Structural Optimization of Product Families
With Application to Vehicle Body Structures

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Some products share one or two modules and while developing these products, structural optimization with stiffness as the objective function can be a useful tool. There might be no or very little CAD-data available in the pre-development phase and it is not certain that existing designs can be, or is desirable to use as a reference. The main objective of this thesis is to establish an accurate and fast-to-use methodology which can be utilized while developing new cars.

In this thesis, the Volvo products S40, V50 and C70 serve as a base for this case study. All the models are beam structures and the masses of components are added as point and line masses. Several optimization analyses are performed on one or three products exposed to seven load cases. Additional analyses with shell elements, more simplified models and changed load case balance achieved by normalization of the different load case compliances are also studied to investigate how these factors influence the results.

Analyses show that front crash to a great extent dominates the results while normalization increases the influence of the remaining load cases. Since front crash is dominating and the front area is shared in all products, the performance is remarkably similar when three products are optimized compared to separate analyses of one product. Analysis of models without added point or line masses gives a result which greatly differs from previous results and therefore shows that added masses are required. The methodology is applicable to develop products and detect new load paths through the car.
Preface

This thesis is the final assignment before the examination in Master of Science in mechanical engineering. The work has been performed at Volvo Car Corporation and Linköpings tekniska högskola.

We would like to thank our supervisors Bo Torstenfelt and Harald Hasselblad for all the help and interest during this time. We would also like to thank the employees at Volvo for the useful discussions.

Linköping in February 2006

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Abstract

Some products share one or two modules and while developing these products, structural optimization with stiffness as the objective function can be a useful tool. There might be no or very little CAD-data available in the pre-development phase and it is not certain that existing designs can be, or is desirable to use as a reference. The main objective of this thesis is to establish an accurate and fast-to-use methodology which can be utilized while developing new cars.

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Contents

1 Introduction ........................................................................................................ 1
   1.1 Background .................................................................................................. 1
   1.2 Objectives .................................................................................................. 2
   1.3 Restrictions ............................................................................................... 2

2 Volvo Cars ........................................................................................................ 3
   2.1 The products ............................................................................................... 3
      2.1.1 V50 .................................................................................................. 3
      2.1.2 S40 .................................................................................................. 4
      2.1.3 C70 .................................................................................................. 4

3 Theory .............................................................................................................. 5
   3.1 The finite element method ......................................................................... 5
      3.1.1 Trinitas ............................................................................................. 5
   3.2 Structural optimization ............................................................................. 5
   3.3 Method of moving asymptotes .................................................................. 7
      3.3.1 General description of the method .................................................... 7
      3.3.2 The dual sub problem ...................................................................... 8
   3.4 Application of the theory ......................................................................... 10
   3.5 Area moments of inertia ........................................................................... 11

4 Modeling ......................................................................................................... 13
   4.1 Modeling of the products ......................................................................... 13
      4.1.1 Submodels ....................................................................................... 13
      4.1.2 Point and line masses ..................................................................... 14
      4.1.3 Design variable limits ..................................................................... 15
      4.1.4 Final models ................................................................................... 15
   4.2 Load cases ................................................................................................. 18
      4.2.1 Bending ............................................................................................ 18
      4.2.2 Twisting ........................................................................................... 19
      4.2.3 Roll-over .......................................................................................... 19
      4.2.4 Rear crash ....................................................................................... 20
      4.2.5 Front crash ..................................................................................... 20

5 Analyses of the products ............................................................................. 23
   5.1 Study of the created models .................................................................... 23
      5.1.1 Engine suspension .......................................................................... 23
      5.1.2 Other point masses ......................................................................... 23
      5.1.3 Design variables ............................................................................. 24
      5.1.4 Front crash ..................................................................................... 24
   5.2 Optimization analyses ............................................................................. 24
   5.3 Selected sensitivity analyses .................................................................... 24
      5.3.1 Normalization .................................................................................. 25
      5.3.2 Model with shell elements ............................................................... 25
      5.3.3 New load paths .............................................................................. 26
      5.3.4 Simplified models .......................................................................... 26
6 Results

6.1 Analyses of one product and one load case ................................................. 27
6.2 Analyses of one product and seven load cases ............................................. 29
6.3 Analysis of three products and seven load cases .......................................... 31
6.4 Normalization ............................................................................................... 33
6.5 Model with shell elements .......................................................................... 33
6.6 New load paths ............................................................................................. 34
6.7 Simplified models .......................................................................................... 34
6.8 Rotation and principle area moments of inertia ............................................. 35
   6.8.1 Section 1 .................................................................................................. 36
   6.8.2 Section 2 .................................................................................................. 37
   6.8.3 Section 3 .................................................................................................. 38

7 Discussion

7.1 Analyses of single products .......................................................................... 39
7.2 Analysis of three products and seven load cases .......................................... 40
7.3 Normalization ............................................................................................... 41
7.4 Model with shell elements .......................................................................... 41
7.5 New load paths ............................................................................................. 41
7.6 Simplified models .......................................................................................... 42

8 Conclusions

References
1 Introduction

1.1 Background

Often the short time of pre-development of new cars is very critical and the decisions can be crucial but also, if relevant made, they can lead to reduced production development cost and time. The competition between car-manufactures makes it important to shorten the pre-development time, and the customers want a variety of products to choose between. Therefore the possibility of sharing modules between products is important to observe.

A method which is easy to use, safe and accurate enough can be utilized to come to relevant early decisions with more certainty. It is also desirable to be able to use the method to try new solutions in the body of an already existing car, and from there make changes in the more complicated and detailed CAD- or FEM-models.

In the early phases of developing new cars, structural optimization can be a useful tool. There might be no or very little CAD-data available and it is not certain that existing designs can be, or is desirable to use as a reference.

Previous work in this area is the dissertation “Topology Optimization of Vehicle Body Structures” by Harald Hasselblad [1]. This master thesis is a continuation and handles platform optimization and graphical user interface which is discussed in the dissertation, chapter 10 – Further research.

It is possible to optimize over one or several products, called a product family with respect to the mass or the stiffness. The family is exposed to different load cases and is optimized with design variables associated to the cross section geometries of beams.

1.2 Objectives

The main objective of this thesis work is to establish an accurate and user-friendly methodology which can be utilized while developing new cars. The idea is to obtain a first outline of a model in an early phase of the developing process, where no or very little CAD-data are available. A useful tool in this context can be structural optimization over one or several products exposed to different load cases.

It is going to be investigated what an individual product loses in performance while sharing different parts with other products. It is also interesting to study the different load cases separately to observe how each one of them affects the result and the final structures.

Another desirable application is to use the methodology in order to find new alternative ways for loads to transfer by using topology optimization. Additional analyses with shell elements, normalization and more simplified models are also to be studied to investigate how these factors influence the results.
1.3 Restrictions

The structures which are studied and analyzed in this work are mainly built up as beam models. The existing car bodies are in fact built up by shells spot-welded together. In some reinforced parts the shells form structures which can be simulated as beams.

