FEM-modelling the Hybrid III’s neck
a validated neck model of the 50th percentile adult Hybrid III dummy in MADYMO 5.1

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FEM-MODELLING THE HYBRID III's NECK

A validated neck model of the 50th percentile adult Hybrid III dummy in MADYMO 5.1.

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Isaac Newton

Applying a force $F$ on a body with mass $m$ results in an acceleration $a$ in the same direction as the force: $F = ma$.

Bob Reuser

When something's stuck, force it. If it breaks, it needed replacement anyway.
Summary

This report describes the finite element modelling of the 50th percentile Hybrid III’s neck in the multi-body/finite element code MADYMO 5.1. The neck of the Hybrid III dummy consists of five aluminium discs, which are interconnected by four rubber discs. The aluminium endplates connect the neck to the head and thorax of the Hybrid III dummy. At the anterior side of the rubber discs, fissures have been applied in order to decrease the extension stiffness of the neck. Inside the Hybrid III’s neck, a pretensioned steel cable increases the axial stiffness of the neck.

Due to the incompressible material behaviour of the rubber discs, a linear material model did not suffice to model the material of the rubber discs. Instead, a rubber material model had to be used. However, MADYMO 5.0 did not incorporate such a material model. Therefore, a rubber material model had to be developed and embodied in MADYMO 5.1.

In this report, several rubber material models were surveyed. For reasons of computation times and incompressibility behaviour of the material model, the compressible Mooney-Rivlin rubber material model was chosen to model the material of the rubber discs of the Hybrid III’s neck. The aluminium discs were modelled as rigid bodies, as the stiffness of the aluminium discs is much higher than the stiffness of the rubber discs. The fissures in the rubber discs were included in the FEM-mesh. The steel cable inside the Hybrid III’s neck was modelled as a belt system, a special feature of MADYMO 5.1.

In order to validate the Hybrid III’s neck model, computed responses of the neck were compared with measurements on experiments for the static validation and with a validated two-pivot model for the dynamic validation. In the dynamic validation, a Hybrid III dummy head was connected to the neck. From the comparisons, it was concluded that static flexion bending and extension bending of the neck could be predicted very well. The dynamic simulations predicted the linear and angular accelerations of the dummy’s head centre of gravity with satisfactory accuracy. However, the computed displacements of the head centre of gravity showed small differences with the two-pivot model. This could be caused by the lack of damping in the material model. Therefore, it is recommended to include a rate dependent component to the compressible Mooney-Rivlin material model in order to describe high-speed deformations more accurately.
Samenvatting

In dit rapport wordt beschreven hoe de nek van de vijftig percentiel Hybrid III dummy is gemodelleerd met het multi-body/eindige elementen programma MADYMO 5.1. De nek van de Hybrid III dummy is opgebouwd uit vijf aluminium schijven die afgewisseld worden door vier rubber schijven. De eerste en laatste aluminium schijf van de nek wordt gebruikt om de nek aan het dummyhoofd en aan de dummy thorax te verbinden. Aan de voorzijde van de rubber schijven zijn inkepingen aangebracht om de extensie stijfheid van de nek te verlagen. Binnenin de nek van de dummy bevindt zich een voorgespannen stalen kabel, die de axiale stijfheid van de nek verhoogd.

Als gevolg van het onsamendrukbare materiaalgedrag van de rubber schijven, voldeed een lineair materiaal model niet om het materiaal van de rubber schijven te modelleren. In plaats daarvan moest een rubber materiaalmodel worden gebruikt. In MADYMO 5.0 was echter geen rubber materiaalmodel beschikbaar. Daarom moest een rubber materiaalmodel worden ontwikkeld en worden toegevoegd aan MADYMO 5.1.

Allereerst wordt in dit rapport een aantal rubber materiaalmodellen gepresenteerd. Aan de hand van voorwaarden aan rekentijd en onsamendrukbaar gedrag van het rubber, werd het samendrukbare Mooney-Rivlin materiaalmodel gekozen om het materiaal van de rubber schijven van de Hybrid III dummy te beschrijven. Omdat de stijfheid van de aluminium schijven veel hoger is dan de stijfheid van de rubber schijven, zijn de aluminium schijven gecodeerd als starre lichamen. De inkepingen in de rubber schijven zijn opgenomen in de FEM-mesh. De stalen kabel in de nek van de Hybrid III dummy is gecomputeerd als een gordelsysteem, een optie van MADYMO 5.1.

Het nekmodel van de Hybrid III-nek wordt gevalideerd door berekende responsies van het nekmodel te vergelijken met metingen aan experimenten voor de statische simulaties en met een gevalideerde two-pivot model voor de dynamische simulaties. Bij de dynamische validatie was een Hybrid III dummyhoofd verbonden aan de nek. Uit de vergelijkingen kan worden geconcludeerd dat de statische buigingen in flexie en extensie goed konden worden voorspeld. Ook de lineaire- en hoekversnellingen van het dummyhoofd bij de dynamische validatie werden naar tevredenheid voorspeld. Helaas traden er wel kleine verschillen op in de verplaatsingen van het zwaartepunt van het dummyhoofd wanneer het FEM-model werd vergeleken met het two-pivot model. Dit wordt waarschijnlijk veroorzaakt door de afwezigheid van een snelheidsafhankelijk gedrag in het materiaalmodel. Het wordt daarom dan ook aanbevolen om het materiaalmodel uit te breiden met een snelheidsafhankelijke component om ook hoge snelheidsdeformaties goed te kunnen beschrijven.
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Chapter 1: Preliminaries

1 Preliminaries

The ever increasing amount of traffic causes more and more casualties. Not only cyclists and pedestrians are victims of accidents, but occupants of motor vehicles can also be hurt as a result of an impact with the interior of the vehicle like the dashboard or the steering wheel. A vulnerable part of the human body in car crashes is the neck and the head. Damaging the neck can be caused by an impact to the head, but also subjecting the head to an acceleration can strain the spinal cord, dislocate the vertebrae of the spinal column or cause head injuries.

The Snell Memorial Foundation was founded in 1957 after the death of the sports car racer William Snell at a race meet in 1956. The goals of the foundation were set to study and develop better head protection for individuals participating in racing activities or involved in vehicular transportation. Nowadays, the foundation has a leading role in the U.S. and throughout the world, setting authoritative safety standards for helmets. Educating the public about head injury and fatality prevention has also become one of the occupations of the foundation.

One way of testing new helmet designs is to apply contact loads and inertial loads to the new helmet when the helmet is put on an artificial neck and head. In order to test the helmet properly, the artificial neck and head must have the biofidelity of a human neck and head. The Snell Memorial Foundation, therefore, sponsors a project at the biomechanics section of the TNO Road-Vehicles Research Institute to develop an omni-directional biofidelic dummy neck.

The project is divided into three stages. In the first stage, performance requirements for the dummy neck are set. These requirements in flexion, oblique and lateral direction are the result of acceleration sled tests with human volunteers and cadavers. The thrust vector of the acceleration field in a flexion, oblique and lateral test, respectively, points from the back to the front of the head, from the right back to the left front of the head and from the right to the left of the head.

The second stage of the project deals with the analysis and optimizing of various mathematical models of the human neck. On the one hand, multi-body models are used to predict the influence of, for example, the position of the centre of gravity of the head on the responses of the neck. On the other hand, finite element models are developed in order to obtain the dimensions of a new omni-directional biofidelic neck. The responses of the new omni-directional biofidelic neck has to meet the set performance requirements. In the third and last stage of the Snell project, a prototype of the new neck will be built and tested.

Up till now, the first stage is almost finished. The head-neck behaviour was analyzed (Bruijs et al., 1992), which resulted in preliminary omni-directional performance corridors. Multi-pivot (Schaap, 1992) and two-pivot (Siemerink, 1993) models were developed in MADYMO 5.0 to determine the sensitivity of the model responses to different design
parameters. MADYMO, which stands for MAthematical DYnamical MOdels, is an integrated multi-body/finite element code, developed at TNO Road-Vehicles Research Institute. It is especially developed to simulate the behaviour of humans in traffic accidents. It can also be used, of course, to solve general multi-body/finite element problems. The two-pivot models model the human neck and head as two rigid links and two pivots, whereas the multi-pivot models represent the neck and head as five links.

This report describes the finite element modelling of the neck of the existing 50th percentile frontal crash test Hybrid III dummy. The reason to model an existing neck of which the static and dynamic properties are known, is to validate the new rubber material model in MADYMO 5.1 and to obtain experience in the field of finite element modelling with MADYMO 5.1. Also, the model of the Hybrid III’s neck provides for a basis, from which the new omni-directional biofidelic neck will be designed and is an aid in the understanding of designing new dummy necks.

The design of the neck model of the Hybrid III’s neck begins with a review of a number of rubber material models. For reasons of computation times and incompressible behaviour of the material model, it appeared that the compressible Mooney-Rivlin model was the best choice for the material model. The compressible Mooney-Rivlin model is described and derived in chapter 3. The basis of the material subroutine is presented as well as some general problems such as the locking phenomenon. The finite element modelling of the Hybrid III’s neck is described in chapter 4. Also the contact interaction of the aluminium discs with each other and the modelling of the steel cable inside the Hybrid III’s neck is dealt with. The complete neck model is validated by comparing computed responses with static measurements and dynamic two-pivot simulations. The static validation can be found in chapter 5, whereas the dynamic validation is presented in chapter 6. The report ends with the conclusions and recommendations for the finite element model.

Finally, I would like to thank the Board of Examiners, without whom this report would not have its present form. Especially, I express my gratitude to Jan Thunnissen for his constructive criticism after his endless reviews of my report, to Professor Jac Wismans who gave me the opportunity to perform this research and to the MADYMO-group, who helped me solve many MADYMO problems.

Bob Reuser.
2 Material Models for Rubber and Rubbery Materials

2.1 Introduction

The Hybrid III’s neck is constructed out of two different materials. The rigid part consists of aluminium discs, which are interconnected by flexible rubber discs, see appendix A. In the literature (Slaats, 1993), it can be found that the rubber discs in the neck of the dummy have a large effect on the accelerations and displacements of the dummy’s head. As the Hybrid III’s neck will be modelled in FEM, a hyperelastic rubber material model for the rubber discs has to be chosen. The rubber of the Hybrid III’s neck possesses three important properties, i.e. complete elasticity, isotropic behaviour and near incompressibility. These properties should be reflected in the constitutive equations of the material model. In MADYMO 5.0, however, no material model which could describe the rubber properties was available. Therefore, it was decided to design a rubber material model, which could be embodied in the new MADYMO 5.1.

This chapter contains a collection of material models, which all have their advantages and disadvantages. From the original FEM-model of the Hybrid III’s neck in PAMCRASH (Slaats, 1990) and LS/DYNA3D (Slaats, 1993), three possible candidate material models are reviewed in the sections 2.2 to 2.4. In the sections 2.5 to 2.8, four more rubber material models are described (Treloar, 1975).

In the sections 2.4 to 2.8, the material models are based on a strain energy density function. This function can either be written in terms of the right Cauchy-Green strain tensor invariants \( J_1, J_2 \) and \( J_3 \) or in the principal stretches \( \lambda_1, \lambda_2 \) and \( \lambda_3 \). The principal stretches are defined as the ratio of current to reference lengths of a line element in the direction of the principal axes of strain in the reference configuration. The principal stretches are equivalent to the right Cauchy-Green strain tensors, as they are the squared eigenvalues of the right Cauchy-Green strain tensor.

