A Multi-Scale Approach to Modeling of Tire Road Interaction

R.J. van Roij
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Committee
prof.dr.ir. M.G.D. Geers (Chairman, TU/e)
dr.ir. J.A.W. van Dommelen (Coach, TU/e)
dr.ir. I. Lopez Arteaga (TU/e)
ir. R. van der Steen (TU/e)
Eindhoven University of Technology
Department of Mechanical Engineering
Section Mechanics of Materials
Chapter 1

Summary

The concept of a rolling tire in contact with the road surface is a complicated problem with many non-linearities such as large deformations, non-linear boundary constraints (contact between the tire and the road surface), non-linear tire and pavement material characteristics and complex geometry and structure of the tire and the road. Modeling all of these details will result in a large model requiring enormous CPU power. The main goal of this investigation is describing the interaction between the road surface and the car tire. As minimizing the size of the model results in decreasing computation time and reducing costs, the detailed modeling will be restricted to the contact region, as opposed to the whole tire. However, partly modeling a car tire like this will result in the application of complex boundary conditions. For this purpose a multi-scale model has been developed in Abaqus. It consists of two coupled models describing two characteristic length scales of a car tire-road surface interaction. In the global model, the larger length scale is represented, which captures the the whole car tire and general geometry of the road surface. In the local model the smaller length scale is represented, which captures the contact patch with tread pattern and the detailed geometry of the road surface. The multi-scale coupling is applied to three test cases. The friction homogenization introduces a small error, however, the coupling still shows acceptable results. When the multi-scale approach is applied to a car tire-road surface interaction problem a corresponding result between the global model’s friction pattern and the local model’s friction pattern is achieved. However, the symmetry conditions applied are not ideal, since an undesired shearing of elements occurs, which is probably caused by the reinforcements. Furthermore it appears that applying the stiffness homogenization to only the rubber material is insufficient.
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Chapter 2

Introduction

The automobile industry has been expanding continuously since the car has been introduced. Our society not only thrives but very much depends on it. Since the beginning, the car has been subject to development and improvement to fulfil the population’s desire. The cars are now faster, safer and more fuel economic than ever. But not only the cars have undergone major developments, the roads that lead these cars the way to their destination as well. Where there used to be only sandy roads for coaches, there are now six-lane freeways, viaducts and kilometers long tunnels. And science isn’t standing still. Both car and road surface are being studied to improve their performance even more. One of the aspects that has been given a lot of attention is the tire road surface interaction. The contact between the car tire and the road-surface is of major importance for the safety, noise production and tire/pavement wear. A lot of research has been done to optimize the tire and the pavement in favor of these aspects. Since there are a lot of different types of tires and pavement, each with its own characteristics, one can imagine that performing either experiments or simulations with all of these different tire road interactions would cost a huge amount of time and money. So one of the challenges is to minimize this cost.

This study focuses on the interaction between a tire and the road surface. To gain more insight in this interaction and how it influences the aspects mentioned earlier, a numerical three-dimensional model of this contact problem is required. The concept of a rolling tire in contact with the road surface is a complicated problem with many non-linearities such as large deformations, non-linear boundary constraints (contact between the tire and the road surface), non-linear tire and pavement material characteristics and complex geometry and structure of the tire and the road. In modeling this problem, three different aspects can be distinguished:

- Geometry
- Boundary conditions and contact
Material

The geometry of a car tire is very complex; all kinds of grooves and sipes (the tire tread) cover the outer surface. The road surface with a grid of stones and pebbles is complex as well. In the contact region all details that are characteristic for describing the interaction have to be captured. Herein, the tread and the road surface grid are key elements. However, modeling all of that precisely would result in an enormous model.

When modeling the geometry, one should keep in mind that the main goal of this investigation is describing the interaction between the road surface and the car tire. And as minimizing the size of the model results in decreasing computation time and reducing costs, the detailed modeling will therefore be restricted to the contact region, as opposed to the whole tire. However, partly modeling a car tire like this, results in the application of complex boundary conditions. In this study these boundary conditions are derived from a less detailed whole tire model. So the approach is split up into two parts. A detailed model that accurately describes the interaction region and a global model which is used for the derivation of the boundary conditions for the detailed model. To complete the coupling, this detailed model can then again be used for improving the interaction characteristics of the global model. In figure 2.1 the length scales that are important for the global model are shown and in figures 2.2 and 2.3 the length scales that are important for the local model are shown.

In the next chapter an overview is given of several tire modeling techniques used in literature. One of these techniques, a global-local approach, is improved and used in this investigation. In chapter 4 the improved approach is discussed and the global and local model are presented. In chapter 5 the technical details of the multi-scale approach are elaborated. A validation of the coupling procedure of the multi-scale approach is discussed in chapter 6. In chapter 7 the multi-scale procedure is applied in a tire-road simulation. Finally, conclusions and recommendations are given in chapter 8.
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Figure 2.1: Characteristic length scales of the whole tire.

Figure 2.2: Characteristic length scales of the profile.

Figure 2.3: Characteristic length scale of the road surface.
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Chapter 3

Literature survey

A tire has a complex structure. That is a soft rubber material reinforced with stiff cords. These cords can be made of steel, polyester, nylon or rayon. In figure 3.1 a cross-section of a radial tire is shown. Generally the basic radial tire design is built up of five main components. These are the beads that lock the tire onto the rim, the plies that extend from bead to bead and ensure radial strength, the belts that provide rigidity to the tread, the body, in which all is embedded, and the tread that is the part in contact with the road surface. For the characteristics of the geometry and the structure of tires one is referred to Clark [1].

Figure 3.1: Structure of a tire, modified from Rodgers [19]

Much research has already been done on the interaction between a tire and a road surface and related topics. In literature tire models vary from simple 2D models to more complex 3D models. 2D models do generally have a much lower CPU time than 3D models. However, the 2D models lack the
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ability to take tread detail into account. 3D models generally take tread detail into account. Most of the more simple 2D models found in literature, are not used for the investigation of tire wear and/or pavement damage, but for research on acoustic and vibrational phenomena [2, 3]. In [4], however, a 2D model is used for tire-soil interaction. The tread, the air-filled volume, the cords and the rim are assumed not to be significantly influenced in the third direction, only the carcass is. The carcass is then modeled by a force deflection relation, derived from an equilibrium between the tire pressure and the local tension forces in the carcass.

For the study of tire pavement interaction, 3D geometric characteristics and full tread detail have been found to be essential [5]. Nowadays, these complex and heavy numerical tire models are allowed for by the increasing power of computers. Nevertheless reducing CPU time still remains a major objective.

