as rigid plate otherwise the simulation became more complicated and had some convergency problems.

- Because in model B, which has almost the real conditions, there are several sources of non-linearity, it is very difficult to obtain a reasonable set of results. As it was noticed in Section 6.2.1, most of errors in model B come from the contact between the rigid plate and the clip which is one of the important sources of non-linearity. Therefore, we have more stable results in model C (Section 6.1.3), despite its coarser elements. However, the reaction forces obtained from solution of this model need to be modified using a correction factor.

- We cannot accurately judge that which factor in model C makes the model more stable because this model is completely different to model B. The element type, the type of rigid plate - clip contact, the solver and the contact algorithm have been changed in model C. Probably, for their part, all these factors affect on modifying of the model but to find the effect of each one it is necessary to examine changes in them one-by-one.

- Use of symmetry in non-linear problems requires to be sure that significant changes will not occur in the geometry as well as forces directions in the direction of normal to the symmetry plate. Usually, in non-linear problems, it is difficult to predict the behaviour of the model. In many cases, the models seems symmetric but during the process, conditions change and make the models asymmetric. Therefore, we must be very careful when we want to use symmetric conditions.

- In some cases, such as complicated geometries, change in loading curve can be useful in order to reduce the penetration. By changing in loading curve, we can control the value of the increase of the load in each step and it prevents the large displacement in each step. As a
result, the amount of penetration decreases in each step and it will be more controllable. The disadvantage of this method is the time consuming and expensive solution.

- One of important reasons of the penetration in contact simulations is the big difference between the size of master and slave elements. Therefore, in models which in we need to have fine elements in one part, e.g. in complicated geometries, all parts in contact with this part must have almost the same size of elements at contact surfaces or contact lines. It causes higher number of elements and consequently longer time and more expensive solution.

- It does not look very reasonable that in model A the only factor effects on force-displacement curve is SOFSCL in contact between the rigid plate and the clip. Figure 7.4 shows that in Finite Element model, the maximum forces for friction coefficient equal to zero and 0.2 are approximately identical. Therefore, probably, it is the reason that force-displacement curve does not change in these two frictional conditions. However, the most important reason can be that the penalty force in contact between the rigid plate and the clip is that too much that the effect of changes in factors of the contact between the steel plate and the clip become negligible.

- As a comparison between using MD-Nastran and LS-DYNA (Section 6.1.3), the results are more or less similar and there is no significant difference in time cost but MD-Nastran makes bigger result files and it needs more memory to save. Therefore, for big models, MD-Nastran needs much more memory to save files than LS-DYNA.

- In Figures 7.4 and 7.7 it can be seen that the maximum pulling forces for both clip #1 and clip #2 unexpectedly increase when friction coefficient becomes almost zero. Although some experiments were performed to investigate the effect of lubrication, there was not any test available to measure the friction coefficient between the clip and the steel plate. If there was, we could investigate that if the same increase happens in experiment. However, it is most probable that this unexpected increase in force occurs because of some problem in Finite Element model, such as coarse elements. Investigation of this behaviour can be itself a subject of a research. By the way, decrease in friction coefficient can somewhat reduce the maximum pulling force.
Figure 7.3 shows that the effect of change in modulus of elasticity in the simulation of clip #1 with the purely elastic material is completely different to which in elsto-plastic one. The increase in Young’s modulus decreases the value of the maximum pulling force in the elsto-plastic clip while it increases the maximum pulling force in purely plastic one. Also, in clip #2, when Young’s modulus rises, the maximum pulling force falls. This behaviour is not expected because we suppose that increasing the Young’s modulus causes the increase in the reaction forces, as we see in pushing forces. The reason might be that with lower stiffness, the surface of the clip goes to plastic phase more easily and it makes the surface uneven. When we increase the elasticity modulus, the surface will be harder, and consequently, more even. To check the validity of these results, it is necessary to do some experiments on materials with different stiffness and compare the results.

One of the most important problems occur in using clips is high dismounting force. We expect that the clip can be mounted and dismounted easily by human’s force while it should fix parts together enough strongly. Also, during mounting and dismounting it supposed to do not occur any failure in parts and even in the clip but simulations and experiments show that the dismounting forces in both clips are much more than we suppose and it might destroy parts during dismounting. Therefore, we need to find some ways to reduce the dismounting forces without degradation in the fixing function of clips.

In the investigation of the effect of the hole diameter on the maximum forces, we can see that the results from simulation and experiment are very similar (Figure 7.5). It shows that we can trust these results to modify the designing. As we expected, both pulling and pushing forces fall by increase in hole diameter. However, we can see a negligible fluctuation in the simulation which might be because of coarse elements. Also, the difference in rate of reduction in pulling and pushing is remarkable.

As we noticed in experiments and simulations, both clip #1 and clip #2 were damaged after dismounting. These damages are more considerable in clip #1. It means we cannot use the clips again after dismounting. Figure 5.10 shows that how pushing and pulling forces decline in the second and the third times of use in comparison with
the first use. Since many different clips are used in a vehicle and their shape, size and properties are completely different, it increases the cost of maintenance if clips cannot be reusable.