There are three different types of optimization analyses. In this study, size and topology optimization are being used. These types of optimization allow the size of the cross sections to change and, in the topology case, reduce to zero. Shape optimization, where nodes are allowed to move, is possible to use in Trinitas but has not been utilized in this work.

When optimizing, two different objective functions can be used. Either the compliance is minimized with a mass constraint or the analysis is performed by minimizing the mass with a restriction of the compliance value. In this case only the first alternative is used.
2 Volvo Car Corporation

Volvo was founded by Assar Gabrielsson and Gustaf Larson in 1924 [2]. Three years later the first series-built car, the ÖV4, left the factory in Gothenburg. Since then Volvo has become one of the best-known car brands in the world. Over 13 million cars have been produced over the years. Since 1999 Volvo has been wholly owned by Ford Motor Company. Together with Jaguar, Land Rover and Aston Martin, Volvo is part of Ford’s premium car division.

The total amount of employees was in the end of 2004 about 27 500. Almost 20 000 work in Sweden. The head office is located in Gothenburg, where also the departments in design and development, marketing and administration are. The manufacture of cars is concentrated at the production in Torslanda, Gothenburg and Ghent, Belgium.

Volvo sells cars in over 100 countries and the four largest markets are USA, Sweden, Great Britain and Germany. In 2004 about 450 000 cars were sold. The best-selling car model was XC90, where nearly half the amount was sold in USA. The second best-selling car was V70, which is the most popular Volvo car in Sweden.

2.1 The products

The cars studied in this thesis are the V50, the old version of S40 and C70.

2.1.1 V50

Just before the end of 2003, a new Volvo was launched, Volvo V50 [3]. Volvo V50, see Figure 1, is the replacement for the earlier model Volvo V40. The exterior design of Volvo V50 has a clear Volvo identity and this product continued the strong Volvo heritage as a manufacturer of estate cars. This can be identified by the shape of the rear of the car and also the high mounted tail lamps.

Figure 1. Volvo V50

Torsion stiffness has been increased by almost 70% and contributes to a high level of safety. Concerning safety, the Volvo V50 has got a new, patented front structure with several crumple zones for increased protection of the occupants.
2.1.2 S40
The older version of S40, see Figure 2, was first produced in 1995 [3]. Soon, the two original versions were supplemented with new economic and exciting models. The old S40 was manufactured until 2004 and in September 2003 a new generation with a more modern styling of the sedan product was launched.

Figure 2. The old version of Volvo S40

2.1.3 C70
The story of Volvo convertibles is as old as the Volvo car itself [3]. The first Volvo car was an open car, the ÖV4. The C70 Convertible was presented in 1997, see Figure 3. Volvo had for long hesitated to make an open car for safety reasons. But, thanks to the Volvo ROPS (Roll Over Protection System) innovation, safety was integrated even if the open car should turn upside down.

Figure 3. Volvo C70
3 Theory

3.1 The Finite element method

Often the solid mechanic problems are too complicated to solve by analytical methods which is why the finite element method is used [4]. In this way general differential equations can be solved in an approximate manner. The region, which is being considered, is divided into several elements called finite elements. It is being assumed that the displacements vary in a linear or higher order fashion over each element. The behaviour of each element can be established and are patched together to form the entire region. This makes it possible to obtain an approximate solution for the entire region.

3.1.1 Trinitas

The FEM-program used for the structural optimization problems is Trinitas. Trinitas is created by Bo Torstenfelt and is an integrated graphical stand alone finite element program for conceptual design and optimization [5].

The program includes activities for geometry modeling, boundary condition definition, mesh generation, simulation and evaluation of the results. Boundary conditions can be defined on the geometry and/or on the mesh. Trinitas also handles structural optimization problems with one or several products where the user controls the adjustments of the analysis. This is the routine used in this thesis.

3.2 Structural optimization

One or several structures are being optimized with respect to different load cases with mass as the objective function and stiffness as a constraint or vice versa [6]. Structural optimization is, in this thesis, done with respect to size and topology.

Figure 4 shows the most simple product family structural optimization problem with two products and three modules. Out of these modules, two are unique and one is shared by both products. Three levels are identified. The product family is level zero, the product is level one and the module is level two. Every module contains several elements.

![Figure 4. A product family consisting of two products](image)
When the problem is formulated, a vector of design variables is created.

\( y(\alpha) \) is the design variable vector for product \( \alpha \) on level one.
\( x(\alpha,i,e) \) is the design variable vector for product \( \alpha \), module \( i \) and element \( e \) on level two.

To formulate a design problem, mathematical functions are needed. Here, the compliance is minimized with mass as a constraint. The compliance is the inverse of the stiffness.

For every chosen load case \( l \) and product \( \alpha \), the stiffness is calculated. For discretized structures a displacement vector \( u^l_\alpha \) and a force vector \( F^l_\alpha \), are received. The compliance is defined as

\[
C^l_\alpha = (F^l_\alpha)^T u^l_\alpha.
\]

\( u^l_\alpha \) and \( F^l_\alpha \) are related through a symmetric positive definite stiffness matrix \( K^l_\alpha \) under the paradigm of linear elastic displacements.

It may be assumed that both the stiffness matrix as well as force vector depend on level two design variables. The size of \( K \) depends on the product and what load case is being investigated. \( K \) is assumed to be non-singular.

\[
K^l_\alpha = K^l_\alpha(x(\alpha)) \quad \text{and} \quad F^l_\alpha = F^l_\alpha(x(\alpha))
\]

The state equation is written as

\[
F^l_\alpha(x(\alpha)) = K^l_\alpha(x(\alpha))u^l_\alpha. \quad (1)
\]

Since \( K^l_\alpha(x(\alpha)) \) is non-singular, (1) can be solved and \( u^l_\alpha \) is regarded as a function of the design \( x(\alpha) \).

\[
u^l_\alpha = u^l_\alpha(x(\alpha)) = K^{-1}_\alpha(x(\alpha)) F^l_\alpha(x(\alpha)) \quad (2)
\]

The compliance of product \( \alpha \) and load case \( l \) can be written as a function of the design \( x(\alpha) \):

\[
C^l_\alpha(x(\alpha)) = (F^l_\alpha(x(\alpha))^T u^l_\alpha(x(\alpha)). \quad (3)
\]

It is now possible to formulate an optimization problem. Assume that there are \( n_p \) products in the product family which is going to be optimized.