A technique used for reducing computational time is the global-local procedure by Mundl [6]. This method separates the model into two parts: the global tire and the detailed tire tread. By first determining the global behavior of the tire in contact with the road surface using a simple model and then using the obtained results for simulation of a local detailed model, computational time can be reduced. For the underestimation of the tire stiffness, a correction shift of the road surface is done. This method allows for easily interchanging and comparing various tread patterns. However, since this method doesn’t take into account the interaction between the tread blocks and the belts inside the tire, it results in less accurate predictions. Therefore Cho found this ‘global-local’ technique not an appropriate choice for the analysis of a car-tire road surface contact problem [5].

The finite element models used in some of the papers use brick elements. However, this choice may not be the best considering the geometry of the tire, which is axisymmetric in the unloaded configuration. Although brick elements are generally more cost effective than cylindrical elements, an axisymmetrical model can usually be described much more accurate using cylindrical elements. Danielson et al. [7] claim that in a tire analysis cylindrical elements should be preferred above bricks for computational efficiency reasons. Herein it is shown that cylindrical or axisymmetrical structures can be described with cylindrical elements just as accurate as using brick elements, only with less element. A combination of cylindrical and brick elements belongs also to the possibilities, which allows to benefit from both the advantages [18].

Material modeling is one of the most extended and complex parts of tire modeling. The model of the tire can be separated in the same five parts as discussed earlier. These parts can be modeled in different ways, and in that way taking more or less detail into account. The body and the tread part are commonly modeled using solid elements with a specific isotropic hyperelastic material model, e.g. Mooney Rivlin [8, 9]. These constitutive
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models are capable of describing global large strain elastomeric behavior, however, do not take effects such as hysteresis, rate dependence, Mullins effect and temperature dependence into account. A study of Bergström on suitable tire material models [10] shows that the Bergström-Boyce model [11] combined with either the Qi-Boyce model for the Mullins effect or Ogden-Roxburgh model for the Mullins effect describes the time dependent behavior of elastomers very well. However no material parameter values are given. A paper that does report a material model and its corresponding parameter values for a tire rubber is that of Hofstetter [17]. In this research Hofstetter uses the material model of Yeoh.

The reinforced cords embedded in the tire are made of steel, polyester, nylon or rayon. The material of the body and the tread is considered to be isotropic. The belts, plies and beads could be modeled using separate elements for the beads and for the surrounding material. This would, however, result in an extremely detailed model with a huge computation time. Therefore these composite materials are commonly considered as homogeneous highly anisotropic materials. The Representative Volume (RV) technique used by [12] is a homogenization method that first considers a small part of the material in detail. This small part is then modeled in such a way that it represents the real composite material by periodic repetition. Another technique commonly seen in tire modeling is using elements with rebar layers [9, 13], in which two elements are laid on top of each other and the nodes are constrained to each other. The two elements both describe a different material behavior. In the case of a tire, that is one element describing the rubber material behavior and one element describing the belt/plie material. This results in an increase of elements, however, since the nodes of the elements are constrained to each other no additional degrees of freedom are introduced. Although using solid elements (with rebar planes) for the homogenized material will be effective for a static analysis, for a dynamic analysis it will result in enormous CPU time. Cho et al. [14] modeled the belt and plie parts with composite shell elements, and thereby reduced the CPU time. However one should keep in mind that, because of the high ratio between the stiffness of the cords and the rubber, it is likely to result in a poorly conditioned elasticity matrix [13].

One of the most crucial aspects of a tire simulation is the contact model that describes the interaction between the tire and pavement. Therefore in literature several methods are found. The penalty method [15] uses a penalty term, that increases with increasing overlap. This method, however, results in overlapping of the two interacting faces. The variable constraint method avoids that and is a simpler and more accurate technique [16]. This method assumes a contact boundary, however it does not let the nodes settle down until a converged solution is obtained.

The friction between the two surfaces is part of the contact model as well. One of the simplest is that of Coulomb, however, since velocity and
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temperature may play an essential role in this tire road contact, a more advanced model may be required. Hofstetter [17] successfully uses a friction model taking the velocity and temperature dependence into account by the Huemer formulation and WLF (Williams-Landel-Ferry) transportation respectively.

Determining the boundary conditions can be another difficult task when modeling. Tire models are often models of the whole tire (or axi-symmetric models), an exception being the global-local model of [6]. On the other hand, partly modeling the tire requires more difficult boundary conditions, which may be retrieved from a simple whole tire model.

3.1 Discussion

The knowledge that has become available through these studies will be a guideline for this research. In the following the several aspects that will be used for this research are discussed.

In the first place the global-local technique used by Mundl [6] is going to be the basis of this research, although, as discussed before, Mundl’s approach had its deficiencies. Cho found the global-local technique inadequate, at least in the way Mundl performed it. To take into account the essential interaction between the tread and the reinforcements in the tire, the local model can simply be extended. Another lack of the Mundls approach is the uncoupled character, which assumes that the detailed tread pattern doesn’t influence the overall behaviour. In this investigation Mundls global-local technique is improved on these points.

Danielson [7] found cylindrical elements to be the best choice for modeling circular shaped structures, although only when it will result in a significant decrease of the number of elements. These elements therefore might very well be used in the global model. Since the local model will describe the tire geometry in detail, and will thus consist of a lot of elements, no improvement is expected there.

Although Bergström [10] found the Bergström-Boyce model to be the best for describing the constitutive behaviour of rubbers, no parameter values corresponding to a specific rubber were reported. For practical reasons a material model of which the corresponding parameters are known will be used in this model. Since Hofstetter was found to be the only one reporting a tire rubber material model, the Yeoh model, accompanied with the parameters, this model will be used in this research.

Several techniques for modeling the reinforcements efficiently were found in literature. In the Abaqus example tire the Rebar technique is used. Therefore, and because of the ease of application, this method will be used here as well.

Finally several contact relationships are discussed. In Abaqus two main
contact relationships are available: hard and softened contact. Hard contact can be enforced by three constraint enforcement methods: direct, penalty and augmented Lagrange method. However, softened contact can only be enforced by the direct method. The techniques used in literature were the penalty method and the variable constraint method. These two can be related to the adapted hard contact relationship and the hard contact relationship enforced by the penalty method. Since Abaqus offers more than the two options discussed before, various contact algorithms will be used and the most efficient will be chosen. Another part of contact is the friction model. Hofstetter [17] discussed an adapted Coulombic friction model that described rubber contact accurately. However in this project the standard Coulombic friction model will be used, thereby leaving thermal and velocity effects unconsidered.
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Chapter 4

Modeling at different length scales

4.1 Introduction

In this research the tire-road interaction problem is modeled by a multi-scale approach. Herein the characteristic length scales are modeled separately from each other. The finite element software Abaqus Version 6.7-1 is used to model the different levels. The multi-scale approach is schematically depicted in figure 4.1. In this chapter the two levels of the multi-scale approach will be discussed: the global model and the local model. The global model is used for determining the boundary conditions for the local model, which are nodal displacements. The global model must be a model of the whole tire, however, it can be a simple model, only containing the main characteristics of the tire. The local model is used for determining the friction characteristics of the contact region, which will be coupled back to the global model. Therefore the local model can be restricted to the contact region. It should be modeled with sufficient detail to capture all of the aspects that influence those friction characteristics. In particular tread profile and the geometry of the road surface are essential aspects. Thus for the local model a detailed geometry and a fine mesh are required.