The material models presented in the sections 2.2 to 2.8, are merely an overview of a number of material models found in the literature. The models are not compared in simulations, but a material model for the Hybrid III’s neck is chosen on the basis of model properties and easy implementation in MADYMO 5.1. In the last section, the material models are reviewed critically and the model is chosen which will be used in the rubber material subroutine of MADYMO 5.1.

2.2 Linear Elastic Material Model

The linear elastic rubber material model can be described (Brekelmans, 1986) by the constitutive equation (2.1).

\[
P_2 = 2\mu E + \lambda \text{tr}(E) I \quad (2.1)
\]
where $P_2$ is the second Piola-Kirchhoff stress tensor, $E$ is the Green-Lagrange strain tensor, $I$ is the identity tensor and $\lambda$ and $\mu$ are the so-called Lamé elastic constants. The Lamé constants depend on the Young's modulus $E^*$ and Poisson's ratio $\nu$ in the following way:

$$
\lambda = \frac{\nu E^*}{(1+\nu)(1-2\nu)} \quad (2.2)
$$

and

$$
\mu = \frac{E^*}{2(1+\nu)} \quad (2.3)
$$

According to Slaats (1993), the rubber of the Hybrid III can best be described with the material parameters set to $E^* = 6.35 \times 10^6$ [N/m$^2$] and $\nu = 0.499$ [-]. The material parameters were obtained from simple simulations of uniaxial stress tests. The major disadvantage of this linear material model is the inability to describe the non-linear behaviour between the stresses and the strains. This model can, therefore, only be used to predict small deformations or deformations in a limited range (linearization of a small range of the strain-stress curve) of the Hybrid III's neck. Since the Hybrid III's neck will have large deformations in both static and dynamic tests, makes the linear material model unsuitable to be used in the simulations.

2.3 Blatz-Ko Hyperelastic Rubber Model

According to Whirley et al. (1991), the Blatz-Ko hyperelastic rubber model is useful for "moderately large strains". Constitutive equation (2.4) relates the second Piola-Kirchhoff stress tensor $P_2$ to the right Cauchy-Green strain tensor $C$ and to the relative volume $J = \frac{V}{V_0}$ as the ratio current volume/reference volume.

$$
P_2 = G(J^{-1}C - J^{-1}F^T F) \quad (2.4)
$$

The right Cauchy-Green strain tensor $C$ equals $C = F^T F$, with $F^T$ the conjugated deformation tensor $F$. The constant $G$ appeared to be equal to the shear modulus of the rubber. The shear modulus of the Hybrid III neck's rubber was estimated from a rubber hardness table (Slaats, 1993). It was found to be $G = 1.90 \times 10^6$ [N/m$^2$].

2.4 Compressible Mooney-Rivlin Rubber Model

For the compressible Mooney-Rivlin rubber model (Whirley et al., 1991), the relationship between the strain energy density function $W$ and the three invariants of the right Cauchy-Green strain tensor $C$ is given by equation (2.5). The strain energy density function denotes the stored elastic energy per reference volume. The dimension of the strain energy
density function is \([J/m^3]\).

\[
W = A^*(J_1^{-3}) + B^*(J_2^{-3}) + C^*(J_3^{-2} - 1) + D^*(J_3 - 1)^2
\]  \(2.5\)

\(J_1, J_2\) and \(J_3\) are the first, second and third invariant of the right Cauchy-Green strain tensor \(C\) and can be written as:

\[
J_1 = \text{tr}(C)
\]

\[
J_2 = \frac{1}{2}(\text{tr}^2(C) - \text{tr}(C^2))
\]  \(2.6\)

\[
J_3 = \det(C) = \left(\frac{V}{V_0}\right)^3
\]

The strain invariants written in the principal stretches are:

\[
J_1 = \lambda_1^2 + \lambda_2^2 + \lambda_3^2
\]

\[
J_2 = \lambda_1^2\lambda_2^2 + \lambda_2^2\lambda_3^2 + \lambda_3^2\lambda_1^2
\]  \(2.7\)

\[
J_3 = \lambda_1^2\lambda_2^2\lambda_3^2
\]

The three strain invariants of the right Cauchy-Green strain tensor are used for their independence of the orientation of the right Cauchy-Green strain tensor basis. The material parameters \(C^*\) and \(D^*\) in equation (2.5) can be seen as penalty factors for the volumetric deformation \(J_3\). A high value for \(C^*\) and \(D^*\) causes a high resistance against volumetric deformation. As a result of hydrostatic tests (Bogert, 1991), the compressibility modulus of rubber was found to be \(\kappa = 5.0 \times 10^8 [Pa]\). The compressibility of the Mooney-Rivlin material model is not determined by one material parameter but is, unfortunately, determined by all four material parameters \(A^*, B^*, C^*\) and \(D^*\). Slaats (1993) recommended for the input parameters as a result of FEM-computations with the Hybrid III's neck: \(A^* = 9.3 \times 10^5 [N/m^2]\), \(B^* = 0.0 [N/m^2]\) and \(v = 0.499 [-]\). The material parameters \(C^*\) and \(D^*\) depend on \(A^*, B^*\) and \(v\), see appendix B.

According to Boogaard (1988), the second Piola-Kirchhoff stresses can be determined from the strain energy density function by:

\[
P_{\text{2ij}} = \frac{\partial W}{\partial E_{\text{ij}}}
\]  \(2.8\)

The subscript \(ij\) denotes the direction of the stresses.
With the relationship between the Green-Lagrange strain and the Cauchy-Green strain:

\[ C = 2E + I \]  

(2.9)

it follows:

\[ P_{2ij} = 2 \frac{\partial W}{\partial C_{ij}} \]  

(2.10)

In a strain state with no shear, the diagonal Cauchy-Green strain components can be written in terms of the principal stretches \( \lambda_1, \lambda_2 \) and \( \lambda_3 \):

\[ C_{xx} = \lambda_1^2; \quad C_{yy} = \lambda_2^2; \quad C_{zz} = \lambda_3^2 \]  

(2.11)

Substituting equation (2.11) into equation (2.10) yields the principal second Piola-Kirchhoff stresses as function of the principal stretches:

\[ P_{2i} = \frac{1}{\lambda_i} \frac{\partial W}{\partial \lambda_i} \]  

(2.12)

with the subscript \( i \) as the direction of the principal second Piola-Kirchhoff stresses.

2.5 Mooney Model

The Mooney model (Mooney, 1940) is, according to the literature (Treloar, 1975), the earliest significant theory of large elastic deformations. Mooney’s theory is based on the assumptions that the rubber is incompressible, that the rubber is isotropic in the unstrained state and that Hooke’s law is obeyed in shear. On the basis of these assumptions, Mooney derived the following strain energy density function:

\[ W = C_1(J_1 - 3) + C_2(J_2 - 3) \]  

(2.13)

with \( J_1 \) and \( J_2 \) the first and second invariant of the right Cauchy-Green strain tensor \( C \) and \( C_1 \) and \( C_2 \) the material parameters.

Equation (2.13) can be seen as the compressible Mooney-Rivlin rubber model with \( C^* = D^* = 0 \). As a result of the incompressibility assumption, the third invariant of the right Cauchy-Green strain tensor can be written as \( J_3 = 1 \). Substituting this relation into equation (2.13) yields:

\[ W = C_1(\lambda_1^2 + \lambda_2^2 + \lambda_3^2 - 3) + C_2(1/\lambda_1^2 + 1/\lambda_2^2 + 1/\lambda_3^2 - 3) \]  

(2.14)
2.6 Rivlin Model

Rivlin (1948) formulated a material model on the basis of the assumptions that the material is incompressible and that the material is isotropic in the unstrained state. The condition for isotropy requires that the strain energy density function \( W \) is symmetrical with respect to the three principal stretches \( \lambda_1, \lambda_2 \) and \( \lambda_3 \). The three simplest functions which meet these requirements have been presented in equation (2.7). These expressions are called the strain invariants and are equivalent with the invariants as function of the right Cauchy-Green strain tensor, see equation (2.6). More complex functions, which meet the symmetry requirements, can always be written as a function of the already stated invariants. The incompressibility of the material is taken into account by setting \( J \), \( \lambda_3 \) to one. According to the Rivlin model, the strain energy density function for an incompressible isotropic elastic material may be expressed as the sum of a series of terms:

\[
W = \sum_{i=0}^{i_{\text{max}}} \sum_{j=0}^{j_{\text{max}}} C_{ij} (J_1 - 3)^i (J_2 - 3)^j \quad C_{00} = 0
\]  

(2.15)

It can be noted that setting \( i_{\text{max}} = 1, j_{\text{max}} = 1 \) and \( C_{11} = 0 \), the Rivlin formulation transforms into the Mooney material model.

2.7 Ogden Model

The restriction of even powers of the principal stretches in the strain energy density function was dispensed with in the Ogden model (Ogden, 1984). The strain energy density function for an incompressible rubber could, according to Ogden, be written in the form of the series:

\[
W = \sum_{j=1}^{n} \mu_j (\lambda_1^\alpha_j + \lambda_2^\alpha_j + \lambda_3^\beta_j - 3)
\]  

(2.16)

in which \( \alpha_j \) and \( \mu_j \) are material parameters. The Ogden model does not take the incompressibility condition into account and is, therefore, less suitable for the rubber of the Hybrid III’s neck. The Mooney material model is obtained for \( \alpha_1 = 2 \) and \( \alpha_2 = -2 \).

2.8 Valanis-Landel Model

In the literature (Valanis et al., 1967), it was suggested that the strain energy density function should consist of three separate functions of the three principal stretches:

\[
W = w(\lambda_1) + w(\lambda_2) + w(\lambda_3)
\]  

(2.17)

From isotropy considerations in the unstrained state, the separate functions \( w(\lambda_i) \) are identical.
Chapter 2: Material Models for Rubber and Rubbery Materials

The Mooney formulation is a variant of the Valanis-Landel hypothesis. Valanis and Landel suggested as a result from shear experiments:

\[ w' (\lambda) = 2\mu \ln(\lambda) + \frac{c}{\lambda} \]  

(2.18)

As can be found in section 2.7, Ogden’s formulation is consistent with Valanis-Landel’s hypothesis. Comparing equation (2.16) to equation (2.17) yields:

\[ w' (\lambda) = \sum_{j=1}^{n} \mu_j \lambda_j^{\alpha_j-1} \]  

(2.19)

2.9 Discussion

In the sections 2.2 to 2.8, several material models have been presented. The different models can traditionally be divided into two categories. In the first category, the constitutive models depend on the principal stretches \( \lambda_1, \lambda_2, \) and \( \lambda_3 \). The material models that belong to this category are the Mooney model, the Rivlin model, the Ogden model and the Valanis-Landel model. In the second group of models, the constitutive relationship is dependent on the strain (Green-Lagrange E or right Cauchy-Green C) directly, as can be seen with the linear elastic, Blatz-Ko material model and the compressible Mooney-Rivlin rubber model. The compressible Mooney-Rivlin material model actually depends on the invariants of the right Cauchy-Green strain tensor. It must be mentioned, however, that the principal stretches \( \lambda_1, \lambda_2, \) and \( \lambda_3 \) are equivalent to the right Cauchy-Green strain tensor. The eigenvalues of the right Cauchy-Green strain tensor are the squared principal stretches. To determine the stresses of first category of materials, an arbitrary right Cauchy-Green strain state of a deformed piece of material has to be transformed first into the principal stretches. The calculated principal second Piola-Kirchhoff stresses, thereupon, have to be transformed back to the stresses belonging to the basis of the original strain tensor state. In general, this costs too much computational time and the models which depend on the principal stretches \( \lambda_1, \lambda_2, \) and \( \lambda_3 \) are, therefore, less suitable to be implemented in MADYMO 5.1.