If there are \( n_l(\alpha) \) load cases for product \( \alpha \), the total compliance or performance is

\[
C(x) = \sum_{\alpha=1}^{n_p} \sum_{l=1}^{n_l(\alpha)} w^l_\alpha C^l_\alpha(x(\alpha)), \quad (4)
\]

where \( w^l_\alpha \) are weighted factors that reflect the relative importance of different products and load cases. The mass or the constraint is

\[
M(x) = \sum_{\alpha=1}^{n_p} M_\alpha(x(\alpha)). \quad (5)
\]
3.3 Method of moving asymptotes

The method used for structural optimization Trinitas is MMA, the method of moving asymptotes [7], which is well-known and often used. MMA is developed to be flexible and general, both with respect to design variables and constraints. It is important that the method is stable and generates a sequence of improved feasible solutions of the problem.

3.3.1 General description of the method

\[
\begin{align*}
P: \quad \text{minimize} & \quad f_0(x) \quad x \in \mathbb{R}^n \\
\text{subject to} & \quad f_i(x) \leq \hat{f}_i \quad \text{for } i = 1, \ldots, m \\
\text{and} & \quad x_j \leq x_j \leq \bar{x}_j \quad \text{for } j = 1, \ldots, m
\end{align*}
\]

\(f_0(x)\) is the objective function, in this thesis, the compliance of the structure. \(x = (x_1, \ldots, n)^T\) is the vector of design variables and \(x_j, \bar{x}_j\) are lower and upper bounds on the design variables. \(f_i(x) \leq \hat{f}_i\) are behaviour constraints, i.e. the mass of the structure.

In MMA, each \(f_i^{(k)}\) is obtained by a linearization of \(f_i\) in variables of the type

\[
y_j = \frac{1}{x_j - L_j} \quad \text{or} \quad y_j = \frac{1}{U_j - x_j}
\]
dependent on the sign of the first order derivatives of the objective and constraints. The values of the lower and upper bounds, \(L_j\) and \(U_j\), are normally changed between iterations, they are moving asymptotes.

For each \(i = 0, 1, \ldots, m\) \(f_i^{(k)}\) is defined by

\[
f_i^{(k)}(x) = r_i^{(k)} + \sum_{j=1}^n \left( \frac{p_{ij}^{(k)}}{U_j^{(k)} - x_j^{(k)}} + \frac{q_{ij}^{(k)}}{x_j^{(k)} - L_j^{(k)}} \right),
\]

where \(q_{ij}^{(k)} = \begin{cases} 
(U_j^{(k)} - x_j^{(k)})^2 \frac{\partial f_i}{\partial x_j}, & \text{if } \frac{\partial f_i}{\partial x_j} > 0 \\
0, & \text{if } \frac{\partial f_i}{\partial x_j} \leq 0 
\end{cases}\)

and \(q_{ij}^{(k)} = \begin{cases} 
0, & \text{if } \frac{\partial f_i}{\partial x_j} \leq 0 \\
-(x_j^{(k)} - L_j^{(k)})^2 \frac{\partial f_i}{\partial x_j}, & \text{if } \frac{\partial f_i}{\partial x_j} < 0 
\end{cases}\)

\(r_i^{(k)} = f_i(x^{(k)}) - \sum_{j=1}^n \left( \frac{p_{ij}^{(k)}}{U_j^{(k)} - x_j^{(k)}} + \frac{q_{ij}^{(k)}}{x_j^{(k)} - L_j^{(k)}} \right)\).
In MMA, the values of \( L_j^{(k)} \) and \( U_j^{(k)} \) are always finite.

Now the sub problem \( P^{(k)} \) can be written as:

\[
P^{(k)}: \quad \text{minimize} \quad \sum_{j=1}^{n} \left( \frac{p_{oj}^{(k)}}{U_j^{(k)} - x_j} + \frac{q_{oj}^{(k)}}{x_j - L_j^{(k)}} \right) + r_o^{(k)}
\]

subject to \[
\sum_{j=1}^{n} \left( \frac{p_{ij}^{(k)}}{U_j^{(k)} - x_j} + \frac{q_{ij}^{(k)}}{x_j - L_j^{(k)}} \right) + r_i^{(k)} \leq f_i, \quad \text{for } i = 1, \ldots, m
\]

and \[
\max \{ \sum_j \alpha_j^{(k)} \} \leq x_j \leq \min \{ \sum_j \beta_j^{(k)} \} \quad \text{for } j = 1, \ldots, n
\]

\( \alpha_j^{(k)} \) and \( \beta_j^{(k)} \) are called move limits and are chosen such as \( L_j^{(k)} < \alpha_j^{(k)} < x_j^{(k)} < \beta_j^{(k)} < U_j^{(k)} \), to avoid division by zero.

### 3.3.2 The dual sub problem

To solve the sub problem, a dual method can be used since \( P^{(k)} \) is a convex and separable problem. The Lagrangian function is defined by

\[
L(x, \lambda) = f_0^{(k)}(x) + \sum_{i=1}^{n} \lambda_i f_i^{(k)}(x).
\] (7)

\( \lambda \) are Lagrange multipliers. The index \( k \) is unnecessary since the sub problem is defined for each iteration.

The function equals

\[
r_o - \lambda^T b + \sum_{j=1}^{n} \left( p_{oj} + \lambda^T p_j \right) \frac{x_j - L_j}{U_j - x_j} + \sum_{j=1}^{n} L_j \left( x_j, \lambda \right)
\] (8)

where

\[
b = (b_1, \ldots, b_m)^T, \quad p_j = (p_{1j}, \ldots, p_{mj})^T
\]

\[
q_j = (q_{1j}, \ldots, q_{mj})^T, \quad \lambda = (\lambda_1, \ldots, \lambda_m)^T
\]

and

\[
L_j(x_j, \lambda) = \frac{p_{oj} + \lambda^T p_j}{U_j - x_j} + \frac{q_{oj} + \lambda^T q_j}{x_j - L_j}.
\] (9)

The dual objective function is defined for \( \lambda \geq 0 \) as

\[
W(\lambda) = \min \{ L(x, \lambda); \alpha_j \leq x_j \leq \beta_j \quad \text{for all } j \} \nonumber
\]

\[
= r_o - \lambda^T b + \sum_{j=1}^{n} W_j(\lambda)
\]

and

\[
W_j(\lambda) = \min \{ L_j(x_j, \lambda); \alpha_j \leq x_j \leq \beta_j \}.
\]
The dual problem corresponding to $P^{(k)}$ is

D: \[ \text{maximize } W(\lambda) \]
subject to \[ \lambda \geq 0 \]

The minimizing $x_j$ that depends on $\lambda$ will be found. This $x_j$ is denoted by $x_j(\lambda)$. The derivative of (7) with respect $x_j$ to is shown below.

\[
L_j'(x_j, \lambda) = \left( \frac{p_{0j} + \lambda^T p_j}{U_j - x_j} \right)^2 + \frac{q_{0j} + \lambda^T q_j}{(x_j - L_j)^2} \tag{10}
\]

Since $L_j'(x_j, \lambda)$ is strictly increasing in $x_j$ there are some possibilities for the minimum of $L_j$.