First the global model will be discussed, then the local model and in the next chapter the coupling between those will be elaborated.

4.2 The global model

The model of the global simulation is based on the Abaqus example tire model. In the following sections the geometry of the example model, the material models used, and the adaptations made will be discussed.
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Figure 4.1: Schematic view of the coupling between the global and the local model.

Figure 4.2: Axi-symmetric model in Abaqus of one quarter of a tire. Belts, plie, rim nodes are shown. (a) the original model and (b) the adapted model.

4.2.1 The tire geometry

The tire model from the abaqus example problem that is used as a basis has a simple tread pattern. However, it is not desirable to have a tread
pattern in the global model, because it will decrease the modeling flexibility of the local model. Therefore the Abaqus example tire model is adapted to make a treadless tire. Furthermore, the example tire model makes use of 3 noded elements, these are replaced by 4 noded elements for interpolation convenience later on. In figure 4.2, for both the adapted tire and the Abaqus example tire, one quarter of the cross section of the tire is shown. The quarter is discretized with 55 4-noded elements and 49 embedded 2-noded surface elements.

The embedded elements are used to model the reinforcements in the tire. This is done with a rebar layer in surface elements, embedded in continuum elements (host elements). In the Abaqus example problem used, there are three reinforcements: two belt sectors and a carcass sector. The embedded elements method constrains the embedded element to the host element. The geometric rebar layer properties are given by the cross-sectional area of each rebar ($A$), the spacing in the plane of the element ($s$) and the initial angular orientation $\phi$ in degrees measured to the local 1-direction are defined. Now the rebar layer is a plane with an equivalent thickness of $t = A/s$.

A three dimensional tire model is then generated by revolving the axisymmetric quarter tire model about the rotational symmetry axis. This is a symmetric partial three dimensional tire model, further referred to as the half-tire model. The half-tire model consists of 360 sectors of 8-noded linear elements covering a range of 360 degrees. 360 sectors are chosen such that no cylindrical elements are needed, which makes linear interpolation sufficient, and the boundary between two global elements corresponds with the boundary of the local model.

Because it is assumed that the tire is symmetric in the xz-plane, it is sufficient to model only half the tire. The boundary conditions enforcing this assumption are discussed in section 7.2. The rim is represented by several nodes constrained to a fixed node on the position of the axis of rotation. For the axisymmetric model this are three nodes, highlighted in figure 4.2. The global model is shown in figure 4.3.

### 4.2.2 The tire material

A tire is a complex structure. It consists of several layers of highly anisotropic stiff cords embedded in a matrix of low stiffness rubber. In the example tire model the rubber is modeled as a simple hyperelastic material. In literature a more sophisticated model is found, used by Hofstetter [17], including the parameters describing a tire rubber material. This is the Yeoh material model, which is given by:

$$W(I_1) = C_{10}(I_1 - 3) + C_{20}(I_1 - 3)^2 + C_{30}(I_1 - 3)^3.$$  \hspace{1cm} (4.1)

Herein $W$ is the strain energy and $I_1$ the first invariant of the left Cauchy-Green strain tensor. The values for the parameters are given in table 4.1.
Figure 4.3: The global model: three dimensional half-tire model.

[17]. The belts and the carcass are modeled by a linear elastic material model, of which the Young’s moduli and poisson ratios are given in table 4.1. The geometrical parameters $A$, $s$ and $\phi$ are given in table 4.1 as well. In figure 4.4 the engineering stress $t$ is plotted versus the engineering strain $\epsilon$ for a uniaxial tensile case. The Yeoh model will give inaccurate results for strains of more than 3, however, the strains in the tire analysis are limited to about 0.3.

4.3 The local model

As stated before, the contact region is the most important part when considering tire-road interaction. The contact region contains the parts of the tire and road that are in direct contact. To model the interaction correctly, this contact region should be modeled accurately and thus detailed. The small geometric characteristics of both tire and road surface are essential
Table 4.1: Parameter values of the Yeoh material model for tire rubber [18, 17].

<table>
<thead>
<tr>
<th>coefficient</th>
<th>value (N/mm$^2$)</th>
</tr>
</thead>
<tbody>
<tr>
<td>$C_{10}$</td>
<td>0.3294 MPa</td>
</tr>
<tr>
<td>$C_{20}$</td>
<td>0.0232 MPa</td>
</tr>
<tr>
<td>$C_{30}$</td>
<td>-0.0003 MPa</td>
</tr>
<tr>
<td>$E_{\text{cass}}$</td>
<td>9.87 GPa</td>
</tr>
<tr>
<td>$E_{\text{bels}}$</td>
<td>172.2 GPa</td>
</tr>
<tr>
<td>$\nu_{\text{cass}}$</td>
<td>0.3</td>
</tr>
<tr>
<td>$\nu_{\text{bels}}$</td>
<td>0.3</td>
</tr>
<tr>
<td>$A$</td>
<td>0.42 m$^2$</td>
</tr>
<tr>
<td>$s$</td>
<td>0.001 m</td>
</tr>
<tr>
<td>$\phi_{\text{cass}}$</td>
<td>0.0°</td>
</tr>
<tr>
<td>$\phi_{\text{bels}}$</td>
<td>70°</td>
</tr>
<tr>
<td>$\phi_{\text{bels}}$</td>
<td>110°</td>
</tr>
</tbody>
</table>

Figure 4.4: Stress strain curve of the Yeoh material model used.

for modeling this interaction. The contact region can be divided in two parts, the contact patch and the road surface. In the following sections, the modeling of the these two are elaborated.