Although rubber seems a rather flexible material, it has a high resistance to volumetric deformations. This property must also be reflected in the material model. The third invariant of the right Cauchy-Green strain equals the squared relative volume. This provides for an excellent way to model the incompressibility (or constant volume) property of rubber. The compressible Mooney-Rivlin model is, next to the first and second invariant, explicit dependent on the third invariant. For its incompressibility feature and its non-linearity, the compressible Mooney-Rivlin material model has been chosen to model the rubber in the Hybrid III’s neck.

The material parameters of the compressible Mooney-Rivlin material model (Slaats, 1993)
are based on uniaxial extension and compression tests of the Hybrid III neck's rubber (Tolner, 1991). The material parameters of Slaats were tuned on uniaxial tests, while the deformation of the rubber discs is three-dimensional. The parameters obtained in this way can, therefore, be seen as a first approximation for the rubber material. The final material parameters can be obtained by comparing simulated full-neck responses with actual experiments with the head-neck complex.
3 The Hybrid III Neck’s Rubber Model

3.1 Introduction

Because of its dependence on the right Cauchy-Green strain invariants and its explicit resistance to volumetric deformation, the compressible Mooney-Rivlin material model was chosen to model the rubber discs in the Hybrid III’s neck. This chapter deals with the underlying theory of the compressible Mooney-Rivlin material model. The theory of the strain energy density function is presented and the constitutive equations of the compressible Mooney-Rivlin model are derived in section 3.2. In section 3.3, the way in which the material model has been embodied in MADYMO 5.1 (MADYMO, 1994) is described. The locking phenomenon can be found in section 3.4. Locking is the impossibility of certain types of finite elements to deform according to reality under certain loading conditions when the material is modelled as completely or nearly incompressible. The chapter concludes with a discussion.

3.2 Compressible Mooney-Rivlin Model (Extended)

In general, the mechanical behaviour of a material can be modelled by means of a strain energy density function. This function relates the accumulated elastic energy of a reference volume of material to the deformation of a reference volume of material. The stored energy is a result of the work done on the reference volume. To obtain the second Piola-Kirchhoff stresses $P_{2,ij}$ from the strain energy density function, this function has to be differentiated with respect to the Green-Lagrange strain tensor (Boogaard, 1988). In the compressible Mooney-Rivlin material model, however, only the right Cauchy-Green strain tensor is used. With the chain rules follows:

$$P_{2,ij} = \frac{\partial W}{\partial C_{kl}} \frac{\partial C_{kl}}{\partial E_{ij}}$$  \hspace{1cm} (3.1)

The subscripts $ij$ and $kl$ denote the directions of the stress. Because

$$C = 2E + I$$  \hspace{1cm} (3.2)

it follows:

$$P_{2,ij} = 2 \frac{\partial W}{\partial C_{ij}}$$  \hspace{1cm} (3.3)
Whirley et al. (1991) formulated the compressible Mooney-Rivlin material model as:

\[ W = A'(J_1 - 3) + B'(J_2 - 3) + C'(J_3^2 - 1) + D'(J_3 - 1)^2 \]  

(3.4)

\( J_1, J_2 \) and \( J_3 \) are the invariants of the right Cauchy-Green strain tensor \( C \):

\[ J_1 = \text{tr}(C) \]

\[ J_2 = \frac{1}{2}(\text{tr}^2(C) - \text{tr}(C^2)) \]

\[ J_3 = \det(C) = \left( \frac{V}{V_0} \right)^2 \]

or written in the right Cauchy-Green's components:

\[ J_1 = C_{xx} + C_{yy} + C_{zz} \]

\[ J_2 = C_{xx}C_{yy} + C_{yy}C_{zz} + C_{zz}C_{xx} - C_{xy}C_{yx} - C_{yz}C_{zy} - C_{zx}C_{xz} \]

\[ J_3 = C_{xx}C_{yy}C_{zz} + C_{xy}C_{yz}C_{zx} + C_{yx}C_{zy}C_{xz} - C_{xx}C_{yz}C_{zx} - C_{xx}C_{zx}C_{yz} - C_{xx}C_{yx}C_{zy} \]

(3.6)

Differentiating the strain energy density function with respect to the components of the right Cauchy-Green strain tensor according to equation (3.3), gives the constitutive relationship between the second Piola-Kirchhoff stresses and the right Cauchy-Green strains:

\[ P_{2,xx} = 2(A + B'((C_{yy} + C_{zz}) - 2J_3C_{yy} - C_{yz}^2) + 2D'(J_3 - 1)(C_{yy}C_{zz} - C_{yz}^2)) \]

\[ P_{2,yy} = 2(A + B'((C_{xx} + C_{zz}) - 2J_3C_{xx} - C_{xx}^2) + 2D'(J_3 - 1)(C_{xx}C_{zz} - C_{xx}^2)) \]

\[ P_{2,zz} = 2(A + B'((C_{xx} + C_{yy}) - 2J_3C_{yy} - C_{yy}^2) + 2D'(J_3 - 1)(C_{xx}C_{yy} - C_{yy}^2)) \]

\[ P_{2,xy} = 2(-B'\sqrt{C_{xy}^2 - 2J_3C_{xy}C_{y} - C_{xx}C_{yy}^2} + 2D'(J_3 - 1)(C_{xy}C_{xy} - C_{xx}C_{yy})) \]

\[ P_{2,yz} = 2(-B'\sqrt{C_{yz}^2 - 2J_3C_{yz}C_{y} - C_{yy}C_{xx}^2} + 2D'(J_3 - 1)(C_{xy}C_{xy} - C_{xx}C_{yy})) \]

\[ P_{2,xz} = 2(-B'\sqrt{C_{xz}^2 - 2J_3C_{xz}C_{x} - C_{xx}C_{zz}^2} + 2D'(J_3 - 1)(C_{xz}C_{xz} - C_{xx}C_{yy})) \]

(3.7)

The material behaviour is determined by three parameters \( A^* \), \( B^* \) and \( v \). The material parameters \( C^* \) and \( D^* \) can be calculated from the former parameters according to:

\[ C^* = \frac{1}{2}A^* + B^* \]

(3.8)
Chapter 3: The Hybrid III Neck's Rubber Model

\[ D^* = \frac{(5v-2)}{2(1-2v)}A^* + \frac{(11v-5)}{2(1-2v)}B^* \]  

(3.9)

Equation (3.8) is a necessary condition to ensure that no stresses occur when the material is unstrained. When equation (3.9) is applied to the Mooney-Rivlin material model, the model is valid for an uniaxial stress test for small deformations. The equations (3.8) and (3.9) are derived in appendix B. The factors \( C^* \) and \( D^* \) can be seen as penalty factors for \( J_3 \), a measure for the volumetric deformation. A high resistance against volumetric deformation is obtained by setting Poisson’s ratio to nearly 0.5 [-].

The compressible Mooney-Rivlin material model has a distinct different stress-strain characteristic with respect to a linear material. In figure 3.1, the characteristic of an uniaxial stress test with free contraction of both an isotropic linear material and a compressible Mooney-Rivlin material is depicted. It can be seen that for large compression and extension strains, the stresses of the Mooney-Rivlin model exceed the stresses of the linear model. This is due to the resistance against volumetric deformations. When an uniaxial stress test without free contraction had been performed, the stiffness of the Mooney-Rivlin material would be even more higher.

3.3 The MADYMO 5.1 Subroutine

As can be seen in equation (3.7), the compressible Mooney-Rivlin strain energy density function relates the second Piola-Kirchhoff stresses to the right Cauchy-Green strains. However, MADYMO 5.1 uses natural logarithmic strains and Cauchy stresses in finite element calculations. Because of the differences between MADYMO 5.1 and the Mooney-Rivlin material model concerning the input and output format, adjustments have to be made for the input of strains and the output of stresses. From the compressible Mooney-Rivlin model, second Piola-Kirchhoff stresses are calculated. The second Piola-Kirchhoff stresses can be transformed to Cauchy stresses with the help of the deformation tensor \( F^{0-n} \):

\[ \sigma_{ij} = \frac{F^{0-n}P_{2ij}(F^{0-n})^T}{det(F^{0-n})} \]  

(3.10)

with \( \sigma_{ij} \) the Cauchy stresses, \( P_{2ij} \) the second Piola-Kirchhoff stresses, and \( det(F^{0-n}) \) the
Chapter 3: The Hybrid III Neck’s Rubber Model

determinant of the deformation tensor. The right Cauchy-Green strain also is not standard available in MADYMO 5.1. Again, the deformation tensor is used to determine the right Cauchy-Green strain:

$$C^{0-H} = (F^{0-H})^TF^{0-H}$$  \hspace{1cm} (3.11)

with the deformation tensor defined as:

$$F^{0-H} = (V^0(x^t))^T$$  \hspace{1cm} (3.12)

In equation (3.12), $x^t$ represents the current nodal coordinates at time $t$ and $V^0$ is the gradient operator at time zero. $x^t$ and $V^0$ can be written more elaborately as:

$$V^0 = \begin{bmatrix} \frac{\partial}{\partial x_1^0} & \frac{\partial}{\partial x_2^0} & \frac{\partial}{\partial x_3^0} \end{bmatrix}^T ; \hspace{0.5cm} (x^t)^T = [x_1^t \hspace{0.2cm} x_2^t \hspace{0.2cm} x_3^t]$$  \hspace{1cm} (3.13)

with $x_1^t$, $x_2^t$ and $x_3^t$ as the three components of the current position of a material point in the global coordinate system and $x_1^0$, $x_2^0$ and $x_3^0$ as the three components of the position of a material point at time zero in the global coordinate system, see figure 3.2. Combining equations (3.12) and (3.13) gives the deformation tensor, see equation (3.14).

![Figure 3.2. Reference and current coordinates of a material point.](image)

The basis of the finite element method lies in the discretization of the continuum. With the help of interpolation functions and nodal positions, it is possible to obtain the positions of material points throughout the element. The compressible Mooney-Rivlin material model will be applied to an eight-node linear volume-element with reduced integration in order to prevent locking and to reduce the computational time.
The interpolation functions multiplied by the current or the reference nodal position give the current or reference position of any material point within the element (see figure 3.3) in question:

\[ x_i^{0l}(\eta, \rho, \xi) = \sum_{k=1}^{8} N_k(\eta, \rho, \xi)(x_i^{0l})^k \]  

(3.15)

with \( N_k \) the interpolation functions, \( x_i^{0l} \) the current or reference position of a material point within the element and \( (x_i^{0l})^k \) the current or reference position of node \( k \).