1. If $L_j'(\alpha_j, \lambda) \geq 0$, then $x_j(\lambda) = \alpha_j$

2. If $L_j'(\beta_j, \lambda) \leq 0$, then $x_j(\lambda) = \beta_j$

3. If $L_j'(\alpha_j, \lambda) < 0$ and $L_j'(\beta_j, \lambda) > 0$, then there is a unique solution of $x_j(\lambda)$ which gives $L_j'(x_j, \lambda) = 0$.

The unique solution is

\[
x_j(\lambda) = \frac{\left( p_{0j} + \lambda^T p_j \right)^{1/2} L_j + \left( q_{0j} + \lambda^T q_j \right)^{1/2} U_j}{\left( p_{0j} + \lambda^T p_j \right)^{1/2} + \left( q_{0j} + \lambda^T q_j \right)^{1/2}}.
\]

The dual objective function is

\[
W(\lambda) = r_o - \lambda^T b + \sum_{j=1}^n \left( \frac{p_{0j, \lambda^T p_j}}{U_j - x_j(\lambda)} + \frac{q_{0j, \lambda^T q_j}}{x_j(\lambda) - L_j} \right).
\]
3.4 Application of the theory

In this thesis, the car bodies of three products will be analyzed. The car bodies consist of beams with rectangular cross sections. How the products are modeled is being presented in chapter 4. The product family consists of three products and there is one module which is shared between all products, one module shared between two products and three unique modules.

Topology and size optimization problems are the structural optimization problems used in this study. The objective function is the compliance with the mass as a constraint. The modeled mass as an upper limit of the constraint permits the compliance to be compared between different products and load cases after implemented analyses.

The design variables are the restrictions on how the modeled beams are allowed to vary during optimization as

\[
B_{\text{min}} \leq B \leq B_{\text{max}} \\
H/B_{\text{min}} \leq H/B \leq H/B_{\text{max}} \\
T/B_{\text{min}} \leq T/B \leq T/B_{\text{max}} \\
\theta_{\text{min}} \leq \theta \leq \theta_{\text{max}}
\]

![Figure 5. A hollow rectangular cross section](image)

The weight factor \(w_j\) = 1 in every load case, except when normalization is studied.
### 3.5 Area moments of inertia

When studying the results of analyses it is desirable to compare area moments of inertia in three chosen sections, seen in Figure 6. This is to achieve some local quantitative results in addition to the global compliance. The figure also shows some terms that will be used in this thesis.

![Figure 6. Sections where the moments of inertia are of interest](image)

The area moments of inertia [8] are defined as

\[
I_y = \int_A z^2 \, dA = \frac{TH^3}{6} + \frac{1}{2} TBH^2
\]

\[
I_z = \int_A y^2 \, dA = \frac{TB^3}{6} + \frac{1}{2} TB^2 H
\]

The dimensions of the moments are \([\text{m}^4]\) and they are always positive.

The area moments of inertia are calculated in Trinitas and each element has a local coordinate system. Therefore one analysis will serve as a reference model and the rotation \(R\) in the other analyses relative this model will be studied.

![Figure 7. Rotation relative reference model](image)
4 Modeling
To be able to design the models, which are going to be analyzed, reference cars are needed. In this thesis, the Volvo products S40, V50 and C70 are used.

4.1 Modeling of the products
The geometries of the models are obtained by collecting nodes and points from existing FEM-models at Volvo. All the models are beam structures and the used material is 1550-01 steel with the parameters \(E = 205 \text{ GPa}, \rho = 7790 \text{ kg/m}^3 \text{ and } \nu = 0.3\), where \(E\) is the elasticity modulus, \(\rho\) is the density of the material and \(\nu\) is Poisson ratio.

All beams have hollow rectangular cross sections as seen in Figure 5. Complicated cross sections are assumed to be rectangular. All cross sections are the same before analyzed. The measures of the cross sections are: \(B = 0.05 \text{ m}, H = 0.05 \text{ m} \text{ and } T = 0.002 \text{ m}\).

Since the structures consist of only beams, it is necessary to add masses. Theses masses are added as point or line masses depending on how the mass is connected to the structure. The masses and the nodes, to where in the structure the masses are connected, are collected from existing Volvo FEM-models. This is to add them as correctly as possible. Relevant masses are those which are easy to model and contribute with more than ten kilos. Point masses are for instance chairs, doors and components in the front area, which are joined together as one point mass. Roof and floor, for instance, are masses which are added as constant mass distribution per unit length. The cars also consist of many small components which also contribute to the total mass. These are in most cases added to modeled masses, such as the roof and the floor.

4.1.1 Submodels
The three products which are being studied in this thesis are all built on the same platform. Because of this, some modules of the structures are shared, by all products or by two products. The front module is common for all three structures, Figure 8(a), while product S40 and product V50 also share the B-pillar and partly the roof, Figure 8(b).

To obtain the same geometry for the shared modules, the products are divided into five modules called submodels. These submodels are modeled separately and are then patched together to form final products.

(a) Shared submodel of all products  (b) Shared submodel of S40 and V50

Figure 8. Shared submodels
The shared submodels are combined with the unique submodels in Figure 9 to establish the final products.

![Unique submodel of S40](image1)
![Unique submodel of V50](image2)
![Unique submodel of C70](image3)

Figure 9. Unique submodels

The masses visible in the submodels are those which are possible to add in this early stage. The point masses are connected to the structure by bars. The bars do not contribute with any mass and have an adjusted stiffness to avoid non-physical contributions to the global stiffness. They are also excluded from taking part in the optimization.

**4.1.2 Point and line masses**

According to Volvo, when the car is standing still, the weight from the engine will put most loads on the bolts; see Figure 10, point A. When the car is exposed to front crash, some load is also put on the bushings; see point B and C. To simulate this load distribution, the structure which holds the engine and other parts in the front area, is modeled as in the following figure.
The point mass is supported with bars mirrored to the plane to eliminate the risk of singular stiffness matrices, see Figure 11. This method is used for the engine, doors, fuel tank, spare wheel and instrument panel. The doors have fixed displacements perpendicular to the length of the car, to prevent them from opening. The rear seat is divided into three point masses for S40 and V50 and into two point masses for C70. All products carry four passengers.

The luggage doors in all unique submodels and the rear window for the unique submodel of S40 have been modeled as a constant mass distribution per unit length. In the unique submodel of C70, torsion sheets are modeled which are important in the case of twisting.

### 4.1.3 Design variable limits

Relevant limits for the design variables; width, relative height, relative thickness and twisting of the beams have to be decided. As references of the upper limits of the widths, the existing cross sections of beams are used. The remaining limits of the design variables are preliminary and are studied in analyses. Relevant limits are achieved when not too many elements reach their highest or lowest allowed value.