4.3.1 The detailed contact patch

The detailed contact patch model is the tire part of the local model. There are two conditions that should be satisfied. In the first place the model should represent the contact patch of a realistic car tire. Secondly the model should be compatible with the global model. Since the objective of this project is to develop a multi-scale procedure for simulating tire-road
interaction, a simplified tire geometry is used. This simplified model will still be having at least the characteristics of a real car tire; the tread surface is modeled, but only characteristic measures are captured, such as the length and width of the tread blocks and grooves of a real passenger car tire. There are many types of tires with great differences in tread pattern, but realistic values are 30 to 60 mm for the length and width of the tread blocks, 5 to 8 mm for the groove depth and width. The angle and width of the contact patch depend on tire type, inflation pressure and tire load. But in general, one can assume a tire is in contact with the road along the whole width, and about 150 mm in length. Considering a tire circumference of 1.8 m will result in an angle of about 40 degrees. The local tire model will only differ in geometry from the global model. The material and the reinforcements will be modeled the same way as described in section 4.2.

Figure 4.5 shows the contact patch model. The faces where the boundary conditions apply, are highlighted in red. The tread pattern can be anything, but according to the second condition the overall geometry has to be compatible with the global model. That means that at the boundaries where the boundary conditions will be applied all nodes must lie within the outer geometry of the global model. If this is not the case, the boundary conditions can not be interpolated anymore. Within the outer geometry some deviation is allowed, since this will be corrected by a stiffness homogenization. In figure 4.6, the outer geometry of the boundary region is highlighted.
4.3.2 The detailed road surface

Besides the profile of the tire, the interaction between a tire and a road surface will be affected by the surface texture of the road as well. Therefore, geometrical details of the road surface at the length scale of the details of the tire profile are incorporated into the model. The model will be based on an actual piece of road surface. First this piece of road will be described. Then the modeling will be elaborated. In this research project, the road surface is considered to be much stiffer than the tire and is therefore modeled as rigid.

The road surface used in this simulation is made of a special type ZOK\textsuperscript{1}. This is a variant on the mainly used open graded asphalt concrete (OGAC). In ZOK, instead of bitumen a synthetic material is used as a binder. ZOK has the same advantages as OGAC concerning draining water and noise reduction, but it is much stronger. The pebbles and stone chips have a characteristic size of up to 16 mm. The choice for this type of pavement is based on availability. A piece of 71.6 x 72.2 mm\textsuperscript{2} has been scanned by an optical imaging profiler (Sensofar) with a resolution of 1270 x 1280. In figure 4.7 this block is shown. The size of this block has been chosen based on the capabilities of the profiler. The total contact is larger than this block, therefore the scan will be copied four times and stitched together to form the whole contact patch.

The high amount of deep and steep holes in the surface limits the applicability of the profiler. A restriction of the Sensofar is that when the beam of light hits a slope greater than 60 degrees no measurement can be obtained. When too many measurements have failed stitching neighbouring regions will not be possible, as is the case for the road surface shown in figure 4.7. To solve this problem of the deep holes in the road surface, an impression of the piece of road is made in clay of which subsequently a height scan is made. Since the holes are small, a car tire will only sink a limited distance into those holes. The total depth of those holes is therefore of no importance, which validates the use of the clay impression. The deep holes are now represented in this negative print by small peaks. In figure 4.8 the scanned clay print is shown.

\textsuperscript{1}Dutch for: zeer open kunstof
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Figure 4.6: Cross section of global model. The red line outlines the outer geometry.

Figure 4.7: Road surface block used for scanning.

Figure 4.8: Clay print of road surface scanned by the Sensofar.
Chapter 5

Coupling of the global and local modeling

5.1 Introduction

In the previous chapter the different length scales of the multi-scale approach have been discussed. The models at the two length scales have to be coupled. The boundary conditions of the local model, which are prescribed displacements of the nodes of the coupling region, are obtained from the global model. The coupling region is the region where the contact patch is 'cut out' of the global model. The feedback to the global model consists of two aspects. In the first place the friction characteristics that correspond to and are supplied by the local model and secondly, the stiffness adaption of the global model on the basis of the strain energy ratio. In the following sections these boundary conditions and the feedback procedure are further discussed.

5.2 Boundary conditions for the local model

In figure 5.1 the contact patch is schematically represented. The coupling region, on which the boundary conditions will be placed, is represented by the four areas that are distinguished. The displacements of these areas have to be the same for the global and local simulation. Therefore the following condition must hold:

\[ \vec{u}_{A_i,l}(\vec{x}) = \vec{u}_{A_i,g}(\vec{x}), \quad i = 1, 2, 3, 4 \]  

(5.1)

Herein is \( \vec{u}(\vec{x}) \) the displacement of a point \( \vec{x} \) located in area \( A_i \). \( A_{i,l} \) and \( A_{i,g} \) are the corresponding areas of respectively the local and global model.

Because of the discretization and the mesh difference between the local and global model, interpolation is necessary. The interpolation is linear
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and done on the basis of shape functions. Therefore first an isoparametric element is defined with a normalized coordinate vector $\xi$. In figure 5.2(a) and 5.2(b) the normal and isoparametric element are shown, respectively. The shape functions then have a standard form. For a bilinear quadrilateral master element they can be written as:

$$
N^e(\xi) = \begin{bmatrix}
\frac{1}{4}(1 - \xi_1)(1 - \xi_2) \\
\frac{1}{4}(1 + \xi_1)(1 - \xi_2) \\
\frac{1}{4}(1 + \xi_1)(1 + \xi_2) \\
\frac{1}{4}(1 - \xi_1)(1 + \xi_2)
\end{bmatrix}.
$$

(5.2)

However, to interpolate, now the normalized coordinate $\xi$ should be known. The physical coordinate $\vec{x}$ can be mapped on the normalized coordinate $\vec{\xi}$ by the same shape functions:

$$
\vec{x}(\vec{\xi}) = N^{eT}(\vec{\xi})\vec{x}^e.
$$

(5.3)

Where $\vec{x}^e$ contains the coordinates of the nodes related to a specific element. The global isoparametric coordinates of a given point $\vec{x}$ corresponding to a local node, are determined from this equation by the method of Hua [20].

Finally, the nodal displacement $\vec{u}_l$ of a local node can be interpolated via

$$
\vec{u}_l = N^{eT}(\vec{\xi})\vec{u}_g^e.
$$

(5.4)

Herein, $\vec{u}_g^e$ are the displacements of the global nodes.
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![Figure 5.2: (a) the regular element and (b) the isoparametric element.](image)

5.3 Feedback to the global model

In the previous section the coupling from the global to the local simulation is elaborated. In this section the feedback of the local simulation to the global simulation is discussed. Two aspects are highly influenced by the geometric difference between the global model and the local model: the friction characteristics and the stiffness of the tire. First the aspect of friction characteristics will be discussed.