The components of the deformation tensor can be obtained by differentiating equation (3.15) with respect to the coordinates in the reference configuration:

\[ \frac{\partial x_i^{t}}{\partial x_j^{0}} = \sum_{k=1}^{8} \left( \frac{\partial N_k}{\partial x_j^{0}} \right)(x_i^{0l})^k \]  

(3.16)

The origin of the reference coordinate system \((\eta, \rho, \xi)\) is situated in the middle of the element. For the reference coordinates \( \eta = \pm 1, \rho = \pm 1 \) and \( \xi = \pm 1 \), the element nodes are reached. According to the chain rule, the derivative of the current global coordinates to the reference global coordinates, can be written as:

\[ \frac{\partial x_i^{t}}{\partial x_j^{0}} = \frac{\partial x_i^{t}}{\partial \eta} \frac{\partial \eta}{\partial x_j^{0}} + \frac{\partial x_i^{t}}{\partial \rho} \frac{\partial \rho}{\partial x_j^{0}} + \frac{\partial x_i^{t}}{\partial \xi} \frac{\partial \xi}{\partial x_j^{0}} \]  

(3.17)

To obtain the components of equation (3.17), two matrices \( U \) and \( V \) are calculated. Matrix \( U \) contains the derivatives of the current global positions to respectively \( \eta, \rho \) and \( \xi \), while...
Chapter 3: The Hybrid III Neck's Rubber Model

Matrix V contains the derivatives of the reference global positions to \( \eta \), \( \rho \) and \( \xi \). The derivatives can be calculated from equation (3.15) with \( N_k \) as the interpolation functions. The eight interpolation functions depend on the three variables of the reference coordinate system \((\eta, \rho, \xi)\) in the following way:

\[
N_k = \frac{1}{8}(1 \pm \eta)(1 \pm \rho)(1 \pm \xi)
\]  

with \( k \) the node number. Each node has its unique interpolation function, which can be deduced from figure 3.3, i.e. interpolation function \( N_i \) is equal to:

\[
N_i = \frac{1}{8}(1 - \eta)(1 - \rho)(1 - \xi)
\]  

Assuming that the matrices U and V have the following structure:

\[
U = \begin{bmatrix}
\frac{\partial x_1}{\partial \xi} & \frac{\partial x_2}{\partial \xi} & \frac{\partial x_3}{\partial \xi} \\
\frac{\partial x_1}{\partial \eta} & \frac{\partial x_2}{\partial \eta} & \frac{\partial x_3}{\partial \eta} \\
\frac{\partial x_1}{\partial \rho} & \frac{\partial x_2}{\partial \rho} & \frac{\partial x_3}{\partial \rho}
\end{bmatrix}
\]

\[
V = \begin{bmatrix}
\frac{\partial x_1^0}{\partial \xi} & \frac{\partial x_2^0}{\partial \xi} & \frac{\partial x_3^0}{\partial \xi} \\
\frac{\partial x_1^0}{\partial \eta} & \frac{\partial x_2^0}{\partial \eta} & \frac{\partial x_3^0}{\partial \eta} \\
\frac{\partial x_1^0}{\partial \rho} & \frac{\partial x_2^0}{\partial \rho} & \frac{\partial x_3^0}{\partial \rho}
\end{bmatrix}
\]

and inverting matrix V:

\[
V^{-1} = \begin{bmatrix}
\frac{\partial \xi}{\partial x_1^0} & \frac{\partial \eta}{\partial x_1^0} & \frac{\partial \rho}{\partial x_1^0} \\
\frac{\partial \xi}{\partial x_2^0} & \frac{\partial \eta}{\partial x_2^0} & \frac{\partial \rho}{\partial x_2^0} \\
\frac{\partial \xi}{\partial x_3^0} & \frac{\partial \eta}{\partial x_3^0} & \frac{\partial \rho}{\partial x_3^0}
\end{bmatrix}
\]

it follows that:

\[
F^{0-\eta} = (V^{-1}U)^T
\]  

Now that the deformation tensor \( F^{0-\eta} \) is known, it is possible to calculate the right Cauchy-Green strain tensor \( C^{0-\eta} \) for any integration point. With the help of equations (3.7), the second Piola-Kirchhoff stresses can be calculated. For the volume element with reduced
integration, the integration point for a volume element has the position \( \eta = 0, \rho = 0 \) and \( \xi = 0 \). In Reuser (1994), the Mooney-Rivlin material subroutine of MADYMO 5.1 for full integration is presented. The integration points of this element do not coincide with the element nodes, but lie within the element at the reference positions \( \eta = \pm \frac{\sqrt{3}}{3}, \rho = \pm \frac{\sqrt{3}}{3} \) and \( \xi = \pm \frac{\sqrt{3}}{3} \).

Before the material subroutine is used to model the rubber discs of the Hybrid III's neck, the subroutine is tested at an one element cube. The isotropy of the material model was tested by subjecting the cube to a tension and compression test in the principal axes of strain in different cube orientations. It was found that the stress-strain relationship was not dependent on the orientation of the cube, which means that the material is isotropic. Substituting a strain state in the constitutive relationship (3.7) offers a way to determine the stresses manually. Subjecting the cube to same strain state and comparing the stresses of the simulation with the manually obtained stresses yielded the same result.

### 3.4 Locking

The rubber discs of the Hybrid III's neck will be modelled with eight-node solid elements. The used material for the rubber discs is almost incompressible. The combination of standard eight-node solid elements with an (almost) incompressible material model can cause locking problems in certain deformations. The locking-mechanism can be explained on the basis of the figures 3.4 and 3.5.

![Figure 3.4. Deformation of an incompressible reference volume modelled with a linear volume element.](image)

![Figure 3.5. Deformation of an incompressible reference volume in reality.](image)

When incompressibility is assumed for the material model, the volume of a reference volume of material remains constant. Suppose two nodes of an eight-node linear volume element are loaded according to figure 3.4. The volume element is three-dimensional, although the element is depicted in two dimensions. The remaining six nodes are supported in three directions. The two loaded nodes are supported in the direction normal to the paper. The volume of the element remains only constant if the two loaded nodes move along the dashed line. However, physically seen, the loaded nodes should move according to figure 3.5.
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A known solution for the numerical problem of locking is the use of reduced integration for eight-node volume elements. An adventitious advantage is the reduction of the total computational time with approximately a factor three.

3.5 Discussion

In the preceding chapter, the compressible Mooney-Rivlin material model was described and derived. Section 3.3 dealt with the implementation in MADYMO 5.1. Since the required right Cauchy-Green strain tensor, used by the compressible Mooney-Rivlin model, is not standard available in MADYMO 5.1, it had to be calculated first. It was calculated by multiplying the conjugate of the deformation tensor with the deformation tensor itself. The deformation tensor was also needed to convert the second Piola-Kirchhoff stresses (the standard output of the Mooney-Rivlin model) into the (by MADYMO 5.1) required Cauchy stresses.

The locking phenomenon was solved by the use of reduced integration for the eight-node volume elements. As a consequence, the total computational was reduced by approximately a factor four.
4 Modelling the Hybrid III’s Neck

4.1 Introduction

This chapter describes the design and modelling of the Hybrid III’s neck. In section 4.2 the evolution of the first dummy neck to the Hybrid III’s neck is presented, as well as the most significant design features. Slaats’ original finite element model (Slaats, 1990) of the neck can be found in section 4.3. Slaats’ model has been improved because it did not model the steel cable inside the neck. Also the contact interaction between the separate aluminium discs and the material model did not work properly. The improved finite element model is described in section 4.4. The chapter ends with a discussion.

4.2 The Hybrid III’s Neck

One of the early neck designs (Mertz et al., 1973 and Foster et al., 1973) consisted of five steel segments connected by a steel cable. Because of the poor repeatability of the neck in experiments, split ball and socket necks without a cable were introduced. Bending resistance was achieved by placing rubber rings between the separate split ball and socket joints. In the neck of the Hybrid I and Hybrid II, the split ball and socket joints were moulded in rubber. The Hybrid III was developed because the preceding (Hybrid I and Hybrid II) dummies lacked of biofidelity, especially the neck, thorax and knees. The purpose of the split ball and socket joints, to obtain a high axial stiffness, had been taken over by a steel cable inside the Hybrid III’s neck. The split ball and socket joints were removed, however, the aluminium discs were preserved (Philippens, 1989a).

The neck of the Hybrid III dummy is constructed out of two different materials, i.e. rubber and aluminium. Five aluminium circular discs are interconnected by four butyl elastomer rubber circular discs, see appendix A. The rubber discs enable omni-directional stiffness of the neck and have an eccentricity of 6.0 [mm] to the anterior side with respect to the aluminium discs. To decrease the extension stiffness of the Hybrid III’s neck, each rubber disc has a horizontal fissure at the anterior side of the neck. The aluminium discs at the top and bottom of the neck connect the neck to the thorax (bottom side) and head (top side) of the dummy.

Inside the neck of the Hybrid III dummy, a steel cable connects the top aluminium disc to the bottom aluminium disc in order to increase the axial stiffness of the neck. The cable has a pretension of 600 ± 100 [N], see appendix C. The dummy’s head is connected to the neck at the top disc with a revolute joint. Between the head and the top disc, two (anterior side and posterior side) rubber nodding blocks are positioned to eliminate rotational play and to prevent vibrations due to the contact between the head and neck.
Chapter 4: Modelling the Hybrid III's Neck

4.3 The Original Finite Element Model

The finite element model of the Hybrid III's neck has been derived from the original finite element model of Slaats. Slaats (1993) modelled the Hybrid III's neck both in LS/DYNA3D (Whirley et al., 1991) and in PAM-CRASH (Slaats, 1990) for reasons of comparison. The nodal coordinates of the FEM-mesh of the Hybrid III's neck in PAM-CRASH format have been converted to a HyperMesh input file (Altair Computing, 1990) with the aid of a conversion program. The HyperMesh input file was needed to edit the original FEM-mesh. The original finite element model contained volume elements for the rubber discs and shell4 elements (PAM-CRASH version) or volume elements with a rigid material model (LS/DYNA3D version) for the aluminium discs. The choice of the volume elements for the rubber discs is obvious because the dimensions of the rubber discs have the same order of magnitude in the three perpendicular directions.

The aluminium discs in the PAM-CRASH version were modelled as shell4 elements in a honeycomb-like structure. It would have been easier to model the aluminium discs as rigid bodies because the stiffness of the aluminium is much larger than the stiffness of the rubber. However, at that time, supports between nodes of a FEM-structure and rigid bodies gave large problems. Supports attach nodes in one or more of the six degrees of freedom (three translational and three rotational degrees of freedom) to a rigid body. Slaats did model the aluminium discs as rigid bodies in the LS/DYNA3D version.

The contact interaction in the fissures of the rubber discs were modelled differently in the LS/DYNA3D and in the PAM-CRASH version. In LS/DYNA3D, two FEM-meshes were used. One FEM-mesh with fissures for extension of the neck and one FEM-mesh without fissures for the flexion of the neck. The PAM-CRASH version models the contact in the fissure by means of a master-slave penetration algorithm. Therefore, only one FEM-mesh was needed in this version.

In both models, Slaats did not model the steel cable inside the Hybrid III's neck.

4.4 The MADYMO 5.1 Finite Element Model

The improved model of the Hybrid III's neck differs on a number of points from Slaats' neck model, see table 4.1. The three major improvements are the use of rigid bodies for the aluminium discs, the introduction of contact interaction between the aluminium discs and the modelling of the steel cable inside the Hybrid III's neck. In the subsections 4.4.1 to 4.4.7, the most significant improvements are presented as well as the loading and supports of the neck.

4.4.1 Rigid Bodies

The aluminium discs can be seen as rigid with respect to the rubber discs, as the stiffness of aluminium is about ten thousand times the stiffness of the rubber of the Hybrid III. Therefore, it is logical to model the aluminium discs as rigid bodies. By using rigid bodies,
an enormous gain of computational time is achieved, since the integration time step depends on the highest stiffness in the FEM-structure.