### 4.1.4 Final models

The submodels are connected to create the complete models, according to Figure 12. Remaining line masses, which put load on several submodels, are modeled. Such as roof, floor, side panels and windshield, see Figure 13.
Figure 12. Connection of submodels

(a) S40

(b) V50
Wheels and their suspensions are not modeled. Instead, constraints are used to avoid vertical displacement where these loads affect the structure, see Figure 14.

The total weight of each model is the constraint for the product in the optimizations. The extra weight from passengers in the modeled cars has approximately the same mass as the wheels and the wheel suspensions. Therefore the weight of the models and the real cars are very close. Since only the obvious components which contribute with more than 10 kg have been modeled it is hard to achieve the exact weight.
4.2 Load cases

During optimization the structures are exposed to different load cases. The chosen load cases are listed below and will be simulated in the same way in all products.

Bending
Twisting
Roll-over
Rear crash
Front crash

These load cases are possible and relatively easy to simulate. More complex cases like side crash where large deformations are hard to calculate have been rejected. Also in front crash there are large deformations but this case will be studied in an early stage of the crash, and the comprehension of what happens at front crash is today relatively good.

Twisting and roll-over are simulated as two cases each in order to achieve symmetric results. Totally there are seven load cases.

4.2.1 Bending

The first case to study is bending. A gravity load of 1g is applied on the entire car and its masses, and the structure has fixed displacement boundary conditions according to the encirclements in Figure 15. In addition to vertical constraints in four nodes the structure needs to be fixed in all three directions in one of these nodes and in the y-direction, in another one. With these fixed displacements rigid body motion is prevented and the four nodes will during analysis stay in the same plane. There are as few constraints as possible but the ones mentioned above are necessary to avoid singular stiffness matrices. The gravity load affects the beam structure as well as all point and line masses and will remain also during the following load cases.

![Figure 15. Bending](image-url)
4.2.2 Twisting

In the twisting case, see Figure 16, the boundary conditions at one front wheel are released. This can for instance simulate the car driving into a hole in the road. Otherwise the boundary conditions are the same as in the bending case. To achieve a symmetric result two cases of twisting are studied, where one front wheel at a time is released.

![Figure 16. Twisting](image)

4.2.3 Roll-over

There are several ways to simulate the load case roll-over. The case used in this thesis can be described as a plate that is pushed against the roof at the A-pillar. The plate is rotated 25° around the x-axis and 5° around the y-axis. The requirement by law is that the plate is pushed by a force corresponding to 1.5 times the curb weight of the car times g. In Trinitas this is simulated by a point load according to Figure 17. To respond to the load in the x-direction both rear wheels are fixed in this direction. In the y-direction the load is transferred through the opposite long side by displacement conditions in this direction at front and rear wheels. Also at roll-over two cases are studied by symmetrical reasons.

![Figure 17. Roll-over](image)
4.2.4 Rear crash

This load case is a rear crash at low speed. To simulate that type of crash an acceleration field in the positive x-direction is applied on the beam structure and all point and line masses. At the same time displacements in this direction are fixed in the rear of the car, see Figure 18. The structure is fixed at the back in the x-direction and these constraints are of course the only ones in this direction. This way the optimization analysis will not be too restricted.

Figure 18. Rear crash

4.2.5 Front crash

Contrary to rear crash, two x-displacements in the front are fixed to simulate the front crash case, see Figure 19, and a high acceleration is applied in the negative x-direction. According to other calculations done at Volvo, see Figure 20, the acceleration does not vary remarkably in the car at a certain position in time during front crash. Hence, one value of the acceleration can be used through the entire car. Since parts of the load are transferred through the seat belts at front crash these are modeled as bars connected between the point masses of the passengers and the B- and C-pillars.

Figure 19. Front crash
At front crash, large deformations occur in the front area and the acceleration changes with time, see Figure 21. This diagram, which can be compared to Figure 20, shows two obvious steps. This curve could be simulated by two separate load cases, one case for each time step. Even so the front crash is only analyzed as one load case at the first time step and corresponding acceleration. This load case is very complex and severe plastic deformations will occur in the front, which are difficult to simulate. At the first time step there are no deformations but load paths through the car can still be observed.
5 Analyses of the products

5.1 Study of the created models

5.1.1 Engine suspension
To establish an accurate enough FE-model of the engine suspension, it is studied separately. Static analyses of different bar models are analyzed to find the correct load distribution. In these analyses both the structure and the stiffness of the bars are varied. The final structure according to Figure 8, distributes load according to data from Volvo.

5.1.2 Other point masses
The point masses representing doors could in addition to the two front bar connections also be connected to the B- and C-pillars according to Figure 22(a). In such a case these extra bars transfer loads which affect the B- and C-pillars as well as the door locks, where no loads are desirable. Instead, a fixed displacement condition, see Figure 22(b), is placed at the point in order to avoid the door from opening.

![Possible connection](a) Possible connection
![Chosen connection](b) Chosen connection

Figure 22. Connection of point masses

When point masses of seats and passengers are connected with bars to the floor, the load does not transfer fully vertical which is desirable. To avoid the problem the connecting bars can be fixed to nodes, which are placed near the floor and only connected to it through frictionless displacement constraints according to Figure23.

![Frictionless displacement constraints](Figure23. Frictionless displacement constraints)
Analyses with the above shown solution do not give a remarkable change in stress, hence it is possible to fix the bar structure directly to the floor. The same type of studies has been performed for other bar systems like doors, engine and fuel tank.

5.1.3 Design variables

When deciding the limits of the relative variables H/B, a greater freedom is allowed to avoid that too many beams reach the upper limits and hence give a result of less interest. The thickness is allowed to vary within 4-10% of the width. It is obvious that the relative thickness strives to attain its lowest value during optimization. By reducing this value the outer dimensions of the beams would increase and probably reach the upper limits of admissible values. For that reason a lower value of the relative thickness is not used and that is why some beams obtain quite a large value of the thickness after optimization. This is not optimal and large thicknesses such as over three millimeters are not manufactured. It is difficult to balance the design variables and it is important to be aware of this problem.

5.1.4 Front crash

When analyzing the products it is obvious that front crash is the most dominating case. The influences of the other load cases are considerably smaller. This result strengthens the decision to study only one case of front crash. Two time steps would decrease the importance of the remaining load cases even more.