### 8.2 Suggestions

- The most simple and most effective way to modify function of clip #1 is increasing diameter of holes on the plate. This change not only decreases the pulling load, but also reduces the damages occur in clips. Increase in diameter of hole may not be appropriate for clip #2 because of the steps there are on top of their wings.

- The experiments show that using a washer with an appropriate thickness between the clip and the steel plate affect on dismounting force. Therefore, using washers is also a suggestion to improve the function of clips.

- Another suggestion to modify the function of clips is to modify the plastic material. The results of this research show that using a material with higher stiffness and lower friction coefficient improves the function of clips, especially in dismounting. This improvement in material can be obtained by using some additives to row material during manufacturing.

- As it was studied, the clips will be deformed in plastic mode during the first use. Therefore, to have better function, it is recommended that to use a new clip for remounting the parts.

- In order to reduce the cost of the manufacturing and the maintenance, it is recommended to use clips, as similar as possible, because the sameness in clips makes the assembly process simpler and decreases the costs.
9 Conclusion

In this project, the function of the two plastic clips chosen randomly from many types of plastic clips that are used in SAAB automobiles was studied. This study was done both numerically and experimentally.

The parameter which has been studied in the thesis is vertical reaction force which applied to the support which holds the clip. This force has been investigated with respect to the vertical displacement of clip.

The similarity between results obtained from experiments and which achieved from numerical simulation shows that the results from simulation are largely reliable. However, some remarkable approximations are included in Finite Element simulation.

Some parametrical investigations has been done on clips which show that how the behaviour of plastic clips are been affected with respect to these parameters. These parameters are the modulus of elasticity of clips, the friction coefficient between the clip and the steel plate, the thickness of washers (if be used between the clip and the steel plate), and the diameter of holes on steel plates. The investigations were carried out numerically and experimentally, and then the results were compared together.

The parametric investigations show that the function of plastic clips can be modified by change of these parameters in an appropriate way.

9.1 Future work

- This thesis has focused on analysis of reaction forces applied from clips to the supports which hold them and some visible deformations on clips and plates. In order to undertake further studies, stress analysis of clip and steel plate can be investigated to find which parts of clips tolerate more stress and which part goes to plastic mode during mounting and dismounting processes. This study can be more complete with investigation of deformation clips and plates.
- The simulations and experiments have been done in quasi-static motion, while the load that applies on clips in reality is usually an impact load. The study of clips under impact loading is required to
achieve more accurate information about function of plastic clips. Since this study will be a dynamic analysis, it is recommended to use explicit direct integration method instead of implicit.

- In order to investigate the function of different Finite Element solvers in contact problems, as a research, the problem can be solved using different solvers. The comparison between the results and also between difficulties may occur during analysis, examines the ability and accuracy of those solvers.

- The effects of changes in modulus of elasticity and friction coefficient have been only studied numerically in this thesis. These effects can be investigated experimentally as a future work to examine the results which are obtained from simulation.

- The steel plates and the supports of clips have been assumed as rigid materials while they are not rigid in reality. The further studies can investigate the function of clips in contact with other parts as an analysis of a system of components.
10 References


Appendix I: Material Properties

<table>
<thead>
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<th>Product Description</th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Ultraform® H 2320 006 UNC Q600 Polyoxymethylene</td>
<td></td>
</tr>
</tbody>
</table>

**Applications**

Ultraform H 2320 006 UNC Q600 is a POM with high molecular weight grade for injection molding.

**Physical**

<table>
<thead>
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<th>Property</th>
<th>ISO Test Method</th>
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<td>1.4</td>
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<td>Mold Shrinkage, parallel, %</td>
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<td>2.1</td>
</tr>
<tr>
<td>Mold Shrinkage, normal, %</td>
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<td>2.1</td>
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<tr>
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<td>(50% RH)</td>
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<tr>
<td>(Saturation)</td>
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**Rheological**

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**Mechanical**

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<td>Tensile stress at yield, MPa</td>
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<td>23°C</td>
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<td>Tensile strain at break, %</td>
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<tr>
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**Impact**

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<td></td>
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<td></td>
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<td>-40°C</td>
<td>0.5</td>
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<td>Charpy Unnotch, kJ/m</td>
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<td>280</td>
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<td>HDT A, °C</td>
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</table>
Ultraform® H 2320 006 UNC Q600

Coefficient of Linear Thermal Expansion, Parallel: 1.1 X10^-4

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<tr>
<td>Dielectric Strength, KW/mm</td>
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</table>

Processing Guidelines

Material Handling
Max. Water content: 0.15%
Product is supplied in polyethylene bags and drying prior to molding is not required. However, after long storage or when handling material from previously opened containers, preliminary drying is recommended in order to remove any moisture which has been absorbed. If drying is required, a dehumidifying or desiccant dryer operating at 90 - 110 degC (178 - 230 degF) is recommended. Drying time is dependent on moisture level, but 2-3 hours is generally sufficient.