5.3.1 Friction characteristics

The friction characteristics have to be adapted in the global model in order to reflect those of the local model. There are certain requirements on the friction characteristics. First of all, the total friction force acting on the local model must equal the total friction force in the global model:

$$\int_{A_g} \vec{p}^s dA_g = \int_{A_l} \vec{p}^s dA_l,$$

where $A_g$ and $A_l$ are the total contact area for the global and local simulation, respectively, and $\vec{p}^s$ the shear stress vector, defined in the plane of the road surface in the global model. Secondly, not only the total friction force should be equal, the pattern of the friction forces, induced by length scales of the global model, should be reflected as well. That means that for a smaller area $a_i$ that same condition should hold:

$$\int_{a_{g,i}} \vec{p}^s da_{g,i} = \int_{a_{l,i}} \vec{p}^s da_{l,i},$$
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with,

\[
\sum_{i=1}^{n} a_{g,i} = A_g \quad \text{and} \quad \sum_{i=1}^{n} a_{l,i} = A_l.
\] (5.7)

Herein \( n \) is the number of small areas that together make up the total contact area.

Equations (5.5), (5.6) and (5.7) are rewritten to

\[
\sum_{i=1}^{n} \vec{F}_{s,g,i} = \sum_{j=1}^{k} \vec{F}_{s,l,j}
\] (5.8)

\[
\int_{a_{g,i}} \vec{p} \, da_{g,i} = \vec{F}_{s,g,i}
\] (5.9)

\[
\int_{a_{l,i}} \vec{p} \, da_{l,i} = \vec{F}_{s,l,j}.
\] (5.10)

Herein, \( n \) is the number of global nodes (which equals the number of small areas), \( k \) represents the number of local nodes and \( a_{g,i} \) is the area that contains all points that lie closest to a certain global node \( i \). In figure 5.3 this is shown. So each area \( a_{g,i} \) is chosen such, that it represents a single node of the global simulation. Now the second condition is rewritten to

\[
\vec{F}_{s,g,i} = \sum_{j=1}^{m_j} \vec{F}_{s,l,j,i}, \quad i = 1, 2, ..., n,
\] (5.11)

Figure 5.3: In grey the area containing all points that lie closest to node \( i \).

![Diagram showing the relationship between global and local nodes and the area represented by each node.](image_url)
with \( m_j \) the number of local nodes lying within area \( a_{g,i} \). Combining equation (5.8) and (5.11) results in

\[
\sum_{j=1}^{n} \left( \sum_{i=1}^{m_j} F_{1,i}^{a_{g,i}} \right) = \sum_{i=1}^{k} F_{1,i}^{a_{g,i}},
\]

with

\[
\sum_{j=1}^{n} m_j = k.
\]

Since there is no overlap between the small areas and the total global area covers the total local area, equation (5.12) is satisfied.

The internal friction forces are summed for all the local nodes that lie closest to this global node and then converted to a friction shear stress corresponding to an area \( a_{g,i}^* \), which is in general not equal to \( a_{g,i} \). Both areas describe all the points that lie closest to a specific global node. However, the difference is that \( a_{g,i}^* \) describes the points that lie closest in an isoparametric as opposed to the current representation. All \( a_{g,i}^* \) don’t overlap and do sum up to \( A_{g} \), which validates its use. Eventually the shear stress is the output of the usersubroutine:

\[
P_{g}^{s,i} = \frac{F_{g,i}^{a_{g,i}^*}}{a_{g,i}^*}, \quad i = 1, 2, ..., n.
\]

One additional remark has to be made. Since it is possible that a local node lies exactly at an equal distance from two (or more) different global nodes, another criterium is used as well. The current slip can be used to determine to what global node the local node will probably lie closer in the next increment. In figure 5.4 this is illustrated. Global node 1 and node 2 lie at the same distance from the local node. The slip direction \( \vec{s} \) of the local node, however, suggests that in the next increment the local node will lie at position \( x \). Now two vectors (\( \vec{v}_1 \) and \( \vec{v}_2 \)) can be determined. The friction force at the local node will apply to the global node with the smallest vector, in this case node 1. Again it may happen that both vectors are equal of length. In that case at both nodes half of the friction force will apply.

### 5.3.2 Stiffness adaption

Besides the friction forces, also the stiffness has to be adapted to correspond with the local model. In figure 5.5(a) and 5.5(b) the contact patch is shown for both models. The global and local model differ geometrically from each
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Figure 5.4: Schematic view of using the current slip for determining the closest global node.

other. This results in a difference in stiffness of the two models. To obtain an averaged equal behavior of both models, the global model’s material properties are adapted.

Figure 5.5: Geometric difference of (a) the global and (b) the local model.

To determine the stiffness difference, the averaged total elastic strain energy of the elements is chosen as a quantitative measurement. The material model will now be adapted such that this averaged total elastic strain energy of the global model equals that of the local model, when both models are deformed according to the same boundary conditions. This adaption is best illustrated with the simple Neo-hookean material model, for which

\[ W = C(I_1 - 3). \]  

(5.16)

Herein, \( C \) is the material parameter, \( I_1 \) the first invariant of the left Cauchy Green strain tensor and \( W \) is the strain energy. Then \( C \) is scaled such that
the global strain energy equals the local strain energy

\[ C_{\text{global}} = \frac{W_{\text{local}}}{W_{\text{global}}} C_{\text{global}}^*, \]  

(5.17)

with \( W_{\text{global}}^* \) the unscaled global strain energy and \( C_{\text{global}}^* \) the unscaled global material parameter. This is for a simple material model. When a more complex material model is used, the basic strategy remains the same.

This approach makes the adaption of the material model dependent on the deformation of both global and local model. To avoid adapting the material model every increment, there is chosen for adapting the material model only once for a deformation that describes the average deformation of the tire best. That is the deformation of the tire when it is fully loaded (see section 7.2).

5.4 Implementation

In figure 5.6 a realization of the coupling between the global and local simulation is is schematically shown. This loop is on an incremental basis, where

![Simulation Loop Diagram](image)

Figure 5.6: Schematic view of the simulation loop

the converged solution of the global model provides the input for an increment of the local model. In an ideal case the local simulation is run every iteration, thereby converging the friction characteristics to an equilibrium as well. However, then the current iterative nodal displacements (as boundary conditions for the local simulation) are required every iteration as well. With Abaqus it appeared to be impossible to obtain these current iterative nodal displacements. The incremental approach implies, however, that the
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friction characteristics of the previous increment do apply at the current increment. To make sure that is the case, the increments should be taken sufficiently small. Furthermore, no friction information is available during the first increment. Since the influence of the first increment is negligible when many increments are used, in the first increment a frictionless contact is assumed.