**Table 4.1. Differences between the LS/DYNA3D, PAM-CRASH and MADYMO 5.1 model.**

<table>
<thead>
<tr>
<th>Hybrid III</th>
<th>LS/DYNA3D</th>
<th>PAM-CRASH</th>
<th>MADYMO 5.1</th>
</tr>
</thead>
<tbody>
<tr>
<td>Aluminium Discs</td>
<td>Volume Elements</td>
<td>Shell4 Elements</td>
<td>Rigid Bodies</td>
</tr>
<tr>
<td>Rubber Discs</td>
<td>1376 Elements</td>
<td>1376 Elements</td>
<td>1232 Elements</td>
</tr>
<tr>
<td>Fissures</td>
<td>Separate Fem-Meshes for Extension/Flexion</td>
<td>Master/Slave Node Penetration Algorithm</td>
<td>Separate Fem-Meshes for Extension/Flexion</td>
</tr>
<tr>
<td>Aluminium Discs Contact</td>
<td>No</td>
<td>No</td>
<td>Yes</td>
</tr>
<tr>
<td>Steel Cable</td>
<td>Not Present</td>
<td>Not Present</td>
<td>Present</td>
</tr>
</tbody>
</table>

By replacing the volume elements (LS/DYNA3D) and shell4 elements (PAM-CRASH) by rigid bodies, both the number of elements of the original model is reduced with about 25% and the integration time step is increased. The inertial properties of the rigid bodies, which model the aluminium discs, can be found in table 4.2.

**Table 4.2. Inertial properties of the rigid bodies.**

<table>
<thead>
<tr>
<th></th>
<th>m [kg]</th>
<th>(I_{xx} [kgm^2])</th>
<th>(I_{yy} [kgm^2])</th>
<th>(I_{zz} [kgm^2])</th>
</tr>
</thead>
<tbody>
<tr>
<td>Bottom Disc</td>
<td>0.133</td>
<td>7.02\times10^{-5}</td>
<td>7.02\times10^{-5}</td>
<td>1.39\times10^{-4}</td>
</tr>
<tr>
<td>Disc 2, 3 and 4</td>
<td>0.113</td>
<td>5.94\times10^{-5}</td>
<td>5.94\times10^{-5}</td>
<td>1.18\times10^{-4}</td>
</tr>
<tr>
<td>Top Disc</td>
<td>0.352</td>
<td>1.89\times10^{-4}</td>
<td>1.89\times10^{-4}</td>
<td>3.59\times10^{-4}</td>
</tr>
</tbody>
</table>

4.4.2 Finite Elements

The rubber discs in Slaats’ model (Slaats, 1993) are modelled as cylinders. The rubber discs of the Hybrid III’s neck (Philippens, 1989a) however, have a cylindrical core and are wedge-formed towards the aluminium discs. To obtain a more accurate geometrical description of the rubber discs, the FEM-mesh of Slaats’ neck model is regenerated by the finite element mesh program HyperMesh (Altair Computing, 1990). Therefore, a major
difference with respect to the original finite element model are the layers of rubber elements, which cover the complete top and/or bottom side of the aluminium discs, see appendix A. In figures 4.1, 4.2 and 4.3, the improved MADYMO 5.1 finite element model of the Hybrid III's neck is depicted.

4.4.3 Fissures

The fissures at the anterior side of the rubber discs open during extension of the Hybrid III's neck and close during flexion. In MADYMO 5.1, it appeared not possible to apply a contact algorithm in order to introduce resistant forces in the fissures during flexion. Therefore, it was chosen to use two different FEM-meshes: one mesh with fissures for extension of the Hybrid III's neck and one FEM-mesh without fissures for flexion bending. A FEM-mesh for lateral bending appeared not possible, because the fissures open only partially during a lateral loading. The extension, to which the fissures opened, was not known in advance.

The disadvantage of the choice of two FEM-meshes is the limited simulation possibilities. In the dynamic simulations, see chapter 6, an acceleration field in either anterior or posterior direction is applied to the neck. The neck bends in the direction of the acceleration field at first, followed by a rebound as a result of the stiffness of the neck. As mentioned before, either a FEM-mesh for flexion or a FEM-mesh for extension is used. Therefore, only the first extension or flexion bending can be used to validate the dynamical properties of the mathematical model. The static flexion and extension bending have of course no disadvantage.

Figure 4.1. Frontal view of the FEM-neck. Five rigid bodies, four FEM-parts.

Figure 4.2. Right side view of the FEM-neck.
of the two different meshes.

4.4.4 Disc-to-Disc Contact

During large bending deformations, the wedge-formed rubber covering the aluminium discs makes contact. This is called the disc-to-disc contact. To insure contact forces, ellipsoids are connected to the rigid bodies modelling the aluminium discs. The outer surface of the ellipsoids describes the contact area of the rubber covering the aluminium discs. The stiffness of the ellipsoids must be chosen in such a way, that the stiffness of the Hybrid III’s neck is in accordance with static measurements.

4.4.5 Supports

To connect the finite elements to the rigid bodies of the model, supports have to be defined. The nodes of the elements adjacent to the aluminium discs are deprived of their translational and rotational degrees of freedom with respect to the rigid bodies. Also, the bottom disc of the neck is attached to the inertial space. This can be accomplished by means of a bracket (zero degrees of freedom) joint between the inertial space and the bottom aluminium disc.

The dummy’s head is connected to the top aluminium disc by means of a revolute joint. In reality, two nodding blocks are mounted between the head and top disc in order to reduce vibration of the head. The stiffness of the rubber blocks is modelled by defining a joint stiffness of 1042 [Nm/rad] (Kaleps, 1988) of the revolute joint.

4.4.6 Loads

The complete assembly of the 50th percentile Hybrid III’s head and neck can be seen in figure 4.4. The Hybrid III’s head is modelled as a rigid body. The head and aluminium discs in figure 4.4 are generated for visualization purposes, since the rigid bodies have no actual shape. The rigid bodies are entirely defined by their centre of gravity and inertia properties.

To simulate a flexion test, the neck can either be loaded (quasi-)statically or dynamically. A (quasi-)static load has been applied by means of a driver, forcing the neck in flexion. Dynamic loads have been applied by subjecting the finite elements and rigid bodies to a prescribed acceleration field. The result of a simulated dynamic flexion test can be seen in
Chapter 4: Modelling the Hybrid III's Neck

4.4.7 The Steel Cable

By performing static and dynamic simulations with the Hybrid III’s neck model, it has become clear that the cable has a significant influence on the behaviour of the dummy’s neck, see chapter 5. In the original model (Slaats, 1990 and Slaats, 1993), the cable was not modelled. The reason for the absence of the steel cable in the model of Slaats was the difficult interaction between the cable and the inside of the neck.

As in Slaats’ model, the major problem of modelling the cable, is the interaction between the inside of the neck and the cable itself. Since aluminium is much stiffer than rubber, the contact will be determined by the contact between the cable and the aluminium discs. The steel cable is modelled as a belt system, a special feature of MADYMO 5.1. The belt begins at the top disc and ends at the bottom disc with a retractor/pretensioner system. The stiffness of the cable in the simulations is set to $1.0 \times 10^8 \,[N/m]$. The integration time step in MADYMO 5.1 is, amongst other factors, dependent on the stiffest component in the neck model. The steel cable in reality is approximately a factor one hundred thousand times stiffer than the rubber discs. To model the real stiffness of the steel cable would increase the total CPU-time enormously. Therefore, a compromise has been chosen between the rigid behaviour of the steel cable and an acceptable rigid body integration time step. The stiffness of the cable in the performed simulations is chosen to be one hundred times the stiffness of the rubber parts for small deformations. Now, both an acceptable integration time step and a rigid behaviour of the steel cable is achieved.

![Figure 4.4. Dummy's head-neck complex in the initial position.](image1)

![Figure 4.5. Dummy's head-neck complex in flexion.](image2)

The retractor/pretensioner system retracts a predefined length of belt material into the retractor and, therefore, simulates a pretension in the steel cable. The inlet of the pretensioner can be determined by performing a compression test with the neck model, see figure 4.6.
Chapter 4: Modelling the Hybrid III's Neck

From the force-displacement characteristic can be determined that an inlet of approximately 3.0 [mm] results in a cable force of 600 [N]. The used material parameters in the compression test were $A^* = 3.5 \times 10^5 [N/m^2]$, $B^* = 0.0 [N/m^2]$ and $v = 0.49 [-]$. The contact interaction between the belt and the aluminium discs is determined by three slip rings. Each slip ring is placed at the centre of the aluminium discs, except at the bottom and top disc. The slip rings allow the belt to slide freely through the rings. The advantage of this model is the easy implementation of the belt system. It must be mentioned, however, that the slip rings cannot translate in radial direction. In reality, the cable has little play in radial direction, because the diameter of the hole inside the Hybrid III's neck is a little larger than the diameter of the cable. This radial play is neglected in the cable model.

4.5 Discussion
In this chapter, the finite element model of the Hybrid III's neck was presented. The model was derived from the finite element model of Slaats. With respect to the model of Slaats, three major differences can be noticed.

• First, the volume or shell4 elements that Slaats used to model the aluminium discs were replaced by rigid bodies. The result is a decrease of the number of elements and, therefore, a decrease in computational time. A second result of the replacement is an increase in the time integration step, because the small time integration step needed for the high stiffness of the aluminium is no longer necessary. Therefore, an extra decrease of computational time is achieved.

• The second major difference is the introduction of contact interaction between the aluminium discs. This is accomplished by the introduction of ellipsoids connected to the aluminium discs. The outer surface of the ellipsoids describes the contact surface of the wedge-formed rubber of the Hybrid III's neck. In large bending deformation, the contact forces are generated by the ellipsoids and not by the rubber elements themselves. The reason that rubber elements cover the aluminium discs, is the use of element contact interaction in the future.

• The third major difference is the modelling of the steel cable in the neck of the Hybrid III dummy. The belt system model has been chosen to represent the steel cable in the neck because the belt model is easy to insert in the finite element model, the belt model has no
significant influence on the total CPU-time of a simulation and the belt model has many options concerning the setting of stiffneses and pretensions.
Chapter 5: Static Validation of the Hybrid III's Neck

5 Static Validation of the Hybrid III's Neck

5.1 Introduction

The validation of the mechanical properties of the Hybrid III's neck can be divided into two parts: static validation and dynamic validation. This chapter describes the static validation which is used to verify the material properties, the belt model and the stiffness of the ellipsoids which enable the contact between the aluminium discs. The dynamic validation is used to verify the inertial properties and to determine the damping of the neck model. The results of the dynamic validation can be found in chapter 6.

First, the static measurements are presented in section 5.2, followed by the results of the simulations in section 5.3. The conclusions of the static tests are summarized in the discussion, which can be found in section 5.4.

5.2 Static Measurements

In the literature (Baughn et al., 1993), a new technique for determining the bending stiffness of mechanical necks is described. As an example, the bending stiffness of a Hybrid III's neck in flexion, extension and lateral direction was determined. The Hybrid III's neck was attached to a rigid frame at the bottom disc, see figure 5.1.