Further information concerning safe handling procedures can be obtained from the Material Safety Data Sheet. Alternatively, please contact your BASF representative.

Typical Profile
Melt Temperature: 190-230 degC (375-440 degF)
Mold Temperature: 60-120 degC (140-248 degF)
Injection and Packing Pressure: 30-70 bar (500-1000 psi)

Processters
Injection speed must be optimized. A filling time which is too high results in anisotropic mechanical properties, while a filling time which is too low yields parts with poor surface finish. The tool must be vented to avoid burn marks and prevent mold deposits. Injection pressure controls the filling of the part and should be applied for 100% of ram travel. Packing pressure affects the final part and can be used effectively in controlling sink marks and shrinkage. It should be applied and maintained until the gate area is completely frozen off.

Back pressure can be utilized to provide uniform melt consistency and reduce trapped air and gas. Minimal back pressure should be utilized to prevent glass breakage, recommended.

Fill Rate
Injection speed must be optimized. A filling time which is too high results in anisotropic mechanical properties, while a filling time which is too low yields parts with poor surface finish. The tool must be vented to avoid burn marks and prevent mold deposits.

Note
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Technical Assistance: 800-527-TECH (734-324-5150)
Web address: http://www.plasticsportal.com/usa

Data sheet of POM, Page 2 (O.BASF ).

80
Tensile stress-strain curve of POM in 23°C and 40°C (O.BASF).
Tensile stress-strain curve of POM in 0°C and -20°C (O.BASF).
Appendix II: Human’s Hand forces

### Hand and thumb-finger strength (N)

<table>
<thead>
<tr>
<th>Degree of elbow flexion (rad)</th>
<th>Pull</th>
<th>Push</th>
<th>Up</th>
<th>Down</th>
<th>In</th>
<th>Out</th>
</tr>
</thead>
<tbody>
<tr>
<td>L&lt;sup&gt;aa&lt;/sup&gt;</td>
<td>R&lt;sup&gt;aa&lt;/sup&gt;</td>
<td>L</td>
<td>R</td>
<td>L</td>
<td>R</td>
<td>L</td>
</tr>
<tr>
<td>π</td>
<td>222</td>
<td>231</td>
<td>107</td>
<td>222</td>
<td>40</td>
<td>62</td>
</tr>
<tr>
<td>5/6 π</td>
<td>187</td>
<td>249</td>
<td>133</td>
<td>187</td>
<td>57</td>
<td>80</td>
</tr>
<tr>
<td>2/3 π</td>
<td>151</td>
<td>137</td>
<td>115</td>
<td>150</td>
<td>107</td>
<td>93</td>
</tr>
<tr>
<td>1/2 π</td>
<td>142</td>
<td>165</td>
<td>98</td>
<td>180</td>
<td>76</td>
<td>89</td>
</tr>
<tr>
<td>1/3 π</td>
<td>116</td>
<td>107</td>
<td>06</td>
<td>161</td>
<td>67</td>
<td>89</td>
</tr>
</tbody>
</table>

### Arm Strength (lb)

<table>
<thead>
<tr>
<th>Degree of elbow flexion (deg)</th>
<th>Pull</th>
<th>Push</th>
<th>Up</th>
<th>Down</th>
<th>In</th>
<th>Out</th>
</tr>
</thead>
<tbody>
<tr>
<td>L&lt;sup&gt;a&lt;/sup&gt;</td>
<td>R&lt;sup&gt;a&lt;/sup&gt;</td>
<td>L</td>
<td>R</td>
<td>L</td>
<td>R</td>
<td>L</td>
</tr>
<tr>
<td>180</td>
<td>50</td>
<td>62</td>
<td>42</td>
<td>60</td>
<td>0</td>
<td>14</td>
</tr>
<tr>
<td>160</td>
<td>42</td>
<td>68</td>
<td>30</td>
<td>42</td>
<td>16</td>
<td>10</td>
</tr>
<tr>
<td>120</td>
<td>34</td>
<td>42</td>
<td>28</td>
<td>36</td>
<td>17</td>
<td>24</td>
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<td>90</td>
<td>32</td>
<td>37</td>
<td>35</td>
<td>35</td>
<td>17</td>
<td>20</td>
</tr>
<tr>
<td>60</td>
<td>38</td>
<td>34</td>
<td>22</td>
<td>34</td>
<td>16</td>
<td>20</td>
</tr>
</tbody>
</table>

### Hand and thumb-finger strength (lb)

<table>
<thead>
<tr>
<th>Hand grip</th>
<th>Thumb-finger grip (Palmer)</th>
<th>Thumb-finger grip (tips)</th>
</tr>
</thead>
<tbody>
<tr>
<td>L</td>
<td>R</td>
<td>L</td>
</tr>
<tr>
<td>Momentary hold</td>
<td>66</td>
<td>60</td>
</tr>
<tr>
<td>Sustained hold</td>
<td>30</td>
<td>35</td>
</tr>
</tbody>
</table>

Note: * L=Left; R=Right

Human’s arm, hand and thumb-finger strength (NASA [9]).