The loop from figure 5.6 starts with the initiation of the global simulation. Every new increment of the global simulation the friction stresses are requested via subroutine FRIC. This subroutine extracts the necessary data for calculating the friction stresses (equation (5.15)) from the output file of the local simulation. Before the data file can be read it has to be opened by subroutine UEXTERNALDB. Every increment of the global simulation this subroutine runs a local simulation as well. The results of the local simulation are used in the next increment of the global simulation. When an increment of the global simulation has converged, the nodal displacements are written to an output file. These are extracted by the subroutine DISP. Again, first the subroutine UEXTERNALDB is needed to open the data file. The extracted data is used to calculate the correct nodal displacements (equation (5.4)), which represent the boundary conditions necessary for the local simulation. Figure 5.7 shows the coupling of the global and local simulation via subroutines.

Figure 5.7: Schematic view of the subroutine coupling.
Chapter 6

Validation of the coupling procedure

6.1 Introduction

In the previous chapter, a multi-scale coupling approach is presented for the purpose of simulating tire-road interaction. Before the coupling procedure is applied in a tire-road simulation, it will first be evaluated for relatively simple test cases. Since there is no good validation data for the tire available yet, these test cases make it possible to see whether the coupling procedure is performing adequately. Three different test cases will be considered. In all three cases, a rubber block of 20 x 20 x 20 cm will be pressed against a rigid surface. The rubber will be modeled with the Neo-hookean material model, with $C_{10}$ is 1:

$$W = C_{10}(I_1 - 3)$$

(6.1)

Herein, $W$ is the strain energy and $I_1$ is the first invariant of the left Cauchy Green strain tensor. In each consecutive test case, some more detail is taken into account. The Coulombic friction model is used in the first two test cases and frictionless contact is assumed in the third test case. Just as for the situation of tire-road interaction, the test cases will consist of a global model and a local model. The global model is used to obtain the boundary conditions for the local model and the local model returns the friction characteristics to the global model. In the first test case, the global model and the local model are meshed with the same discretization and the surface is modeled as flat. In the second test case the surface is again modeled as flat, but for the local model a finer mesh is used than for the global model. In the third test case, an uneven surface is used and the local model is meshed with smaller elements than the global model. The test cases will be compared to direct simulations, which are simulations without a coupling. In these reference simulations a rubber block is pressed against the surface and the friction characteristics are determined within this simulation itself. In the first two
Multi-scale approach to modeling tire road interaction

test cases the von Mises stress will be used for validation. Because of the rough surface, the normal contact pressure will fluctuate more in test case 3 than in the other cases. The local model doesn’t communicate his normal contact pressure back to the global simulation. Since von Mises stresses are strongly affected by these normal forces, this quantity is not suitable for validation purposes. Therefore two other validation criteria are used in test case 3 as well: the friction force acting on the contact surface and the displacement spectrum of the floor plane.

6.2 Test case 1

In the first test case, the global model is a cube discretised with 4x4x4 8-noded elements. It is pressed against a flat rigid surface. In figure 6.1(a) the undeformed global model is shown with the applied boundary conditions. The rubber block is pressed against the surface by a displacement of 1% of the height on the top face. A quarter piece of a larger block is modelled by assuming symmetry conditions on two faces. The local model represents the lower part of the global model’s cube. It is meshed with the same amount of detail, i.e. 4x4x2 elements. This model is shown in figure 6.1(b). The boundary conditions of this local model are the displacements of the top layer nodes and are derived from the global model.

![Figure 6.1: The undeformed (a) global and (b) local model for test case 1 with their mesh and boundary conditions.](image)

Because the global and local model have the same amount of detail, for this test case the results of the coupled model and the direct model should be exactly the same, at least when sufficiently small increments are used. However, because the same mesh discretization is used, no homogenization
of the friction characteristics is needed in this model. The homogenization procedure can therefore not be validated with this test case. In figure 6.2(a) the result of the direct simulation is shown. The results obtained with the coupled simulation are expected to equal this response. In figure 6.2(b) and 6.2(c) the stress fields obtained with the global and local simulations are shown respectively. Both the global and local simulations show a good correspondence with the direct simulation. Even better results are obtained when more increments are used. This is shown in figure 6.3. Herein the relative error of the local and the global model with respect to the direct simulation is plotted against the z-coordinate along the edge of the cube. This edge is highlighted in figure 6.1(a) and 6.1(b).

Figure 6.2: Von Mises stress fields (Pa) of the (a) direct simulation, (b) global simulation and (c) local simulation of test case 1.
6.3 Test case 2

In the second test case a finer discretization than for the global model is used for the local simulation. Therefore the interpolation procedure (eq. 5.4) and homogenization of the shear stresses (eq. 5.15) are necessary and can be tested. The global simulation is modeled the same as in test case 1. The local simulation is modeled with a mesh twice as fine, i.e. 8x8x4 elements. No further changes are made with respect to test case 1, although here two direct simulations are used for comparison: one simulation with a coarse mesh like the global model and one with a detailed mesh like the local model. In figure 6.4(a), 6.4(b), 6.4(c) and 6.4(d), the coarse direct, detailed direct, the global and the local simulation are shown, respectively. First must be mentioned that there is already a difference between the coarse direct and the detailed direct simulation. The detailed direct simulation shows higher stresses in the contact zone. The global simulation shows a better correspondence to the detailed direct simulation than the coarse direct. The local simulation shows higher stresses in the contact zone. Since this overestimation is not seen in test case 1, it is probably caused by the friction homogenization procedure.

6.4 Test case 3

In the third test case, for the local simulation, besides a finer mesh also a rough surface is used as opposed to a flat surface. This rough surface is chosen such that it will generate only friction forces in one direction. In
Figure 6.4: Von Mises stress fields (Pa) for test case 2 (a) the simple direct simulation, (b) the detailed direct simulation, (c) the global simulation and (d) the local simulation.

In figure 6.5 the surface is schematically depicted. As opposed to the former two cases the contact between the block and the surfaces is assumed frictionless. Moreover, no symmetric boundary conditions will be applied here. In figure 6.6 the undeformed models with their boundary conditions are shown.

In this test case the influence of the geometry in the local model on the global model can be validated. The surface has one side that will generate no friction forces and another side that will only generate friction forces in the $z$-direction. This will result in a total resultant friction force acting on the bottom surface of the block. In figure 6.7 for several numbers of increments the friction force acting on the bottom surface is given for the local, global and direct simulation. Striking is that the local simulation is deviating from the direct simulation regardless of the number of increments used. This overestimation of the friction forces in the local model has already
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![Schematic representation of the flat-rough surface.](image)

Figure 6.5: Schematic representation of the flat-rough surface.

been seen in test case 2. For an increasing number of increments the friction force of the global simulation is converging to that of the local simulation, because then the effect of the incremental coupling is less important.

