Prediction of tyre/road contact stress distributions

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Prediction of tyre/road contact stress distributions

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DCT report

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Eindhoven, August, 2007
Summary

In this report the work carried out at the Dynamics & Control group of the Eindhoven University of Technology within the LOT project is described. This research can be divided in three main subjects: dynamic analysis of the test set-up at STUVAtec, investigation of the applicability of Fuji Prescale films to tyre/road contact measurements and Finite Element (FE) modelling of the tyre/road contact.

The goal of the dynamic analysis of the STUVA set-up is to evaluate the influence of the unevenness of the track layout on the contact force and to give recommendations for the optimal track layout that will minimise the contact force variations. This analysis has shown that the main source of force variations is the change in height along the track and that a wavy pattern with two waves in the circumference (2 maxima and 2 minima) should be avoided. As a result additional measures have been taken to ensure an almost flat track profile.

Experimentally determined contact force information is needed in order to validate the FE model of the tyre/road contact. There is a database of measured contact forces available for the LOT project (TireView). However the comparison between the measured data from TireView and the forces calculated in the FE model is not straightforward. Therefore a number of experiments has been performed with Fuji Prescale films in order to determine the contact pressure distribution between a tyre and asphalt. The results of these experiments are promising, but further research is needed before the contact pressure distribution measured in this way can be used for validation purposes.

The tyre/road contact has been modelled by assuming that the size of the contact patch is known and that the belt and the road are infinitely stiff. Since the goal of the model is to predict the contact forces between a tread block in the tyre profile and the road, a single tread block has been modelled in the FE package ABAQUS. The boundary conditions applied to the block are such that the passage of the block along the contact patch is simulated and the corresponding forces are calculated. Two FE models have been developed: a simplified 2D model and a full 3D model. Results from the 2D model show that the effect of the transversal deformation on the longitudinal forces cannot be neglected, since the longitudinal forces are overestimated with the 2D model. The results of the 3D model show a good qualitative and quantitative correlation with the contact forces from the TireView database. This model has been used to predict the contact forces at a travelling velocity of 80 km/h.
1 Introduction

In this report the work carried out at the Dynamics & Control group of the Department of Mechanical Engineering of the Eindhoven University of Technology within the LOT (Lifetime Optimisation Tool) project is described. The aim of the LOT project is to develop a lifetime prediction tool for double layer ZOAB that can be used to predict the effects of variations in the composition of the mixture. The contribution of the Dynamics & Control group to the LOT project is to improve the prediction of the 3 dimensional (3D) contact forces that are the input for the lifetime calculations.

1.1 Background

The LOT project is an assignment of the DWW (Dienst Weg- en Waterbouwkunde) of the Rijkswaterstaat to the Faculty of Civil Engineering and Geosciences of the Delft University of Technology to develop a lifetime prediction tool for double layer ZOAB. The goal of this tool is to predict the damage due to ravelling and the influence of the mixture composition on the lifetime of double layer ZOAB. In this project a fundamental approach has been chosen where an accurate prediction of the material and mechanical behaviour of double layer ZOAB is needed in order to improve the mixture properties.

As a consequence of the limited time available for the project (1 year) and the complexity of the subject, the output of the LOT project is not a complete system, but a first generation version that can be improved in later projects. The LOT is based on already available knowledge which has been extended to make the tool useful for the comparison between different mixtures. The calculation tool makes it possible to predict the relative effect of changes of the composition of the mixture on the lifetime of double layer ZOAB.

The LOT has the following parts: calculation system, load library and material library. The heart of the LOT is the calculation system, which can predict the stress and strain distributions in the bituminous mortar and in the attachment surface between stone and bituminous mortar and is based on the Finite Element package ABAQUS. The load library contains the 3D force distribution for a tyre load of 50 kN and an inflation pressure of 850 kPa. Finally the material library consists of 1 polymer-modified mortar and 2 stone types.

Experimental material research has been carried out at several different laboratories in order to provide data for the material library. Furthermore, the lifetime predictions have been validated in real-life experiments on a test-track under controlled conditions.

The stone size and void ratio of the upper layer of double layer ZOAB has been chosen to comply with the requirements that the noise reduction properties should be maintained and the roughness of the road surface should comply with the latest requirements. These considerations have led to the choice of double layer ZOAB with stone size 4/8 mm and a 20% void ratio in the upper layer.

The following institutes have participated in the LOT project:

- Faculty of Civil Engineering and Geosciences of the Delft University of Technology (project leader, calculation system, material experiments)
- The Adhesion Institute (Hechtinsinstituut) of the Delft University of Technology (material experiments)
- Université Libre de Bruxelles (material experiments)
- STUVAtec Köln (validation experiments)
- RWTH Aachen (supervisor validation experiments)
- Faculty of Mechanical Engineering of the Eindhoven University of Technology (contact forces)
1.2 Tyre/road contact forces

As stated earlier, the contribution of the Dynamics & Control group of the Eindhoven University of Technology to the LOT is to improve the prediction of the 3D contact force distribution that is the input for the lifetime calculations. One of the crucial factors for an accurate prediction of wear in asphalt roads is the contact force between tyre and road. Finding these forces is not a trivial task. Two main approaches are possible: modelling of tyre/road contact and directly measuring the forces. Both approaches have strengths and weaknesses, which means that ultimately a combination of modelling and measurements should be used.