![Figure 5.1. Setup to determine the static stiffness.](image)

Two large discs were mounted (one disc at each side of the neck), of which the axis of rotation was situated at the bottom disc. A pin through the pivot of the top disc slid through two slots in the rotating discs. This construction enabled the neck to lengthen or shorten during flexion or extension of the neck. The moment of force at the bottom disc was measured, as well as the top disc rotation with respect to the bottom disc. A Linear Variable Differential Transformer measured the distance between the pivot of the top disc and the bottom disc. All measured data can be recognized by symbols in the figures: squares (■), circles (●), triangles (▲, ●), and diamonds (◇).

In figure 5.2, the measured bending characteristics of the Hybrid III’s neck is depicted. The neck was loaded in the flexion, extension and lateral direction. It can be seen that the bending stiffness in flexion is larger than in lateral direction, which itself is larger than the
stiffness in extension. The decrease in stiffness is due to the asymmetry of and to the fissures in the anterior side of the rubber discs. The sudden increase of the flexion stiffness at approximately 1.0 \([\text{rad}]\) rotation of the top disc, see figure 5.2, can be ascribed to the disc-to-disc contact. This effect is also known as the bottoming out of the neck. The measured distance between the top pivot of the neck and the bottom disc during flexion can be seen in figure 5.3.

**Figure 5.2.** Measured bending stiffness of the Hybrid III’s neck.

In the literature (Philippens et al., 1989b), several static tests with the Hybrid III’s neck were performed. One of the tests concerned torsion loading of the neck. The result of the torsion test is depicted in figure 5.4. The angle of rotation is plotted on the abscissa, while the measured torque is plotted on the ordinate.

**Figure 5.3.** Measured compression of the Hybrid III’s neck in flexion.

**Figure 5.4.** Measured torque versus top disc angle of torsion.

The static validation of the Hybrid III’s neck consists of comparing the bending and torsion stiffness of the measurements with the simulations. Also the distance between the top disc pivot and the bottom disc are compared to measurements. Only flexion and extension simulations have been performed. As was mentioned in subsection 4.4.3, it was not possible to create a lateral FEM-mesh and, therefore, no lateral bending simulations could be performed.

To determine the sensitivity of the responses of the neck to the belt model inside the neck,
different pretensions of the belt model have been applied as well as no belt model at all. The results of the different pretensions of the belt model in flexion as well as the cable force during flexion can be seen in figures 5.5 and 5.6. The MADYMO 5.1 input file of a static flexion simulation is added in appendix D. For top disc rotations of approximately 1.0 \([\text{rad}]\) and above, disc-to-disc contact is simulated by means of ellipsoid-to-ellipsoid contact. The linear stiffness of the ellipsoids is 3.6 \([\text{MN/m}]\) and is chosen as a result of several iterations.

A first approximation of the contact ellipsoid stiffness can be determined from figure 5.2. The extra increase of the moment of force from 0.9 \([\text{rad}]\) and above can be ascribed to the contact ellipsoids. Together with the dimensions of the neck, a first approximation of the ellipsoid stiffness can be made. The used material parameters in the static simulations are \(A^* = 3.5 \times 10^5 \text{[Pa]}, B^* = 0.0 \text{[Pa]}, \nu = 0.49 [-]\) and the density \(\rho = 1100.0 \text{[kg/m}^3\text{]}\), see equation (3.4).

![Figure 5.5. Measured and simulated moment of force versus top disc rotation in flexion.](image1)

![Figure 5.6. Simulated cable force versus top disc rotation as function of different pretensions in flexion.](image2)

The most important conclusion drawn from figure 5.5, is the significance of the presence of the belt model and the insignificance of the pretension of the cable. The difference between the presence and absence of the cable becomes larger as the top disc rotation increases. The applied pretensions were no pretension, 400, 600 and 800 \([\text{N}]\) pretension. The insignificance of the pretension of the belt is confirmed by figure 5.6, which shows the cable force versus the top disc rotation. At large top disc rotations, the variation of the cable force of the different pretensions is small compared to the magnitude of the cable force.

The comparison between the simulations and measurements are also made for the extension of the neck. The moment of force of the neck in extension is plotted in figure 5.7, the cable force of the same simulations are depicted in figure 5.8.

The stiffness of the neck model is correct. However, for top disc rotations smaller than 0.8 \([\text{rad}]\), there appears to be an offset of the simulations of the moment of force with respect
to the measurements. Variations of the position of the belt may achieve better results in extension, but these simulations have not been performed, due to the limited time within which this research project was conducted.

Also the almost constant belt force in extension during the first 0.8 \([\text{rad}]\) is peculiar. This constant force can be explained by assuming a hinge point at the anterior side of the belt, around which the aluminium discs rotate during a bending motion. During the first 0.9 \([\text{rad}]\) top disc rotation in extension, the belt is not stretched, which results in a constant belt force. At the moment that disc-to-disc contact occurs, the point of contact acts as a new hinge point at the posterior side. Now, the belt is stretched and the belt force increases as can be seen in figure 5.8. The disc-to-disc contact can also clearly be seen in the figure 5.7. However, no cable force measurements are available and, therefore, this explanation must be taken with certain caution.

![Figure 5.7](image1.png)  
**Figure 5.7.** Measured and simulated moment of force versus top disc rotation in extension.

![Figure 5.8](image2.png)  
**Figure 5.8.** Simulated cable force versus top disc rotation as function of different pretensions in extension.

In figure 5.9, the measured and simulated distance between the top disc pivot and the bottom disc during flexion is depicted. The measured data indicate an increase of the neck length at first. This is caused by shear of the rubber discs. At the moment that the steel cable contacts the interior of the neck, the neck length decreases. In the simulations with the belt system, the neck length only decreases. The reason for this is the presence of the slip rings, through which the belt slips. Since the slip rings have a fixed position, the belt cannot move sideways and, therefore, the neck immediately shortens during bending. It can be seen that the presence of the steel cable highly determines the distance between the top disc pivot and the bottom disc. The reason for this is the rigid behaviour of the belt system with respect to the much less stiffer rubber discs. The pretension does not have much influence on the compression of the neck. In the case of absence of the cable, the neck lengthens. As was mentioned before, this is caused by the shear of the rubber discs.
Figure 5.10. Measured and simulated torque bottom disc versus the angle of torsion.

Figure 5.10 shows the measured and simulated torsion of the Hybrid III's neck. In this simulation, the FEM-mesh with the fissures was applied. The simulated result does not correspond with the measured result very well. In reality, friction in the fissures increases the torsion stiffness. In MADYMO 5.1, it was not possible to model friction between volume elements. A second possible cause for the higher torsion stiffness, could be the winding up of the steel cable inside the Hybrid III's neck during torsion. A nonlinear torsion spring in the neck model could be applied between the top disc and bottom disc in order to improve the torsion stiffness. However, it has the preference to introduce friction between volume elements.

5.4 Discussion

In this chapter, a validation of the static behaviour of the Hybrid III's neck model was performed. Four measurements were used to validate the neck model: static bending measurements in flexion and extension, torsion measurements and the distance measurement between the top disc pivot and bottom disc. The bending simulations in flexion and extension were performed with variation of the belt pretension. The belt pretension varied from no pretension to 800 [N] pretension. Also simulations without a belt system were performed.

From the achieved results the following conclusions can be drawn:
- The neck model predicts the static flexion bending stiffness very good. It appears that the pretension of the belt model has not a large influence on the flexion bending stiffness. However, there is a distinct behaviour of the neck in flexion when the belt model is present or absent. Without the belt system, the flexion bending stiffness is much lower. In flexion bending, the belt force increases monotonous.
Chapter 5: Static Validation of the Hybrid III's Neck

- The prediction of the extension bending stiffness is less good compared to the flexion bending stiffness. The predicted extension stiffness seems to have an offset with respect to the measurements. This could be caused by the almost constant belt force during the first 0.8 \( \text{[rad]} \) rotation of the top disc. This is probably also the cause of the same extension stiffness, predicted with or without the belt system.

- The distance between the top disc pivot and the bottom disc is also reasonably well predicted. As with the flexion stiffness prediction, the pretension does not have a large influence on the predicted distance. Removing the belt system causes the distance between the top disc pivot and bottom disc to increase. This is caused by the large shearing deformation of the rubber discs.

- The torsion stiffness of the neck model is too low compared with the torsion measurements. The reason for this is the absence of friction in the fissures. Friction between volume elements cannot be modelled in MADYMO 5.1. Also the winding up of the steel cable inside the Hybrid III's neck during torsion, probably increases the torsion stiffness of the neck.
6 Dynamic Validation of the Hybrid III’s Neck

6.1 Introduction

The static validation of the Hybrid III’s neck in the preceding chapter was the first part of the comparison of the measurements of the Hybrid III’s neck with the simulations of the neck model. The second part deals with the dynamic validation of the neck model and is described in this chapter. The validation consists of the comparison of the dynamic loading of the FEM-neck model with the dynamic loading of a validated two-pivot neck model. The neck is dynamically loaded in flexion and extension. The dynamic loading of the neck model is a result of applying an acceleration field to the neck and is described in section 6.2. Section 6.3 deals with the flexion dynamic loading, whereas section 6.4 contains the extension dynamic loading. The conclusions of the dynamic validation can be found in the section 6.5.

6.2 Pendulum Acceleration Field

The dynamic loading of the neck model is achieved by subjecting the rigid bodies and FEM-parts of the neck model to an acceleration field. The response of the FEM-model is compared to the official MADYMO 5.1 two-pivot model datadeck of the 50th percentile Hybrid III’s neck, see appendix E. Simulating a complete pendulum test and comparing the responses with the measured data, would consume too much computational time. The datadeck was developed from pendulum tests with the Hybrid III’s neck and head complex and is, therefore, used as reference for the FEM neck model. The pendulum test consists of a long bar, the pendulum, to which at the end the head-neck complex of the Hybrid III is attached. The pendulum has an initial angle with respect to the gravitational direction. The pendulum is released and impacts in a honeycomb, which decelerates the pendulum and the Hybrid III’s head and neck complex. The pendulum test is used for dynamic experiments with the necks of different crash test dummies.

The two-pivot model of the Hybrid III’s neck consists of two links and two pivots. The links represent the neck and the head. Compared with the Hybrid III’s neck, the pivots are located at the bottom aluminium disc and at the top disc pivot (OC of the head).

The applied acceleration field is, at first, generated by the datadeck and simulates the pendulum end acceleration in local x-direction. For the orientation of the coordinate system, see the figures 4.1 to 4.3. The pendulum end velocities are 5.9 [m/s] in extension and 6.0 [m/s] in flexion. The first 1000 [ms] of the pendulum test are used to increase the pendulum velocity. After 1000 [ms], the pendulum impacts into the honeycomb and the neck bends in flexion or extension. As the neck virtually does not deform during the first 1000 [ms], the acceleration signal is shifted, in such a way that the acceleration/deceleration of the pendulum begins at approximately 50 [ms], see figure 6.1. As can be seen, the pendulum decelerates with an almost constant level of approximately 250 [m/s²] during
Chapter 6: Dynamic Validation of the Hybrid III's Neck

40 [ms]. The real pendulum acceleration does also contain a z-component but this is omitted, because this component is small compared to the acceleration in local x-direction (15% peak value). The obtained acceleration field is applied to both the FEM-model and to a slightly adjusted datadeck. The adjusted datadeck does not contain the pendulum body anymore, in order to obtain a similar model setup as the FEM-model.