5.2 Optimization analyses

It is important that the analyses give results which well are demonstrating how the structures are affected by the different load cases and how the different products affect each other. Several analyses are needed to obtain a good possibility to interpret and understand the results. The different optimization analyses which are going to be documented in this thesis are:

- V50 optimized over one load case at a time
- V50 optimized over all load cases
- S40 optimized over all load cases
- C70 optimized over all load cases
- All products optimized over all load cases

5.3 Selected sensitivity analyses

Additional analyses are performed on one product to investigate the sensitivity of the models by studying how certain factors affect the results. The selected sensitivity factors are:

- Normalization of the load cases
- Model with shell elements
- New load paths
- Simplified models without masses

It is interesting to study the influence of shell elements instead of certain line masses like roof, floor and windshield. In addition, investigations are made to find alternative ways for loads to transfer at front crash. Also normalization of load cases are studied, which means that the load cases are weighted differently depending on the initial compliance. More simplified models
are also to be analyzed in order to compare results between two different accuracies of modeling. These analyses described above are performed on V50 optimized over all load cases, except for the analysis with alternative load paths where only front crash is affecting the car body.

5.3.1 Normalization

In the beginning of an analysis every load case model obtains an initial value of the compliance. After each iteration, the objective function is divided by this value. When the analysis is completed a weighted value of the objective function is obtained. In this type of analysis the influence of all load cases is equal. The initial compliance for each load case \( l \) is

\[
C_0' = F_0^{l^T} u_0^l.
\]

The objective function for each iteration, is calculated as

\[
C_i = \sum_{l=1}^7 w^l F_i^{l^T} u_i^l,
\]

where the weighted factor \( w^l = \frac{1}{C_0'} \).

The weighting technique implies that after the first iteration the objective function has the value of seven, thus the sum of the number of load cases. This value will decrease until convergence is attained.

5.3.2 Model with shell elements

The modeled shells are 9-node Mindlin shells according to Figure 24. The thicknesses are adjusted so that the masses are the same as the line masses used in the previous models. Both rectangular and triangular elements are used, depending on the area where they are modeled. The windshield is modeled with material parameters equivalent to aluminum which is customary at Volvo. The thickness of the windshield is three millimeters. The roof and floor are modeled with material parameters equivalent to steel and all have thicknesses of one millimeter. To compensate for a loss of mass in the floor, a small line mass is included. A thicker metal sheet than one millimeter would contribute much to the stiffness which is why the line mass is necessary. The analysis is performed for all seven load cases.

Figure 24. Model with shells
5.3.3 New load paths

The new beams are constructed according to Figure 25 to increase the number of possibilities for loads to transfer. These yellow beams are allowed to change in size during optimization in the same way as the other beams in the floor. To keep the mass of the structure, some of the beams have a smaller cross section before analyzed. For the visibility of the new beams all the masses have been removed in the figure.

Figure 25. Volvo V50 with added beams

5.3.4 Simplified models

In addition to analyses of previous models with point and line masses, analyses without any masses are studied. The limits of the design variables for these models are the same as in earlier analyses. Since the point masses representing passengers are removed, point loads are applied in their places.

A simplification of the models implies that at front and rear crash only the beam structure is exposed to the applied acceleration fields.

These analyses are interesting in order to investigate how the geometries change in simplified models compared to the more detailed ones. The compliance is not to be compared in this case.
6 Results

The first analyses which are being presented are those where V50 has been exposed to each of the seven load cases, one at a time. Following analyses are; one product at a time exposed to all seven load cases and three products exposed to all seven load cases.

Analyses of one product affected by seven load cases are being compared with the analysis of three products affected by all load cases. It is desirable to investigate how the performance is changed when consideration to other products has to be taken. The performance is the compliance and the ability of the structure to handle the different load cases. Also, the inertia moments in some selected sections will be studied.

Following analyses are normalization, models with shell elements, models with added beams in the floor and more simple models with point loads and no point or line masses.

All products adapt to the upper mass constraint limit defined by the initial mass.

The colors which represent the stresses are scaled equally to make the stress levels comparable between all figures to come, see Figure 26.

<table>
<thead>
<tr>
<th>Color</th>
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</tr>
</thead>
<tbody>
<tr>
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<tr>
<td>Orange</td>
<td>1E+02</td>
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<td>Yellow</td>
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<tr>
<td>Gray</td>
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</table>

Figure 26. Used scale of stress (MPa)

6.1 Analyses of one product and one load case

These analyses give results where each received structure is optimal for one load case. No consideration to more than one load case has been taken. Twisting and roll-over have been analyzed as two load cases even though these are divided into two each. This is to receive a symmetric result.

Beams which are not contributing to the stiffness almost or totally disappear when optimized. This can be seen in Figure 27 which shows the load case bending, concerning the beams in the front and some of the beams in the roof and the rear area. Depending of loads from the roof and luggage door line mass, these beams are still visible. The stress is comparatively even throughout the structure.
Figure 27. Result bending

Figure 28, which presents the twisting load case, shows legibly that almost all beams except for those in the front area contribute to the stiffness. The roof bows are all rather large and important for the performance. As in the bending load case, the stress is comparatively even throughout the structure.

Figure 28. Result Twisting

Studying roll-over, the roof and the A-pillars are exposed to great loads. Their cross sections are relatively large after optimization. Also the B-pillars and the beam below the windshield tend to adapt large cross sections. It is clear that the significance of the floor and the beams in the front as well as in the rear is not as great as above mentioned beams.

Figure 29. Result roll-over
The most affected beams in rear crash are the beams in the rear area where the fixed displacements unique for this load case are. How the loads have been transferred through the car is easy to detect by studying Figure 30. The beams in the front are not important in rear crash. The roof bows are not affected by any load except for the line mass. The upper beams in the rear part have been reduced as in roll-over and bending.

The differences in stress levels are larger in the front crash load case. The largest compressive stress can be seen in the front and in the front part of the floor, see Figure 31. Beams of less importance for this load case are the roof bows, the cross members in the floor and the beams in the rear.

**6.2 Analyses of one product and seven load cases**

One product at a time has been optimized with respect to seven load cases. The following figures show the front crash case. The cross sections of the beams are obviously identical in all load cases but the shape and the stress levels are different.

The results of S40 and V50 in these types of analyses are very similar regarding geometry, stress distribution and compliance. It is obvious that Figure 32 and 33 very much resemble Figure 31, where V50 has been optimized with respect to only front crash. In the regions of the A-pillars, the frontal beams and the longitudinal beams in the floor, the cross sections are large and compressive stresses are observable. Influences of twisting and roll-over can be noticed when observing the beams in the roof, which are slightly larger than in the case with
only front crash. The lower beams in the rear have due to rear crash also a somewhat larger cross section than in earlier analysis. The remaining beams in the rear area do not need a larger cross section to achieve the required stiffness.

The width B of the cross sections has approached its highest admissible value in parts of the A-pillars and in the roof rails. As expected according to earlier discussion, nearly all beams have reached the lower limit of T/B. All beams are within the tolerable limits of H/B and $\theta$.