The main aspects that have to be taken into account in order to obtain an accurate model of the pressure distribution in the contact between tyre and road for road wear analyses are: road texture, tyre profile, tyre dynamics and vehicle dynamics. The activities at the Dynamic & Control group of the Eindhoven University of Technology have focused on the modelling of the contact between a tread block of the tyre profile and the road and analysing the dynamic behaviour of the test set-up at STUVAtec (the company STUVAtec from Köln has carried out large scale lifetime experiments in controlled conditions at an in-door test track). Additionally the applicability of Fuji Prescale films to tyre/road contact measurements has been investigated.

This document is organised as follows. In chapter 2 the analysis of the dynamic behaviour of the STUVA test set-up is described and guidelines for an optimal track layout are given. The available information about 3D contact forces is presented in chapter 3 and the experiments performed to determine the contact pressure distribution are summarised. In chapter 4 the approach followed to calculate the 3D contact forces is explained, the FE model of the tread block is described and the main results of the contact force calculations are summarised. Finally in chapter 5 the main conclusions of this part of the project are presented and recommendations for future work are given.
2 Dynamic analysis of the STUVA set-up

2.1 Introduction

Building a dynamic model and performing a dynamic analysis of the test set-up at STUVAtect (STUVA test set-up) was not originally included in the project plan. However, as a result of the discussion at the kick-off meeting of 23rd Augustus 2006 about the influence of the track lay-out of the STUVA test set-up on the contact forces, the TU Eindhoven proposed to build a multi-body model of the test set-up and calculate the contact force variations due to the dynamic interaction between the load-axle-tyre system and the test-track. These force variations should be small (below 10% of the nominal load) in order to make the comparison between the simulation results from the asphalt models developed at TU Delft and the experiments at STUVAtect meaningful.

2.2 Description of the STUVA test set-up

The STUVA test-track consists of 16 plates which are shaped to form a circular track as can be seen in the top-view shown in Figure 1(a).

![Figure 1: Schematic view of the STUVA test set-up (courtesy of STUVAtect). (a) Top, (b) Front.](image)

Due to the unevenness of the base ground of the STUVA set-up and the manufacturing process of the plates, there are variations on plate height and evenness (horizontality). The former leads to abrupt height changes in the transition from one plate to the next and the latter to unevenness of the test-track. The interaction of this uneven test-track with the load-axle-tyre system can cause contact force variations.

The load-axle-tyre system consists of a revolving turret that supports a rigid axle which can be loaded with a loading weight. Two truck tyres are attached at both ends of the rigid axle. The axle is 10 meters long, which means that the length of the path the tyres travel is 31.4 meter. The tyres are connected to the axle through a suspension element formed by an air spring and a shock absorber, as shown in Figure 1(b). The full load of the loading weight and axle rests on this suspension and, therefore, on the tyres. The technical specifications of the suspension and loads can be found in Appendix A.1. This information has been used to build a dynamic model of the load-axle-tyre system.

2.3 Dynamic model of the STUVA test set-up

In this project two dynamic models of the STUVA test set-up have been developed: a simplified 2-dimensional (2D) model and a full 3-dimensional (3D) multi-body model. The information about the dynamic
characteristics of the system has been gathered from the technical specifications provided by STUVAtec (Appendix A.1). However, it should be noted that there is always a difference between the estimated dynamic properties and the real dynamic properties, which can be determined experimentally. This aspect should be kept in mind when interpreting the results of the analysis.

The simplified 2D model has been developed in order to speed-up the calculations in the preliminary phase of the analysis, since decisions about the track layout and manufacturing process of the plates had to take place as soon as possible. The main simplifications made in this model are:

- The tyres are modelled as a lumped mass and a spring.
- Point contact between tyre and road is assumed.
- Only vertical displacements (and forces) are taken into account.

Despite the above mentioned simplifications, the 2D model of the STUVA set-up allows for a sufficiently accurate representation of the main dynamic phenomena that influence the vertical contact force. The details of the 2D model can be found in Appendix A.2.

The full 3D multi-body model of the STUVA test set-up has been developed in SimMechanics, multi-body software that runs in Matlab. A view of the model is given in Figure 2.

![Figure 2: Model of the STUVA test set-up](image)

The tyres have been modelled using the DELFT-TYRE model for Matlab developed by TNO, which is based on the SWIFT model of Pacejka [1]. The 3D model allows for the calculation of all 3 contact forces in a realistic contact situation, including the enveloping characteristics of the tyre [2]. Detailed information about the 3D multi-body model can be found in Appendix A.3.

The final advice for the optimal track layout has been given based on calculations performed with the full 3D multi-body model.

2.4 Simulation results

2.4.1 Results of the preliminary calculations

The results of the preliminary analysis indicate that the discontinuities between track sections have relatively little importance compared to the unevenness of the track. Large contact force variations (above 10 kN) can occur for relatively small net height differences along the track (2 cm over a track length of 31.4 m). Not only the net height difference but also the shape of the track unevenness influences the force variations. To illustrate this point the calculated force variation for a travelling velocity of 80 km/h and for two different track layouts is summarised in Table 1. Both layouts have the same net height difference between the lowest and the highest point on the track but different shapes (see Figure 3). These shapes correspond to two extreme situations that have been found in the preliminary analysis.
Table 1: Calculated force variation. Travelling velocity of 80 km/h. Average load per tyre 50 kN.