6.3 Flexion Loading

Before the acceleration field is applied to the rigid bodies and FEM-parts, the belt system in the FEM-neck model is pretensioned during the first 20 [ms] of the simulation. The time between 20 and 50 [ms] is used to damp the oscillations of the neck caused by the sudden pretensioning of the belt model. In order to get a clear signal of the linear and angular accelerations of the FEM-model, the accelerations have been filtered with a CFC 180 filter. This filter is in conformity with SAE document J211, October 1988. The unfiltered acceleration signals are added in appendix F.

At time 50 [ms], an acceleration field of approximately 25g acts upon the model. The MADYMO 5.1 input file of a dynamic flexion simulation is added in appendix G.

The results of the flexion simulations with the FEM-model and the two-pivot datadeck can be seen in the figures 6.2 to 6.6. Only the first 160 [ms] are displayed, because in the interval from 50 to 160 [ms] the neck moves from the initial, upright position to full flexion and back to the upright position. Beyond this interval (>160 [ms]), the neck is loaded in extension. This
part of the simulation cannot be compared to the validated two-pivot model, because the fissures in the anterior side of the neck are not present. For the extension simulation, a different FEM-mesh with fissures is used (section 6.4).

In figure 6.2, the head centre of gravity (CG) local x-acceleration is depicted. The time of impact and amplitude between the FEM-model and two-pivot model corresponds well. The backwards acceleration of the head begins for both models at approximately 90 [ms], but the two-pivot model predicts an acceleration at a higher level than the FEM-model. Between 90 and 120 [ms], the FEM-model neck shows an oscillation, which is an eigen frequency in flexion bending mode of the neck. The x-acceleration beyond 120 [ms] of the FEM-model and two-pivot model are rather similar. Although the filtering, the acceleration in z-direction begins with an oscillation, see figure 6.3. As mentioned before, this is caused by the pretensioning of the neck in z-direction. The path of acceleration in z-direction for both models correspond well, besides a higher frequency oscillation of the FEM-model neck. The reason is, probably, the axial stiffness of the FEM-neck.

The head angular acceleration (figure 6.4) begins at 50 [ms] and ends at approximately 90 [ms]. The head acceleration of the FEM-model is larger in amplitude than the angular acceleration of the two-pivot model from 50 to 80 [ms]. This can be ascribed to an eigen frequency of the FEM-neck. The predictions of both models predict practically the same
It can be seen in figure 6.5, that the maximum displacement of the head CG in x-direction of the FEM-model agrees well with the prediction of the validated two-pivot model. However, the time of maximum displacement occurs somewhat later in the FEM-model. An explanation could be a too low dynamic stiffness of the neck and the lack of material damping. The FEM-model predicts the z-component of the head CG (figure 6.6) too low with respect to the two-pivot model. Again, this is ascribed to a too low dynamic stiffness or damping of the neck model in flexion.

6.4 Extension Loading

As with the flexion loading, the extension loading is also preceded by the pretensioning of the belt system. The pretension is applied from 0 to 20 [ms]. The following 30 [ms] are used to damp the oscillations of the neck. As in flexion, the linear and angular accelerations of the FEM-model are filtered with a CFC 180 filter. The unfiltered signals are added in appendix F.

At 50 [ms] the acceleration field forces the neck in extension. In the FEM-model, the fissures at the anterior side of the rubber discs open.

In the figures 6.7 to 6.11, the head acceleration and displacements in extension are shown. Because the FEM-neck model is only valid in extension, only the first 170 [ms] of the simulation are displayed. Beyond 170 [ms], the neck rebounds in flexion. As the elements which surround the fissures in the FEM-mesh cannot generate a resistant force, the surrounding elements penetrate each other. This is not conform the reality, which makes the simulation useless beyond 170 [ms].

The head CG x-acceleration (figure 6.7) of the FEM-model and the two-pivot model correspond well. The acceleration paths of the two models beyond 90 [ms] differ a little as
a result of high frequency oscillations of the FEM-model neck. In figure 6.8, the z-acceleration of the head CG begins at the first 20 [ms] with some high frequency oscillation caused by the belt system. In the rest of the simulation, the FEM-model acceleration corresponds well with the two-pivot acceleration.

The head angular acceleration is shown in figure 6.9. The global paths of both models correspond, but the amplitude of the FEM-model is larger. An explanation could be a too low stiffness or damping of the material of the FEM-model neck. From the static tests (chapter 5), it was concluded that the stiffness for the static tests was correct. Adding damping to the material model may give better results.

The maximum of the head CG x-component of both models are practically equal, as can be seen in figure 6.10. As in flexion, the time of maximum displacement of the FEM-neck model occurs somewhat later. The maximum of the head CG z-component (figure 6.11) is larger than the maximum displacement of the two-pivot model. As in flexion, this is ascribed to a too low dynamic stiffness and the lack of material damping.

6.5 Discussion

In this chapter, a dynamic validation of the FEM-neck model was presented. Due to the lack of contact interaction of the fissures in the FEM-mesh, it was necessary to use two FEM-meshes: one mesh with fissures and one without fissures. The FEM-mesh with the
fissures was used for the extension simulations, whereas the mesh without the fissures was used for the flexion simulations. During the first 20 [ms] of the simulations, the belt system was pretensioned. This caused frequencies to such a degree, that a CFC 180 filter had to be applied in order to compare the FEM-model accelerations with the two-pivot model accelerations. This filter had practically no influence on the accelerations beyond the 50 [ms], from which on the acceleration field acted upon the FEM-model.

From the dynamic comparison of the FEM-model simulations with the two-pivot model simulations, the following remarks can be stated:

- The linear and angular accelerations of the Hybrid III’s in both flexion and extension for the FEM-model and the two-pivot model globally correspond. The differences between the two models are mostly due to the more and higher eigen frequencies of the FEM-model. The two-pivot model only has two eigen frequencies. The eigen frequencies of the FEM-model resulted mostly in somewhat higher linear and angular accelerations with respect to the two-pivot model.

- The maximum of head CG x-component for the FEM-model and the two-pivot model correspond well. This applies for both the extension and the flexion simulations. The time of maximum displacement is somewhat later of the FEM-model. This also applies for both the extension and flexion simulations.

- The maximum of the head CG z-component are larger for the FEM-model in both extension and flexion. The time of maximum displacements also occurs at a later time in the FEM-model. These defects can probably be ascribed to a too low dynamic stiffness or to the lack of damping of the material model.

- From the dynamic flexion simulations, it followed that the hydrostatic stress in the rubber discs varied from $-1.5\times10^6$ [Pa] (compression, anterior side) to $5\times10^5$ [Pa] (tension, posterior side). The effective logarithmic von Mises strain in the rubber discs varied from 0.7 (compression, anterior side) to 0.4 (tension, posterior side).
Chapter 7: Conclusions and Recommendations

7 Conclusions and Recommendations

7.1 Conclusions

The research described in this report was performed as a part of a larger project concerned with the design and building of a new omni-directional biofideic neck. This report described the modelling of the existing 50th Hybrid III’s neck in MADYMO 5.1. This has been done in order to obtain experience in the field of FEM-modelling with MADYMO 5.1. As the new omni-directional biofideic neck probably also partly consists of rubber, the Hybrid III’s neck acts as a basis from which the new neck will be designed. The developed Hybrid III’s neck model is able to predict the accelerations of the head CG with satisfactory accuracy. The displacements perform a little less good, but none the less the model can be used to start with the design of the new neck on the basis of the preliminary performance requirements.

From the performed research, the following conclusions can be drawn:

- The static simulations showed a significant increase in flexion bending stiffness when a cable model was applied. The extension bending stiffness was practically not influenced during the time that the belt system was not tensioned. The pretension of the belt system in the static flexion and static extension simulations indicated practically no influence on the bending stiffness in flexion and extension.
- The neck model showed a good performance in static flexion. The disc-to-disc contact (or the bottoming out effect), achieved by contact ellipsoids, perform good. The extension stiffness showed an offset with respect to the measurements. This is probably caused by the constant cable force during extension. In flexion, the cable force increased monotonous.
- The static torsion simulation shows a linear torque versus angle of torsion characteristic. The measurements showed a higher stiffness than the simulations. This has been ascribed to the friction in the fissures. MADYMO 5.1 is not able to model the friction between volume elements. Also the steel cable probably winds up during a torsion test, which increases the torsion stiffness.
- Good performance was found for the linear and angular accelerations of the head CG in both flexion and extension. The accelerations between the FEM-model and the two-pivot model differed a little as a result of more and higher eigen frequencies of the FEM-model.
- The maximum of the head CG x-component in both flexion and extension was predicted very well. Only the time of maximum displacement differed a little. The maximum of the head CG z-component was predicted too large by the FEM-model. This has been ascribed to a too low dynamic stiffness of the material model and to the lack of damping of the material model.
- The contact interaction between volume elements does not work in the Hybrid III’s neck.
model, despite the fact that MADYMO 5.1 does claim to have contact interaction. When contact interaction for the fissures in the neck model can be applied, it is no longer necessary to have separate meshes for flexion and extension tests.

7.2 Recommendations

From the conducted research and the preceding conclusions, the following recommendations can be made:

• Since MADYMO 5.1 pretends to support contact interaction between volume elements, it would simplify the Hybrid III’s neck model if contact interaction could be applied to the fissures. Also the development of friction between volume elements is recommended.
• Determining the influence of the belt system position could improve the static behaviour in extension.
• Since the neck model performs reasonably well, it is recommended to start with the design of the new omni-directional biofidelic neck.
Chapter 8: References

8 References

Altair Computing, Incorporated

Baughn, D.J., E.K. Spittle and G. Thompson
A New Technique for Determining Bending Stiffness of Mechanical Necks

Bogert, P.A.J. van den
Computational Modelling of Rubberlike Materials

Boogaard, A.H. van den
De Berekening van Spanningen en Vervormingen in Autobanden, in Dutch

Brekelmans, W.A.M.
Niet-Lineaire Mechanica, Basis, in Dutch
Eindhoven University of Technology, Department of Mechanical Engineering,

Bruijs, W., M. Philippens and J.S.H.M. Wismans
Status Report: Omni-Directional Neck Development Project

Kaleps, Ints.
Measurement of Hybrid III Dummy Properties and Analytical Simulation Data Base Development
Harry G. Armstrong Aerospace Medical Research Laboratory,

Foster, J.K., J.O. Kortge and M.J. Wolanin
Hybrid III - A Biomechanically Based Crash Test Dummy
Chapter 8: References

MADYMO User's Manual, Version 5.1

Mertz, H.J., R.F. Neathery and C.C. Culver
Performance Requirements and Characteristics of Mechanical Necks
Human Impact Response, Measurement & Simulation, edited by W.F. King and

Mooney, M.
A Theory of Elastic Deformations
Journal of Applied Physics 11, Volume 1, pp. 582-592, 1940.

Ogden, R.W.
Non-Linear Elastic Deformations

Philippens, M., J.S.H.M. Wismans and J.J. Nieboer
50th Percentile Hybrid III Database Development, Volume I

Philippens, M., J.S.H.M. Wismans and J.J. Nieboer
50th Percentile Hybrid III Database Development, Volume I

Reuser, R.F.J.
FEM-Modelling the Hybrid III's Neck

Rivlin, R.S.
Large Elastic Deformations of Isotropic Materials IV, Further Developments of the
General Theory

Schaap, P.
Mathematical Modelling of Dummy Head-Neck Assemblies with MADYMO
Siemerink, H.W.
Analysis of Mathematical Hybrid III and Human Head-Neck Analog Systems

Slaats, P.M.A.
Numerical Modelling of the Hybrid III Dummy Neck using PAM-CRASH and
Comparison with Experiments
TNO draft report 751080020, TNO Road-Vehicles Research Institute, Delft, 1990.