Since C70 is without roof, its result is a lot different than the results of S40 and V50, see Figure 34. Although after analysis, the geometry of the beams in the rear area and the inner longitudinal beams resemble the corresponding regions in the other products. The compliance has a larger value and the stress levels are higher throughout C70. The A-pillars absorb load in the roll-over case and the sills have a considerable larger cross section than in S40 and V50.
Parts of the A-pillars have approached the highest permitted values of the design variables B and H/B. Like in the analyses of V50 and S40, the thickness takes its lowest value in most of the beams.

**6.3 Analysis of three products and seven load cases**

Three products have been optimized with partly shared modules and with respect to all load cases. The figures of the result below are also in this analysis illustrating the front crash case. The optimization analysis must take all three products into consideration, which means that the common parts attain the same geometry.

The stresses in V50 and S40 are higher when additional products have been considered, than when separate analyses have been performed on these products over all load cases, see Figure 35 and 36. However, the compliance value and the geometry are close to equal in the same comparison.
Parts of the A-pillars and the roof rails have achieved the highest admissible values of B and H/B and most of the beams attain the lowest value of the thickness.

Also the result of C70, see Figure 37, is similar to the one from the analysis with only this product, but it clearly differs from V50 and S40. The compliance value is about the same as earlier and the highest stresses occur in the sills, in the other beams of the floor and in the front.

Some elements in the A-pillars have just like before increased and reached the upper limits of B and H/B. Also the thickness strives to reduce to the lowest value of T/B.
6.4 Normalization

The normalization analysis has been performed on V50 exposed to all seven load cases. The result in Figure 38 is compared to the corresponding analysis without weighted load cases, see Figure 32.

![Volvo V50, result normalization](image)

When front crash no longer is dominating, clear differences in stresses and geometry can be observed. The front beams as well as some beams at the front wheel suspension have a thinner cross section when normalized. However, several beams achieve a larger cross section, for instance the cross members in the floor, the lower beams in the rear area and the vertical beams at the rear wheel suspension. In the normalization case the stress levels are higher throughout the car body, which is clearly visualized in figure above.

Since the compliance in this case is slightly higher, a somewhat stiffer structure is obtained when the load cases are not normalized.

6.5 Model with shell elements

The analysis is performed on V50 with respect to seven load cases. The achieved result in Figure 39 resembles the result which is obtained from the analysis of V50 without shells. The main differences in geometry are the roof bows, which are to be thinner with shells. This is also the fact when observing the lower beams in the rear. Where stresses occur in each model the numerical values of the stresses are higher in the front crash model without shells. The compliance values are relatively equal.

![Volvo V50, result shell elements](image)
6.6 New load paths

V50 has been optimized with additional beams in the floor, in order to increase the possibilities of load transfer. The load case that is analyzed is front crash, see Figure 41. The optimization can be compared to the earlier performed analysis, where the existing model of V50 is analyzed with respect to only front crash, see figure 31. By such a comparison there are some evident differences that can be observed.

![Figure 41. Volvo V50, result added beams](image)

Most of the new load transfer possibilities are not utilized, except for a few beams. In the area below the rear seat the load selects another way, see beams 1 and 2 in Figure 41. This implies an unloading of beam number 3, which achieves a smaller cross section than in earlier analysis. Other differences in geometry can be noticed in the front, where the front beams are larger when the new beams have been added. The compliance in this analysis is slightly lower.

The stresses are distributed relatively similar in the car bodies both before and after the new beams were added. Nearly no beams have reached their upper limits of the design variable B. However, several beams have achieved the highest permitted value of H/B and like in all other analyses the thickness strives to attain its lowest value in most of the beams.

6.7 Simplified models

Analyses without added point and line masses have been performed on V50 with respect to all load cases, see Figure 40. The mass of the beam structure is equal as in previous analysis with V50 optimized over all load cases. Nevertheless, the mass distribution is different in the same comparison. In the simplified model the front beams as well as the beams in the floor are much thinner. Also in the rear part the cross sections decrease and some beams have entirely disappeared. The roof bows and the beam below the windshield have a larger cross section than before. The stress levels are low in the car and the beams along the roof which have adapted their upper limit of B as in earlier analyses. Nearly all beams in the structure attain the lowest permitted value of the thickness.
6.8 Rotation and principle area moment of inertia

In some chosen sections, see Figure 42, the principle area moments of inertia are studied in order to compare the results more quantitatively. The compared analyses are V50 exposed to seven load cases, all products exposed to seven load cases, V50 with shells, normalization analysis of V50 and simplified model of V50. The principle area moments of inertia are calculated in Trinitas and each element has a local coordinate system. Therefore one analysis, V50 exposed to seven load cases, will serve as a reference model and the rotation $R$ in the other analyses relative this model will be studied. The principle area moments of inertia and the changes in rotation, $\Delta R$, are shown in table 1-3. These tables also include normalized values of the principle area moments of inertia to make them more comparable. Figure 43-45 illustrate how the cross sections have rotated relatively to each other.
Both when V50 has been optimized separately and when consideration has been taken to the other products, similar values of the principle area moments of inertia are achieved, except for section 2.

The largest differences are observed in the analyses with normalized load cases and simplified models. In these analyses the principle area moments of inertia differ remarkably from the other ones and the changes in rotation are large.

### 6.8.1 Section 1

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<th>ΔR (°)</th>
<th>$I_1$ norm</th>
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Table 1. Normalized values of principle area moments of inertia and rotation, section 1
6.8.2 Section 2

Figure 44. Rotation in section 2

<table>
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<th>Section 2</th>
<th>$\Delta R$ ($^\circ$)</th>
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Table 2. Normalized values of principle area moments of inertia and rotation, section 2
6.8.3 Section 3

![Diagram of rotation in section 3](image)

Figure 45. Rotation in section 3

<table>
<thead>
<tr>
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Table 3. Normalized values of principle area moments of inertia and rotation, section 3
7 Discussion

The purpose of the thesis work was to develop a method which can be used early in the developing process of a family of new or existing cars. Several analyses have been carried out to observe how the structures are affected by different conditions and to investigate the trustworthiness of the results. To examine the possibilities of developing existing products topology optimization is a tool to detect new load paths through added beams.

It is relevant to include front crash in the optimizations even if this load case is analyzed statically. The time step which is studied is in an early stage of the crash when small plastic deformations have occurred. Yet it is still possible to investigate the load paths through the car body.

7.1 Analyses of single products

Analyses of one model exposed to one load case at a time have been carried out in order to investigate how each load case affects the structures. For instance, the front beams are important at front crash since a great deal of the load in this case is transferred through this area. When a model is exposed to all load cases it is obvious that front crash is dominating. The beams, whose cross sections strive to increase the most in the analysis of only front crash, also increase when all load cases are considered. On the other hand, the size of the beams in the roof, which are important at roll-over and twisting, increase in an analysis of all load cases compared to if only front crash is studied. Also the beams in the rear of the car are getting larger.