<table>
<thead>
<tr>
<th></th>
<th>Layout 1</th>
<th>Layout 2</th>
</tr>
</thead>
<tbody>
<tr>
<td>Amplitude of contact force variation (kN)</td>
<td>2.3</td>
<td>15.1</td>
</tr>
<tr>
<td>% of average load per tyre</td>
<td>4.6</td>
<td>30.3</td>
</tr>
</tbody>
</table>

The above results show that, despite the fact that the net height difference is the same, there are large differences between the force variations of the two layouts. A thorough analysis of the shape of the layouts is needed to understand this result.

The height difference between the highest and the lowest point on the track is 16 mm for both track layouts. However, in Layout 1 all plates are taken as perfectly horizontal, while in Layout 2 all plates are slightly inclined (3 mm). The height difference between two adjacent plates is 2 mm in Layout 1 and 1 mm in Layout 2. The results in Table 1 show that the discontinuity in between plates does not have a significant influence on the force variation, since the layout with the largest discontinuity gives the lowest force variation.

The mayor difference between the two layouts is that in Layout 1 there is one highest and one lowest point and in Layout 2 there are two highest and two lowest points. Since the tyres are located at opposite positions on the track at all times, on the track with Layout 1 a rocking motion of the axle occurs as it rotates on the track, with one tyre reaching the highest point when the other tyre reaches the lowest point. Whereas on the track with Layout 2 the axle remains horizontal and both tyres move up and down simultaneously as they travel along the track. The interaction of this motion of the tyres with the dynamic behaviour of the load-axle-tyre system originates the force variations. The natural frequency of the load-axle-tyre system corresponding to this up and down motion is 1.48 Hz (see Appendix A.2) which is very close to the frequency associated to Layout 2 for a travelling velocity of 80 km/h (1.43 Hz). As a consequence large oscillations of the sprung-mass (load-axle) occur, which lead to large force variations.

This result should be handled with care, since it has been pointed out in section 2.3 that the real dynamic characteristics of the system are not known and have been estimated from the technical specifications (Appendix A.1). This means that the natural frequency of the up-down motion (1.48 Hz) is not exactly known and variations in this frequency can dramatically influence the vertical contact force variations. Similarly, small changes in the travelling velocity can also have a significant effect on the excitation frequency and, therefore, on the contact force.

The above analysis shows that the shape of the track unevenness has a significant influence on the contact force variations. In practice each plate has a different inclination and the plates are placed to make the
transition from one plate to the other as smooth as possible (small height difference between adjacent plates). As shown above this strategy does not guarantee a small contact force variation, but it should be noted that a layout like Layout 2 is unlikely.

Many different arbitrary track layouts have been analysed in order to determine the guideline to be followed in the manufacturing process of the test plates. As a result the recommendations summarised below have been produced:

- The total height difference of the track should be smaller than 40 mm.
- The height difference between two adjacent plates should be smaller than 2 mm.
- A track layout like Layout 2 (Figure 3(b)) should be avoided by all means.
- The height variations of the base ground of the STUVA set-up should be measured.

The above recommendations have led to an improved manufacturing process in order to minimise the height differences between the 16 plate sections. Added to this, the height variations of the base ground of the STUVA set-up have been measured and the corresponding force variations have been calculated using the 3D multi-body model of the STUVA set-up. The predicted force variation for this profile is 33% of the nominal load (±16kN), which does not comply with the requirement of TU Delft that the force variation should be below 10% of the nominal load. The height profile and corresponding force variation can be found in Appendix A.4. Consequently, the base ground of the STUVA set-up has been modified to reduce the height differences.

2.4.2 Recommendations for the optimal track layout

In order to ensure an as even as possible track at the STUVA set-up, the following strategy has been followed:

- Measure height profile of the final base ground at the STUVA set-up.
- Measure height of the 16 plate sections.
- Search for the base ground/plate combination that gives the lowest force variation.

The measured height profiles of the base ground and of the 16 plates are plotted in Figure 4.

![Figure 4: Measured height profiles [mm]. (a) base ground, (b) test plates.](image)

Table 2: Most promising segment/plate combinations.

<table>
<thead>
<tr>
<th>Segment number</th>
<th>1</th>
<th>2</th>
<th>3</th>
<th>4</th>
<th>5</th>
<th>6</th>
<th>7</th>
<th>8</th>
<th>9</th>
<th>10</th>
<th>11</th>
<th>12</th>
<th>13</th>
<th>14</th>
<th>15</th>
<th>16</th>
</tr>
</thead>
<tbody>
<tr>
<td>Configuration 1</td>
<td>A1</td>
<td>A2</td>
<td>A3</td>
<td>A4</td>
<td>B1</td>
<td>B2</td>
<td>B3</td>
<td>B4</td>
<td>C1</td>
<td>C2</td>
<td>C3</td>
<td>C4</td>
<td>D1</td>
<td>D2</td>
<td>D3</td>
<td>D4</td>
</tr>
<tr>
<td>Configuration 2</td>
<td>D1</td>
<td>D2</td>
<td>D3</td>
<td>D4</td>
<td>A1</td>
<td>A2</td>
<td>A3</td>
<td>A4</td>
<td>B1</td>
<td>B2</td>
<td>B3</td>
<td>B4</td>
<td>C1</td>
<td>C2</td>
<td>C3</td>
<td>C4</td>
</tr>
<tr>
<td>Configuration 3</td>
<td>D1</td>
<td>D2</td>
<td>D3</td>
<td>D4</td>
<td>A1</td>
<td>A2</td>
<td>A3</td>
<td>A4</td>
<td>C1</td>
<td>C2</td>
<td>C3</td>
<td>C4</td>
<td>B1</td>
<td>B2</td>
<td>B3</td>
<td>B4</td>
</tr>
<tr>
<td>Configuration 4</td>
<td>D1</td>
<td>D2</td>
<td>D3</td>
<td>D4</td>
<td>A1</td>
<td>A2</td>
<td>A3</td>
<td>A4</td>
<td>B1</td>
<td>B2</td>
<td>B3</td>
<td>B4</td>
<td>C2</td>
<td>C3</td>
<td>C4</td>
<td>C1</td>
</tr>
</tbody>
</table>