In figure 6.8 the displacements in $z$-direction of the floor plane are given. The $z$-direction is the direction where friction forces are expected due to the surface geometry. As expected, all three simulations give relatively little displacement in the region where the surface is rough and relatively large displacement in the smooth region. The direct simulation, however, shows a significant larger displacement in the smooth region than the global and local simulation. The larger friction force acting on the global and local simulation already seen in figure 6.7 explains this.
Figure 6.6: The undeformed (a) global and (b) local model for test case 3 with their mesh and boundary conditions.
Figure 6.7: Relative error of the local and the global model to the direct simulation.
Figure 6.8: The displacement field (cm) of (a) the direct simulation, (b) the global simulation and (c) local simulation.
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Chapter 7

Results and Discussion

7.1 Introduction

In this chapter the validated multi-scale procedure will be applied to the tire model discussed in chapter 4. However, the local model discussed in that chapter describes a road surface with such a high amount of detail that convergence problems occur. Therefore the road surface is chosen to be described with a more simple model, which is shown in figure 7.1. This model is based on the height scan as well, only the resolution is smaller. The reference height is defined by leveling the highest point of this surface to global road surface level, since then no contact at local level will occur before contact at global level occurs.

![Simplified road surface](image)

Figure 7.1: Simplified road surface.

Before the simulation with a global-local coupling is conducted, the stiffness homogenization (see section 5.3.2) is performed. For this purpose the global and local models are used in an uncoupled as opposed to a coupled
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simulation, because of time efficiency. That is, the boundary conditions of the local model are still determined by the global model, however, the friction forces are not communicated back to the global model. Otherwise the same conditions are used. In the following, first the loading conditions, then the stiffness homogenization and finally the results of the tire-road interaction simulation will be discussed.

7.2 Loading conditions

In figure 7.2 and 7.3 schematic representations are shown for the global and local model, respectively. The global simulation is divided in two steps. During the first step, the tire is inflated. For this purpose a pressure of \( p|_{\Gamma_{1,2,3}} = 0.2 \text{ MPa} \) is applied to the inside of the tire. This pressure is applied in ten increments following a linear ramp function. In the following steps it is hold constant and is not influenced by the deformation of the tire due to any other subsequent loads. The center rim node, which controls all other rim nodes, is fixed in all directions and rotations, and at the central \( xz \)-plane of the tire symmetry conditions apply: \( u_y|_{\Gamma_s} = 0 \). During the second loading step, the tire is pressed against the road surface. For this purpose a displacement of -0.02 m in the \( z \)-direction is applied to the rim node in one hundred increments following a linear ramp function. The internal pressure and symmetry conditions remain active.

For the local model, at \( \Gamma_{1,2,3} \) the nodal displacements are prescribed and as in the global model, a pressure is applied to the inside: \( p|_{\Gamma_{1,2,3}} = 0.2 \text{ MPa} \). For the local model, a frictionless contact between the road and the tire is assumed. The friction forces acting on the local model and transferred to the global model are therefore merely induced by the local model’s road surface geometry.

7.3 Stiffness homogenization

The results of the uncoupled simulations are shown in figure 7.4 and 7.5. For the stiffness homogenization (see section 5.3.2) the ratio \( R \) of the total elastic strain energies of the global and local model is used to adapt the material parameters of the global model:

\[
W_g = \sum W_{el,g} \tag{7.1}
\]

\[
W_l = \sum W_{el,l} \tag{7.2}
\]

\[
R = \frac{W_l}{W_g} \tag{7.3}
\]

Herein, \( W_{el} \) is the strain energy in one element, \( W_g \) and \( W_l \) are the total strain energies in the global or local contact patch, respectively. The geomet-
Figure 7.2: Schematic cross-section of a tire showing the areas where the boundary conditions apply.

Figure 7.3: Schematic cross-section of a tire showing the areas where the boundary conditions apply.

Rirical difference between the global and local model is only apparent in the rubber zone, whereas the reinforcements of the global and the local model are identical. Therefore only the rubber is considered in this homogenization. In this uncoupled simulation $R = 0.3475 \, [-]$. The material parameters of the rubber in the global simulation are multiplied by this ratio. In table 7.1, the new material parameters are shown. To check whether this stiffness homogenization has the desired effect, the total reaction force is considered. In table 7.2 for the global model with the unscaled material parameters, the global model with the scaled material parameters and the local model the reaction forces are given. First must be noted that the reaction force of the local model will be necessarily lower, because the average height of its road surface lies about 1 mm lower. Nevertheless it is striking that the reaction force in the scaled global model is nearly as high as in the unscaled global model. This indicates that the stiffness of the contact patch is significantly
Table 7.1: The parameter values of the Yeoh material model for tire rubber after scaling.

<table>
<thead>
<tr>
<th>coefficient</th>
<th>value (N/mm²)</th>
</tr>
</thead>
<tbody>
<tr>
<td>$C_{10,\text{new}}$</td>
<td>0.1145 MPa</td>
</tr>
<tr>
<td>$C_{20,\text{new}}$</td>
<td>0.0081 MPa</td>
</tr>
<tr>
<td>$C_{30,\text{new}}$</td>
<td>−0.0001 MPa</td>
</tr>
</tbody>
</table>

influenced by the stiffness of the reinforcements, and thus scaling the rubber part only is insufficient.

Table 7.2: The reaction forces acting on the center rim node for the different models.

<table>
<thead>
<tr>
<th>model</th>
<th>reaction force (N)</th>
</tr>
</thead>
<tbody>
<tr>
<td>unscaled global model</td>
<td>1145 N</td>
</tr>
<tr>
<td>scaled global model</td>
<td>1055 N</td>
</tr>
<tr>
<td>local model</td>
<td>687 N</td>
</tr>
</tbody>
</table>

7.4 Tire inflation

After scaling of the rubber stiffness, the first loadcase applied is the inflation of the tire with a pressure of 0.2 MPa. In this simulation no contact occurs, therefore the global-local coupling is not yet of importance. In figure 7.6 the result of the simulation of the inflated global model and of the deflated tire model are shown. Only a small deformation due to the pressure is seen, mostly in the side wall of the tire. In figure 7.7 is zoomed in on the tire at the symmetry plane. A clear shearing can be seen at the inner ring of elements. This shearing doesn’t occur when a-symmetric boundary conditions (not ideal for other reasons) are applied or when the two belt reinforcements are removed. This indicates that these belt reinforcements are responsible for the a-symmetry and that the prescribed symmetry conditions are not ideal in this case.