When comparing the three products which have been analyzed separately with respect to all load cases, quite large similarities can be noticed between V50 and S40. These products share two submodels and only differ from each other in the rear area, which is the area of least significance in optimization with all load cases. Therefore the results of V50 and S40 are similar after analysis.

In the two analyses of V50 and S40 over all load cases and in the analysis of V50 at front crash, the stresses are similar. In all models high compressive and tensile stresses occur in the same areas. The similar stress distribution is another indication of that front crash is dominating and that V50 and S40 have a related optimal geometry.

The result which is achieved when C70 is optimized with respect to all load cases is very different from the results of V50 and S40. There are great differences in geometry, since C70 for instance is out of B-pillar and roof. At roll-over all load is transferred through the A-pillars which probably are larger than if the roll-over protection system had been modeled and thus transferred parts of the load. High stresses, compression on the upper side and tension on the lower side, occur in the sills. Since there is no roof the sills transfer great loads at all load cases, which both explains the large cross section and the high stresses. The torsion sheets contribute to the stiffness at twisting.
When studying these analyses it is interesting to observe how most of the beams are required but used differently in the load cases. It is remarkable that nearly all beams in the models serve a purpose. However, the used models in this work are existing car models, which can explain why the beams are needed. While developing new models there is a larger freedom in order to create structures and greater differences will probably occur.

7.2 Analyses of all products and all load cases

When all products are optimized over all load cases the geometry and the values of compliance are similar to those where the products are optimized separately. It seems that the performance does not decrease remarkably when consideration has been taken to additional products, which is an unexpected and interesting result.

The stresses in the car bodies are higher when all products have been taken into consideration than in the analyses where they are studied individually. This is because several products are optimized at a time and the structures can not distribute their mass in the same optimal way as in previous analyses and the stress levels will raise.

C70 affects the other products in a way which makes the front beams in V50 and S40 thinner than in separate analyses. The different mass distribution occurs partly due to the sills in C70, which require more material than in the other products. Since the front area is shared S40 and V50 have to adapt to C70. The reduction in size of the front beams implies a higher stress level in this region. The remaining material is distributed and even if the sills in V50 and S40 are not shared with C70 they attain a somewhat larger cross section. This means that V50 and S40 obtain a lower stiffness in the front area, but instead the products can improve the result in other regions due to C70.

In the result chapter principle area moments of inertia are documented for some chosen sections. Both when V50 is optimized as an individual product and when consideration has been taken to the other products the moments of inertia are similar except for section two. This section is located in the sill and the difference is present as a consequence of that C70 strives to increase the size of this beam. The two other sections are located in the A- and B-pillar, which are not visibly affected by C70.

Additional analyses with normalization of the load cases, shell elements and models without added masses are studied to investigate the sensitivity of the models and how these factors affect the results.
7.3 Normalization

The purpose of this analysis is to investigate the importance of a relevant balance between the load cases. When analyzing the products it is obvious that front crash is the most dominating case. The influences of the other load cases are considerably smaller.

When the load cases are normalized great differences are apparent. This is because of the reduced influence of front crash. According to previous analyses front crash is the most dominating load case and the stiffness in this case is greatly reduced. The stiffness in twisting and roll-over are considerably improved but only a slightly improvement can be observed in the rear crash case. Even if the stiffness for several load cases is higher, the loss in the front crash case is so greatly that the sum of the compliance values for each load case is higher than in analyses without normalization. Thus, the use of different weight factors for the load cases can result in a total lower stiffness.

The principle area moments of inertia are quite different from the moments achieved earlier. This is not unexpected since roll-over and twisting has a greater influence of the result. The beams have achieved considerably altered orientation compared to V50 exposed to seven load cases. Since there are such obvious differences when load cases have been normalized, the results are of great interest. It is probably more accurate to utilize weight factors as a true engineering tool than letting one load case control the result as much as in the case of front crash. It is desirable to perform a normalization which makes sure that the product takes all load cases into consideration by selecting how much each load case are to be influencing the structure.

7.4 Model with shell elements

The results from an analysis of Volvo V50 with shell elements are similar to those received without shell elements. There is a difference in stress levels which are lower when shells are included. Shells transmit loads and reduce some of the stress levels in the beams, both in the floor and the roof. Compared to earlier analyses, the cross sections in these beams have decreased. The differences in stresses are confirmed by the reduced area moments of inertia, and imply a redistribution of mass in the structure. Because of an expected distinction in stiffness, the similarities between the two compared analyses are of interest. The reason for the small change in stiffness is the dominating front crash where the influence of the shells is relatively small.

7.5 New load paths

The value of the compliance is just slightly lower after this analysis than the analysis of the reference V50 but still shows that it is possible to find new load paths. The analysis has only been performed with respect to front crash since this is the load case which generates most loads through the beams in the floor. Since only one load case has been considered, the area moment of inertia has not been studied. It might be possible since front crash dominates the result which would probably be rather similar even if all load cases are considered.
7.6 Simplified models

The reason for the analysis with simplified models was to study the difference in results and see whether they were similar and accurate enough. The results show great difference and do not seem reasonable which implies that the added masses are necessary. It is obvious that this model do not handle front crash as well as the previous discussed model. The rotations of the beams in the studied sections are similar to those in the normalization analyses but the area moments of inertia are different. The moment in section two is almost the same as the reference model but section one and three are much lower. According to this, the cross sections have decreased. The loads which represent the passengers are not added in the model at the exactly correct positions and this is a source of error. An improved and more accurate result is achieved by modeling the components as point and line masses.
8 Conclusions

An applicable methodology for structural optimization in early phases of development of car product families has been established. By performing analyses and interpreting their results, several conclusions are drawn. One aspect is that point and line masses ought to be added to achieve a correct mass distribution in the models. Another important conclusion is that front crash without normalization is very dominant. If the load cases are weighted differently the results will show differences, which is interesting and desirable to investigate in further work. It is probably more reasonable to let the influences of the other load cases increase. When optimizing all products together there is only a small loss in stiffness.

The analysis of V50 with added beams in the floor shows that there is a potential of using the method to investigate new load paths through the car.

To improve the methodology it is wanted to establish a relevant balance between the load cases in order to receive results influenced not only by front crash. When relevant weight factors have been established there probably will be a more obvious difference in result between analyses of one compared to several products. Since V50 and S40 differ a lot from C70 it might be desirable to normalize the products as well, on how they are allowed to influence each other.

It is possible to extend the analyses by also optimizing the shape and/or change the objective function to mass with the compliance as a constraint.
References


[6] Torstenfelt B., Klarbring A. Structural Optimization of Modular Product Families with Application to Car Space Frame Structures. Accepted for publication in *Structural and Multidisciplinary Optimization*