The numbers 1 to 16 in Figure 4(a) are the segment numbers used to identify track positions and the letters A to D in Figure 4(b) indicate the type of plate. Several different segment/plate combinations have been
simulated in order to find the combination with the lowest force variations. Four of these configurations are represented in Table 2 and the corresponding force variations are summarised in Table 3.

Table 3: Calculated force variations for 80 km/h and nominal load 50 kN.

<table>
<thead>
<tr>
<th>Configuration</th>
<th>1</th>
<th>2</th>
<th>3</th>
<th>4</th>
</tr>
</thead>
<tbody>
<tr>
<td>Force variation (kN)</td>
<td>9.1</td>
<td>5.4</td>
<td>7.4</td>
<td>6.7</td>
</tr>
<tr>
<td>% nominal load</td>
<td>18.2</td>
<td>10.8</td>
<td>14.7</td>
<td>13.3</td>
</tr>
</tbody>
</table>

The combination which gives the lowest force variation is Configuration 2, where plates D1 to D4 are placed on segments 1 to 4, A1 to A4 on segments 5 to 8, B1 to B4 on segments 9 to 12 and C1 to C4 on segments 13 to 16. The test plates have been placed at the STUVA set-up following this configuration.

2.4.3 Results of the final track layout

The measured height profile of the final track layout has been used as input to the 3D multi-body model of the STUVA set-up and the normal contact force variations have been calculated. The results are summarised in Figure 5. The measured profile is plotted in Figure 5(a) and the corresponding normal force in Figure 5(b). The letters indicate the position of the plates of type A to D.

![Figure 5: Final track lay-out. (a) Measured profile (mm), (b) Vertical force variations (kN).](figure)

As can be seen in Figure 5(a), the total height difference on the final track is 12 mm and the corresponding force variation (Figure 5(b)) is 12.4% of the nominal load, which is higher than the desired maximum of 10%. However, in the lifetime calculations a fixed value of the nominal contact force is used for all plates. Since the vertical contact force varies along each plate, the average normal force for each plate should be calculated in order to evaluate the vertical contact force variations from one plate to the next. If these variations remain within 10% of the nominal vertical load, the assumption that all plates have the same nominal vertical load can be accepted. The average force for each plate has been plotted in Figure 6. The plates are identified according to the type of asphalt (A to D) and plate number (1 to 4).

Figure 6 shows that the average load on each plate differs less than 10% from the nominal load of 50 kN. This is considered acceptable by the TU Delft, since it falls within the accuracy limits of their models.
2.5 Conclusions and recommendations

In this chapter the dynamic analysis of the test set-up at STUVAtec has been presented. Two models of the load-axle-tyre system have been developed and the simulation results have led to the following conclusions:

– The contact force variations are mainly due to the fact that the track is not perfectly horizontal. A wavy height profile can cause large force variations.
– The discontinuity between plates (jumps) is not a critical problem, provided the jumps are not too large.
– Precise knowledge of the dynamic properties of the system and of the operating conditions (travelling velocity) is needed in order to provide an accurate estimation of the contact forces.

In the light of the above conclusions, several recommendations can be made for future research:

– An experimental modal analysis of the load-axle-tyre system at STUVAtec should be performed in order to determine the dynamic properties of the system.
– In order to extract additional information from the tests, the vertical acceleration at the connection points of the tyres to the axle should be measured and recorded. This information can be used as input to the model in order to make an estimation of the contact forces at a given number of runs. In this way, the observed changes on the surface of the asphalt could be correlated with changes in the contact force variations.
– The travelling velocity should be monitored and recorded during the tests, in order to be able to relate force variations to variations of the travelling velocity.

Figure 6: Calculated average load per plate (kN).
3 Measured 3D contact forces

In this chapter the experimental data on 3D contact forces available in this project is described and the experiments performed at the TU Eindhoven to measure contact pressure distributions are summarised.

3.1 Description of the TireView data

TireView is a computer program to estimate tyre contact pressure distributions in the longitudinal, lateral and vertical directions. It was developed at the Texas Transportation Institute (TTI) in research project 0-4361 sponsored by the Texas Department of Transportation (TxDOT) [3]. The program uses an interpolation routine on a data base of measured tyre contact pressures, obtained from tests. The tests have been performed on the Vehicle-Road Surface Pressure Transducer Array (VRSPTA) of the Division of Roads and Transport Technology of the CSIR (Pretoria, South Africa) [4]. This system measures the 3D contact pressure distribution of a tyre slowly (0.35 m/s) rolling over a line of 20 instrumented pins. Details about this system can be found in Appendix B.