7.5 Coupled simulation

The shearing that is discussed in the previous section causes the simulation not to converge. Therefore the coupling is applied to the global model with the material parameters that are not corrected for stiffness, since in that case the shearing is less apparent. In figure 7.8 the results are shown for the footprint of the tire for the global and local simulation. In the global simulation several regions can be distinguished that show high or low stresses. The uncoupled global footprint shown in 7.4 shows an almost
Figure 7.4: The Von Mises stress field (Pa) of the uncoupled global model, with (a) the full tire and (b) the footprint.

symmetric stress field. The stress pattern in the coupled global simulation is the result of the feedback of the local friction characteristics. The global model’s stress pattern can be recognized in the local model’s footprint. High contact stresses in the local simulation suggest that high friction forces will occur. In the global simulation high stresses are found at the locations where in the local simulations high contact stress occur and low stresses are found where for instance a groove is located (and thus no contact stresses occur) in the local model.
Figure 7.5: Von Mises stress fields (Pa) of the uncoupled local model, with (a) the full contact patch and (b) the footprint.

Figure 7.6: A quarter of the cross-section of the geometry of the deflated (in green) and the inflated (in blue) global model.
Figure 7.7: Shearing of inner ring of elements after the inflation step.
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Figure 7.8: (a) the Von Mises stress field (Pa) for the footprint of the coupled global model, (b) the Von Mises stress fields (Pa) of the footprint of the coupled local model and (c) the contact stress (Pa) of the footprint of the coupled local model.
Chapter 8

Conclusion and recommendations

8.1 Conclusions

A multi-scale model has been developed in Abaqus v6.7-1. It consists of two coupled models describing two characteristic length scales of a car tire-road surface interaction. In the global model, the larger length scale is represented, which captures the whole tire of a passenger car and general geometry of the road surface. In the local model the smaller length scale is represented, which captures the contact patch with tread pattern and the detailed geometry of the road surface including the small stones and pebbles.

The local model, which is capable of describing the friction characteristics accurately due to its detailed modeling, delivers the friction characteristics to the global model. The global model uses a subroutine to calculate the friction stresses from the data of the local model. The local model uses a subroutine to calculate its boundary conditions, which are nodal displacements of the coupling plane. This is done with use of the global model’s nodal displacements. Because of the mesh difference between the global and the local model, for the calculation of the nodal displacements of the local model, a linear interpolation procedure is used and for the calculation of the friction stresses of the global model, a friction force homogenization procedure is used. In this homogenization procedure the total friction force acting on the local model is transferred to the global model thereby preserving the friction force pattern as well as possible. Moreover, a stiffness homogenization is used to make up for the stiffness difference between the global and local model due to the geometrical difference. In this procedure, the material parameters of the global model are scaled such that the global strain energy equals the local strain energy. The multi-scale coupling procedure is on an incremental basis, because with Abaqus it appeared to be impossible to obtain the current iterative nodal displacements. Because of
this incremental coupling the increments have to be taken sufficiently small.

One type of rubber is considered for both the global and local model and is described by the Yeoh model. Viscoelasticity and temperature dependence are not captured. At both levels, a carcass and two belts are considered as reinforcements, which are modeled by embedded rebar surface elements.

The coupling procedure is validated with three test cases. Each subsequent test case is taking into account a little more detail. The first test case validates the coupling procedure in general. With increasing increments, the simulation shows increasing accuracy. With this test case is shown that the basics of this coupling procedure are working correctly. The second test case validates the interpolation procedure and homogenization of the friction stresses. The local simulation shows an overestimation of the stresses in the contact zone. The global simulation shows a more accurate result than the coarse direct simulation, therefore the higher detail of the local simulation has improved the friction characteristics of the global model. The third test case validates the influence of the local model’s geometry. The friction force acting on the local model is deviating from the direct simulation regardless of the number of increments used. Since this error has already been seen in test case 2, it is most likely that it is introduced by the friction homogenization procedure. However, the friction force of global simulation converges to the friction force of the local simulation for increasing increments, which confirms that the homogenization procedure of the friction forces is technically working correctly.

Finally the coupling procedure is applied at the car tire model. For determining the stiffness scaling parameter an uncoupled simulation is run, because of time efficiency. Scaling only the rubber material parameters to its strain energy ratio seemed insufficient, therefore it is required to scale the reinforcements material parameters as well. Furthermore, an unexpected shearing of the inner ring of elements is noted, which implicates that the applied symmetry conditions are not ideal. The reinforcing structures seem to make the model asymmetric. This deficiency is highly representing itself when the stiffness adaption has been made. This way no converging solution can be achieved for the global model that is corrected for stiffness. The results of the multi-scale coupling applied to the global model that is not corrected for stiffness show that friction characteristics of the contact plane in the local simulation can be recognized in the global model.

8.2 Recommendations

In this report, a multi-scale approach is presented, which is applied in a tire-road interaction simulation. A number of modifications are recommended for further improvement of the method:

- The global model is now described by 360 sectors in circumferential di-
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...ction. This number can be reduced while still describing the global characteristics. However, describing a circular structure accurately with few elements, requires the use of cylindrical elements. Cylindrical elements use linear interpolation in radial direction and trigonometric interpolation along circumferential direction. The interpolation procedure should therefore be adapted at that point.

• The local model is at this point still a relatively simple model. The geometry of the contact patch can be improved to reflect a more realistic tire profile. The reinforcements are modeled by embedded rebar surface elements, however, individual fibers and their interaction with the surrounding matrix could be modeled.

• The choice for the Yeoh material model, is based on availability of tire rubber model data. For both the global and local model one type of rubber is described. When tensile test data is acquired, it can be used to fit the Bergström-Boyce model on multiple rubber types and capture viscoelasticity and temperature dependence.

• The coupling procedure does only feed back the friction characteristics. It would be interesting to investigate the feed back of the normal forces as well. However, then the subroutine FRIC will not be useful anymore. Subroutine UTRACLOAD might be a feasible replacement.

• The coupling procedure is now limited to a global model with a flat road surface. It could be extended such that it can handle non flat surfaces as well.

• The model described in this report is limited to simulations of rotating the tire over a certain angle, because of the measurements of the local model. To make a continuous model the local model could be regenerated after every increment.

• The ALE (Arbitrary Lagrangian Eulerian) method is often used in tire analysis. A drawback of this method is that the model should be continuous in circumferential direction and therefore no detailed tread pattern is possible. If the ALE method is combined with the multi-scale approach, this drawback is eliminated. For this purpose the material streamlines of the global model should be used for the coupling to the local model instead of the nodal displacements.
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Bibliography


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[18] Abaqus version 6.6 documentation