Slaats, P.M.A.
Finite Element Computations and Experiments on the Hybrid III Head-neck Assembly
TNO-report 750210068, TNO Road-Vehicles Research Institute, Delft, 1993.

Tolner, E.
Bepaling van een Materiaalmodel van het Rubber in de Hybrid III-nek, in Dutch

Treloar, L.R.G.
The Physics of Rubber Elasticity

Valanis, K.C. and R.F. Landel
The Strain-Energy Function of a Hyperelastic Material in Terms of Extension Ratios

Whirley, R.G. and J.O. Hallquist
DYNA3D, a Nonlinear, Explicit, Three-Dimensional Finite Element Code for Solid and
Structural Mechanics - User Manual
Lawrence Livermore National Laboratory, Methods Development Group Mechanical
Appendix A

Dimensions of the Hybrid III's Neck as Basis for the FEM-Mesh

Right side view of the Hybrid III's neck.
Frontal view of the Hybrid III’s neck.
Appendix B
Derivation Material Parameters C* and D*

Equations (B.1) depict the constitutive relationship between the stresses and strains of the compressible Mooney-Rivlin material model derived in chapter 3:

\[
P_{2,xx} = (A^*+B^*)(C_{yy}+C_{zz}) - 2J_3^{-3}C^*(C_{yy}C_{zz}-C_{yz}^2) + 2D^*(J_3^{-1})(C_{yy}C_{zz}-C_{yz}^2) \\
P_{2,yy} = (A^*+B^*)(C_{xx}+C_{zz}) - 2J_3^{-3}C^*(C_{xx}C_{zz}-C_{xz}^2) + 2D^*(J_3^{-1})(C_{xx}C_{zz}-C_{xz}^2) \\
P_{2,zz} = (A^*+B^*)(C_{xx}+C_{yy}) - 2J_3^{-3}C^*(C_{xx}C_{yy}-C_{xy}^2) + 2D^*(J_3^{-1})(C_{xx}C_{yy}-C_{xy}^2) \\
P_{2,xy} = 2(-B^*C_{xy}-2J_3^{-3}C^*(C_{yy}C_{xz}-C_{yzzz}C_{xy}) + 2D^*(J_3^{-1})(C_{yy}C_{xz}-C_{yzzz}C_{xy})) \\
P_{2,yz} = 2(-B^*C_{yz}-2J_3^{-3}C^*(C_{xz}C_{xy}-C_{xx}C_{yz}) + 2D^*(J_3^{-1})(C_{xz}C_{xy}-C_{xx}C_{yz})) \\
P_{2,xz} = 2(-B^*C_{xz}-2J_3^{-3}C^*(C_{xz}C_{yz}-C_{xx}C_{xz}) + 2D^*(J_3^{-1})(C_{xz}C_{yz}-C_{xx}C_{xz}))
\]

(B.1)

with:

\[
J_3 = C_{xx}C_{yy}C_{zz} + C_{xy}C_{yz}C_{zz} + C_{yx}C_{zy}C_{zz} - C_{xx}C_{yy}C_{zz} - C_{yy}C_{zz}C_{xz} - C_{zz}C_{xy}C_{xx}
\]

(B.2)

Suppose the material is not pretensioned when the material is not deformed. No strain yields for the right Cauchy-Green strain components:

\[
C_{ij} = \begin{bmatrix}
1 & 0 & 0 \\
0 & 1 & 0 \\
0 & 0 & 1
\end{bmatrix}
\]

(B.3)

Substituting the components of the right Cauchy-Green strain (B.3) into equation (B.1) yields the following condition for C*:

\[
C^* = \frac{1}{2}A^* + B^*
\]

(B.4)

Suppose an uniaxial stress test is performed with a cube of rubber. The right Cauchy-Green strain for this test is:

\[
C_{ij} = \begin{bmatrix}
(1 + \varepsilon)^2 & 0 & 0 \\
0 & (1 - \nu \varepsilon)^2 & 0 \\
0 & 0 & (1 - \nu \varepsilon)^2
\end{bmatrix}
\]

(B.5)

with \(\nu\) as Poisson’s ratio and \(\varepsilon\) as the linear strain in local x-direction. In an uniaxial stress test, no stresses occur perpendicular to the direction of the strain \(\varepsilon\). This means
\( P_{2,yy} = P_{2,zz} = 0 \). Substituting the right Cauchy-Green strain (B.5) into \( P_{2,yy} \) or \( P_{2,zz} \) of equation (B.1) and taking equation (B.2) into account yields:

\[
2(A^* + B^*((1 + \varepsilon)^2 + (1 - \nu e)^2) - 2C^*(1 + \varepsilon)^4(1 - \nu e)^4) + 2D^*((1 + \varepsilon)^2(1 - \nu e)^2 - 1)(1 + \varepsilon)^2(1 - \nu e)^2) = 0 \tag{B.6}
\]

or

\[
A^* + B^*((1 + \varepsilon)^2 + (1 - \nu e)^2) - 2C^*(1 + \varepsilon)^4(1 - \nu e)^4 + 2D^*((1 + \varepsilon)^2(1 - \nu e)^2 - (1 + \varepsilon)^2(1 - \nu e)^2) = 0 \tag{B.7}
\]

Applying a first order Taylor approximation:

\[
y(\varepsilon) = y(0) + \varepsilon y'(\varepsilon) \bigg|_{\varepsilon=0} \tag{B.8}
\]

to equation (B.7) gives:

\[
A^* + B^*((2 + 2\varepsilon - 2\nu e) - 2C^*(1 - 4\varepsilon + 10\nu e) + 2D^*(2\varepsilon - 4\nu e) = 0 \tag{B.9}
\]

Combining equation (B.4) with equation (B.9) yields:

\[
A^* + B^*((2 + 2\varepsilon - 2\nu e) - 2(\frac{1}{2}A^* + B^*) (1 - 4\varepsilon + 10\nu e) + 2D^*(2\varepsilon - 4\nu e) = 0 \tag{B.10}
\]

or

\[
D^* = \frac{(5\nu - 2)}{2(1 - 2\nu)} A^* + \frac{(11\nu - 5)}{2(1 - 2\nu)} B^* \tag{B.11}
\]
Appendix C
Calculation Cable Pretension

The steel cable inside the Hybrid III's neck is pretensioned by applying a moment of force on the nut at the bottom side of the steel cable. The pretension can be calculated\(^1\) by:

\[
M = F_p [0.16p + \mu_t (0.58d_2 + D_m/2)]
\]

with:
- \(M\) = Applied moment of force \([Nm]\).
- \(F_p\) = Pretension steel cable \([N]\).
- \(p\) = Thread \([m]\).
- \(\mu_t\) = Thread and supporting surface nut's mean coefficient of friction.
- \(d_2\) = Thread's basic effective pitch diameter \([m]\).
- \(D_m\) = Mean diameter supporting surface nut \([m]\).

The used variables in equation (C.1) are:

\[
\begin{align*}
M &= 1.35 \pm 0.27 \ [Nm]^2, \\
p &= 1.27 \cdot 10^{-3} \ [m] \text{ (measured)}, \\
\mu_t &= 0.14 \ [-] \text{ (mean average\(^1\))}, \\
d_2 &= 12.1 \cdot 10^{-3} \ [m] \text{ (measured)}, \\
D_m &= 15.4 \cdot 10^{-3} \ [m] \text{ (measured)}.
\end{align*}
\]

Substituting the measured and assumed values into equation (C.1), yields the required pretension: 

\(F_p = 600 \pm 100 \ [N]\).

---

1\(^1\) PolyTechnisch Zakboekje, 45\(^{e}\) druk, 1993, page H/18.
Appendix E

MADYMO 5.1 Input File Two-Pivot Model

for non-rectangular non-linear functions

\[ \text{C(pgamma)} \]

for forward flexion only

5

-3.1415 0.0 -1.5708 1.0 0.0 1.0 1.5708 0.0

3.1415 0.0

9

C(pgamma) function, lateral flexion only

5

-3.1415 0.0 -1.5708 1.0 0.0 0.0 1.5708 1.0

3.1415 0.0

16

C(pgamma) function, rearward flexion only

6

-3.1415 1.0 -1.5708 0.0 1.5708 0.0

3.1415 1.0

neck characteristics: Reumann (1993) per joint in 2 pivot neck

11: forward flexion

7

0.0 0.0

2.018 0.6592

0.1415 0.8544

0.6343 1.0142

0.5236 2.0197

0.6109 3.7215

0.6097 4.7285

0.0000 1.0

12: lateral flexion

7

0.0 0.0

2.054 17.0757

0.2464 78.6408

0.4365 154.6546

0.5236 130.8532

0.6109 242.2224

0.6032 203.4584

13: extension

8

0.0 0.0

0.0073 17.4707

0.2618 38.3595

0.3492 48.2477

0.5432 99.2257

0.5334 71.4467

0.6109 97.6974

0.0702 128.1539

14:

chin limited backward % deg.

1.0 1.0 -1.0

0.0 0.0 0.0

0.0 0.0 0.0

0.0 0.0 0.0

0.0 0.0 0.0

0.0 0.0 0.0

FUNDAMENTAL FUNCTIONS

4: chin to flat plate contact

6

0.0 0.0 0.057 94.7 0.0031 364.8 0.0041 718.4 0.0051 1324.8

0.0569 3007.3

INITIAL CONDITIONS

JOIN 1

P 1 2 3 4 5 6 7

P 1 2 3 4 5 6 7

FORCES

2

0.0 -9.81 2.0 -9.81

COFF: CONTACT INTERACTIONS

ELBOW-ELBOW UND

END CONTACT INTERACTIONS

END FORCE MODELS

OUTPUT CONTROL PARAMETERS

END
ANGEL
1 1 0
1 1 0
-999
ANGACC
1 1 0
1 1 0
-999
MRQU1
1 1 1
-999
END OUTPUT
END INPUT
Appendix F
Unfiltered Acceleration Signals

Figure F.1. Head CG x-acceleration in flexion.

Figure F.2. Head CG z-acceleration in flexion.

Figure F.3. Head angular acceleration in flexion.

Figure F.4. Head CG x-acceleration in extension.
**Figure F.5.** Head CG z-acceleration in extension.

**Figure F.6.** Head angular acceleration in extension.
Appendix G
MADYMO 5.1 Input File Dynamic Simulation

1 Hybrid IIT neck and head for comparison with pendulum experiments.
2 Flexible, dynamic loading.
3 SOLID elements.
4 IRS interaction (F). The 4.4 m/s pendulum X-acceleration is applied to the rigid bodies and the head.
5 4th order ellipsoid-ellipsoid contact, stiffness 3.6 (N/m).
6 Material properties: $A = 3.085$ (PA).

`FUNCTIONS
ORDINATEs:
*head orientations
$A = 4 \times 1.2 \times 3.77$

END SYSTEM 1

END MODEL

FORCE MODEL

ACCELERATION fields
$A = 1.0 \ 0.0$
$1 \ 2 \ 0 \ 0$
$1 \ 3 \ 0 \ 0$
$1 \ 4 \ 0 \ 0$
$\ldots

END MODEL

END CONTACT

END BODY 1

END simulations

G.1