In Figure 7 the contact force distribution obtained from TireView is shown for a Supersingle tyre 425/65 R22.5 with an inflation pressure of 8.5 bar and a load of 50 kN. The x position indicates the position on the contact patch along the driving direction (longitudinal direction). The pin number indicates the corresponding pin in the measurement array across the width of the tyre (transversal direction).

![Contact force distribution](Image)

**Figure 7:** Contact force distribution for an inflation pressure of 8.5 bar and load of 50 kN. (a) Vertical force (Fz), (b) Longitudinal force (Fx), (c) Transversal force (Fy). Forces in kN.

In order to use this information to validate a Finite Element (FE) model of the tyre tread it is crucial to know where the pins are located with respect to the tread profile. Unfortunately the contact forces provided by TireView are not directly measured but interpolated from a limited set of measured data, which means that the position of the pins cannot be known exactly.

![Estimated pin positions](Image)

**Figure 8:** Estimated position of the measuring pins relative to the tread blocks.
The position of the pins has been estimated by visual inspection of the data in Figure 7 and it is shown in Figure 8. The circles give the real size of the pins with respect to the tread blocks. The arrows in Figure 8 indicate the rolling direction (x) and the positive transversal direction (y). Further inspection of the data in Figure 7 has led to the choice of the contact force information from pins 8 to 10 and 14 to 16 as the most suitable to be used as validation data for the FE model. The contact forces from pins 14 to 16 are shown in Figure 9.

The following remarks can be made about the contact force distributions plotted in Figure 9. The largest vertical and longitudinal force are measured at pin 15 (Figure 9(a)), while the transversal force is close to zero (Figure 9(c)). This indicates that pin 15 is located in the middle of the tread block. The transversal force measured at pin 14 is positive, which implies that pin 14 must be located on the right side of the tread block according to the rolling direction. A similar reasoning leads to the conclusion that pin 16 must be located on the left side of the tread block. It can also be seen in Figure 9(b) that all three longitudinal forces are negative (opposite to the rolling direction), which indicates that the pins are in contact with the rear side of the block.

The contact force distributions from Figure 9 have been used to validate the FE model that is described in the next chapter.
3.2 Contact pressure measurements at TU Eindhoven

It has been explained in the previous section that the contact forces at a given pin provided by TireView cannot be related to a specific location on the tyre tread. Furthermore, the available information is limited to a few tyres and predefined loading and inflation pressure conditions. In an attempt to obtain more accurate information about the normal contact pressure distribution, the applicability of Fuji Prescale films to tyres has been investigated.

A Fuji Pre-scale Film consists of microcapsules filled with colour forming material, which break when pressure is applied. When a microcapsule is broken, the colour forming material is released. The pre-scale films are designed with a so called Particle Size Control Technology. This means the microcapsules are layered and therefore the capsules will react to various degrees of pressure. The pressure sensitive films release the colour forming material in a density corresponding to a specific level of applied pressure. Details about Fuji Prescale films can be found in Appendix B.2.

The measurements have been carried out at the Flat Plank Tyre Tester at the Automotive Laboratory of the TU Eindhoven (Figure 10(a)). In this tester a controlled load can be applied to the tyre. However, due to their large radius, it is not possible to mount truck tyres. For this reason, a car tyre has been used in the experiments (Continental SportContact2). The tread of this tyre can be seen in Figure 10(b).

Figure 10: (a) General view of the Flat plank Tyre Tester. (b) Tread of Continental SportContact2.

Several experiments have been carried out for different inflation pressures and tyre loads on a flat surface and on asphalt [5, 6]. Two types of Fuji Films have been tested: LLLW (range 0.2-0.6 MPa) and LLW (range 0.5-2.5 MPa). The LLLW film has been used for measurements on a flat surface and the LLW film has been applied to measurements on asphalt.

The experiments have been performed by increasing the load to the nominal value in 5 seconds, keeping the load constant during 5 seconds and decreasing the load to zero in 5 seconds. The measured footprint for a load of 4 kN and an inflation pressure of 2.3 bar is shown in Figure 11 on a flat surface and on asphalt. The pictures are plotted in gray-scale, but the real colour of the films is magenta.

The footprint for a tyre on a flat surface is significantly different from the footprint on asphalt, as expected. On a flat surface the profile of the tyre tread can be clearly recognised, while on asphalt it is mainly the individual stones that can be seen. The different tonalities of magenta in the Fuji films can be coupled to a pressure value and the contact pressure distribution can be determined. The results are shown in Figure 12.
Comparing Figure 12(a) to Figure 12(b) shows that locally, the normal contact pressure is significantly larger for a tyre on asphalt. The peak pressure values are about 4 times larger on asphalt than on a smooth surface. However this result should be handled with care, since in both measurements the pressure reaches the saturation value of the films and it is uncertain what the real peak values of the pressure are.

In order to estimate the accuracy of the Fuji Films the average contact force has been compared to the applied load and the average contact pressure to the inflation pressure. Depending on the applied load and the inflation pressure the error ranges from 1% to 14% on a flat surface and from 8% to 19% on asphalt [6].

The contact pressure distributions measured with the Fuji Prescale Films give a good qualitative indication of the differences in pressure distribution between a flat surface and asphalt. However the experiments performed up to the present time do not provide reliable quantitative information to be used in the validation process of the FE model. Further experiments with truck tyres on a flat surface and on asphalt are needed in order to establish the applicability of Fuji Prescale Films to gather data for comparison to results from FE models.
3.3 Conclusions

In this chapter the available experimental information about the contact force distribution in the tyre/road contact has been described and the measurements performed at the TU Eindhoven have been briefly described. The main conclusions from this chapter are:

- The contact forces determined with TireView are not directly measured data but values interpolated between a number of measured data points. Therefore, the force at a given pin cannot directly be related to a given position on the tyre tread.

- The vertical contact pressure distributions measured with the Fuji Prescale Films give a good qualitative view of the differences between contact with a flat surface and with asphalt. Although the results are promising, further measurements with a truck tyre are needed in order to produce contact pressure data for comparison with FE calculations.
4 Calculation of the 3D contact forces

4.1 Approach

The interaction between tyre and road is a complex phenomenon which is influenced by many different factors, such as: tread geometry, rubber properties, road texture, tyre dynamic behaviour, vehicle dynamics, road conditions, etc. In order to determine the 3D contact forces between tyre and road all these factors should be considered. However, the most relevant factors that influence the 3D contact forces depend strongly on the research goal. In the present project the goal is to predict the variations of the 3D contact forces that are relevant for lifetime predictions of asphalt. Therefore the focus lies on the local interaction between a tread block and the road.

The model for the prediction of the contact stress distribution is based on the following assumptions:

- The size and the form of the tyre contact patch are known and the tyre belt is considered infinitely stiff.
- The roughness profile (macro-texture) of the road in the contact patch is given and the road is infinitely stiff.
- The dynamics of the vehicle and the interaction with the long wavelength texture is not considered.

In the approach followed in this project the contact patch is fixed and the tread block travels along the contact patch together with the road. In this way the effect of the travelling velocity of the vehicle on the contact forces is included. This is schematically shown in Figure 13, where the block/sensor pair travels opposite to the travelling direction of the tyre.

\[\text{Travelling direction}\]

\[\text{Pressure}\]

\[\text{Belt} \]

\[\text{Road} \]

\[\text{Block} \]

\[\text{Sensor} \]

\[\text{Travelling direction}\]

\[\text{Pressure}\]

\[\text{Belt} \]

\[\text{Road} \]

\[\text{Block} \]

\[\text{Sensor} \]

Figure 13: Tread block travelling along the contact patch. (a) Quasi-static. (b) 80 km/h

As can be seen in Figure 13(b) due to the rotation of the tyre the contact pressure distribution is not symmetric, which means that the local pressure peaks are larger for a tyre rotating at a travelling speed of 80 km/h than for a stationary tyre. This effect can be modelled by using asymmetric spatial weighting functions (spatial windows) which have been determined from the literature [7].

Another important factor of influence on the contact stress distribution is whether the vehicle rides on a straight track or on a bend. It has already been explained in section 2.2 that the STUVA test track is a circular track of radius 5 m. The tyres are forced to ride in a perfect circle, which is an unnatural riding situation. In this situation a phenomenon known as turn-slip occurs. The force increase due to this effect has been estimated through calculations performed with the multi-body model described in section 2.3. The mechanism of turn-slip and the results of the simulations are explained in more detail in section 4.4.
The tyre chosen for this project is the Goodyear tyre 425/65R22.5 G286A 165KL TL. Measured contact force data is available in the TireView program. The inflation pressure has been set at 8.5 bar and the load per tyre is 50 kN. In Figure 14 an impression of the footprint and a close view of one tread block are given.

![Footprint of the Goodyear tyre 425/65R22.5 G286A 165KL TL.](image1)

As explained above, the interaction between a single tread block and the road has been modelled. A general view of the tread block/road model for FEM is given in Figure 15.

![General view of tread block/road model for FEM.](image2)

The contact patch is fixed and the tread block travels along the contact patch together with the road. This implies that the size and the form of the tyre contact patch are known and the tyre belt is considered infinitely stiff. The length of the contact patch (0.294 m) and the effective rolling radius (0.526 m) of the tyre have been determined from simulations with the multi-body model described in section 2.3. With this information, the Koutny model described in [9] has been used to find the deformed shape of the tyre, as shown in Appendix C.3. Once the deformed shape is known, the velocity boundary condition for the upper surface of the tread block has been calculated from the knowledge of the travelling velocity and effective rolling radius. The velocity boundary condition applied to the upper surface of the block is in fact the relative velocity between the belt and the road. This relative velocity changes as the block travels along the contact and causes a shearing motion of the block. This effect is illustrated in Figure 16.
Figure 16: Shearing motion of the block as it travels along the contact patch.

The FE model of the tread block has been built in two steps: validation in quasi static conditions and prediction of the contact forces for a travelling speed of 80 km/h. A schematic view of the validation process in quasi static conditions is given in Figure 17.

The rubber material properties have been taken from the literature [8] and the shape of the quasi-static normal pressure distribution along the contact patch has been obtained from the TireView data (see chapter 3). With this information the parameters of the FE model of the block have been tuned to provide a good qualitative and quantitative comparison between the calculated longitudinal and transversal forces and the experimentally determined forces from TireView. Once the parameters of the FE model are tuned, the next step is to predict the contact forces at 80 km/h. The prediction step is schematically shown in Figure 18. The boundary conditions have been modified to account for the asymmetry of the pressure distribution due to the travelling velocity and to include the difference in relative velocity between the belt and the road.

Figure 17: Schematic view of the validation process.
4.2 Description of the FE models

Two FE models of the tread block have been built: a simplified 2D model and a full 3D model. A general view of these models is given in Figure 19.

![Figure 19: FE models of the tread block. (a) 2D model, (b) 3D model.](image)

The 2D model in Figure 19(a) can predict normal and longitudinal forces and has been built in order to speed up the calculations. The length and depth of the block have been directly measured on the tread of the test tyre. The block in the 2D model is per definition rectangular. The elements in this model are 4-node bilinear plane strain quadrilateral, hybrid, constant pressure elements (CPE4H).

The 3D model of the block (Figure 19(b)) can predict all 3D contact forces (normal, longitudinal and transversal). The geometry of the block has been directly measured on the tread of the test tyre (a sketch can be found in Appendix C.2). The elements in this model are 8-node linear brick, hybrid, constant pressure elements (C3D8H).

In both FE models the hyperelastic behaviour of the rubber material has been modelled with a Mooney-Rivlin material model with parameters C01 and C10. The material properties have been taken from reference [8] and can be found in Appendix C.1.

4.3 FE simulation results

It has already been explained in section 4.1 that the FE simulations have been performed in 2 steps: a validation step in quasi-static conditions and a prediction step at 80 km/h. Before the prediction step dynamic FE simulations have been performed to assess the influence of the inertia and damping effects on the contact forces. The results of the FE simulations are summarised below.

4.3.1 Results of the static simulations

The experimental data from TireView has been obtained at very low travelling velocities (1.5 km/h). In this situation it is reasonable to assume that the mass and damping of the tread block can be neglected and the
deformation of the tread block can be calculated in quasi-static conditions. Therefore, a static FE analysis has been performed in ABAQUS Standard where the load and the longitudinal velocity are varied to simulate the passage of the block along the contact.

The load and boundary conditions applied to the block are a position dependent normal pressure and a position dependent longitudinal velocity profile on the upper surface (belt side) of the block. In this context “position” means longitudinal position of the block within the contact patch. The longitudinal x-axis is defined in the travelling direction with the origin at the centre of the contact patch (see Figure 30). The normal pressure and longitudinal velocity distributions are plotted in Figure 20. The form of the pressure distribution has been derived from the TireView data (section 3.1) and the magnitude from the inflation pressure of the tyre, which has been corrected to account for the effective contact area. The velocity profile has been determined from the travelling velocity of 1.5 km/h and the knowledge of the deformed shape of the tyre. The deformed shape of the tyre has been found using de Koutny model described in [9] and the details can be found in Appendix C.3.

![Figure 20](image)

**Figure 20**: Load and boundary conditions applied in the quasi-static calculations. (a) Normal pressure [N], (b) Longitudinal velocity [m/s].

![Figure 21](image)

**Figure 21**: Sensor positions in the FE models. (a) 2D model, (b) 3D model.

In order to be able to compare the data from the FE simulations with the experimental data from TireView (section 3.1) the resultant force at the position of the sensors has been calculated. In the 2D model the sensor location is given by a line of length 0.01 m (Figure 21(a)) while in the 3D model three square patches of 0.01x0.01 m have been defined to account for the positions of sensors 14 to 16 (Figure 21(b)).
The vertical and longitudinal forces calculated with the 2D model at the location given in Figure 21(a) have been plotted in Figure 22 together with the experimental data from TireView for sensor 15. Results of two different calculations are shown: without a longitudinal velocity (Figure 22(a)) and with the longitudinal velocity given in Figure 20(b) (Figure 22(b)).

The calculated vertical force is significantly lower than the measured vertical force, while the calculated and measured longitudinal forces are similar. Therefore the ratio of the longitudinal to the vertical force is overestimated in the 2D model. A possible explanation for this effect is that the transversal deformation of the block is not taken into account and that the non-uniform geometry of the block is not considered. These two effects lead to an overestimation of the stiffness of the tread block. Added to this, the 2D model does not allow for the calculation of the transversal contact forces. Therefore the 2D FE model does not give an adequate description of the tread block.

![Figure 22: 2D FE model. Comparison of measured (thin) and calculated (thick) forces [N]. (a) No longitudinal velocity, (b) Longitudinal velocity (Figure 20(b)).](image)

It can be seen from the comparison of Figure 22(a) and (b) that the longitudinal velocity causes a non-symmetric longitudinal force, which is closer to the shape of the measured longitudinal force than the longitudinal force for zero velocity.

![Figure 23: Calculated contact pressure distribution (dark areas indicate higher pressures).](image)
The load and boundary conditions from Figure 20 have also been applied to the 3D FE model. In Figure 23 an impression of the calculated contact pressure distribution is given. The normal contact forces at the locations given in Figure 21(b) have been calculated from the integral of the pressure over the corresponding surfaces. The contact forces calculated with the 3D FE model of the tread block are plotted in Figure 24 together with the measured forces for the 3 sensor locations given in Figure 21(b). Results are shown for two different situations: without longitudinal velocity Figure 24(a) and with longitudinal velocity Figure 24(b).

There is a good correlation between the calculated contact forces and the measured data from TireView. The main effect of applying a longitudinal velocity to the upper surface of the block is to increase the longitudinal force at the front side of the contact, which is also seen in the measured data. However, the peak value of the longitudinal force is overestimated when a longitudinal velocity is applied. There are many factors that can (partly) explain the discrepancy between the TireView data and the predicted forces: the material model and material parameters, the geometry of the tread, the velocity profile and the TireView data itself.

The material model and material parameters have a significant influence on the predicted contact forces. It can be easily understood that if the material stiffness is decreased the longitudinal forces will be smaller for the same shear deformation. Dedicated laboratory measurements are needed to extract the material parameters of the tread rubber compound.

The geometry of the tread can also influence the predicted contact forces. In the approach chosen in the present project an individual tread block has been studied, which dimensions have been directly measured on the tyre tread. In practice, the tread blocks are connected to each other, which will affect the individual stiffness of each tread block. Furthermore, more precise information about the dimensions of one block (especially the height) would be desirable. The height of the tread blocks has also a significant influence on the velocity profile, which in turn determines the maximal shear deformation.

Finally it should be noted that the TireView data is not an optimal dataset for comparison to results from FE calculations. The position of the sensors with respect to the tread profile is not known and the predicted contact forces strongly depend on the exact position on the surface of the tread block where the forces are calculated.
Figure 24: 3D FE model. Comparison of measured (thin) and calculated (thick) forces [N].
(a) No longitudinal velocity, (b) Longitudinal velocity (Figure 20(b)).
4.3.2 Results of the dynamic simulations

In the dynamic simulations a time domain analysis of the block travelling along the contact has been carried out with the 3D FE model and ABAQUS Explicit. In order to avoid higher order dynamic effects the load and boundary conditions have been modified to have smoothed begin and end conditions. The load and velocity profile thus determined are given in Figure 25.

Figure 25: Load and boundary conditions applied in the dynamic calculations.
(a) Normal pressure [N], (b) Longitudinal velocity [m/s].

The amplitudes of the velocity profile in Figure 25(b) have been chosen to give the same shear displacement as the velocity profile in Figure 20(b). Furthermore, these amplitudes correspond to a travelling velocity of 1.5 km/h. For the calculation of the contact forces for a travelling velocity of 80 km/h the amplitude of the velocity has been scaled to give the same shear displacement.

The results of the dynamic simulations for 1.5 km/h and 80km/h are shown in Figure 26(a) and (b) respectively. It is clear that the contact forces for the two different travelling velocities are identical. This leads to the conclusion that the inertia and damping effects due to increasing the velocity from 1.5 to 80 km/h can be neglected and that the contact forces can be determined in a static calculation.

Figure 26: 3D FE model. Calculated contact forces for position 15 [N]. (a) 1.5 km/h, (b) 80 km/h.
In the following figure the contact forces from Figure 26 (dynamic simulation) are compared to the contact forces obtained from a static simulation with the load and boundary conditions from Figure 25. The thin lines correspond to the dynamic simulation and the thick lines to the static simulation.

![Figure 27: Comparison of contact forces from a static (thick) and a dynamic (thin) simulation.](image)

The forces obtained from the dynamic and static simulation shown in Figure 27 are very similar but should be identical. The reason for the clear differences is that different element types have been used in the static and dynamic calculations. ABAQUS Explicit does not accept element type C3D8H (used for the static calculations) and element type C3D8R has been used in the dynamic calculations. The result in Figure 27 indicates that the choice of element type influences the predicted contact forces.

Furthermore, it should be noted that the above results correspond to a material model where no viscoelastic behaviour has been included. If the viscoelastic properties of rubber are included in the calculation, the difference between the results of the dynamic calculations for 1.5 and 80 km/h is very small, but the longitudinal force for 80 km/h with viscoelastic effects included is 10% larger than in Figure 26(b). This gives an indication that the influence of the viscoelasticity of rubber should not be neglected.

The main conclusion that can be derived from the dynamic calculations is that the longitudinal contact force is mainly determined by the shear displacement between the upper and lower surface of the block and that the inertia and, partly, damping effects can be neglected. Therefore accurate information about the displacement of the upper surface of the block is needed in order to give a good prediction of the longitudinal contact force.

### 4.3.3 Prediction of the contact forces at 80 km/h

It has been explained in section 4.1 that for a stationary tyre (0 km/h) the pressure distribution is symmetric, the resultant longitudinal force is zero and the resultant vertical force passes through the tyre centre. When the tyre travels with a given non-zero velocity a resultant longitudinal force appears which gives a moment around the tyre centre and the resultant vertical force is displaced with an offset. These offset can be estimated from the moment equilibrium of the resultant vertical and longitudinal forces (see Figure 28). The resultant forces have been calculated with the multi-body model presented in chapter 2 and are summarised in Table 4.
Figure 28: Illustration of the moment equilibrium of the resultant vertical and longitudinal force.

Table 4: Resultant forces and corresponding offsets.

<table>
<thead>
<tr>
<th>Longitudinal force, $F_L$ (N)</th>
<th>Effective rolling radius, $R_{\text{eff}}$ (m)</th>
<th>Vertical force, $F_z$ (N)</th>
<th>Offset, $d$ (m)</th>
</tr>
</thead>
<tbody>
<tr>
<td>535</td>
<td>0.526</td>
<td>50000</td>
<td>0.0056</td>
</tr>
</tbody>
</table>

The small offset found (less than 2% of the contact length) agrees with results from the literature [7]. This offset has been used to find the non-symmetric deformed shape of the tyre with a modified non-symmetric version of the Koutny model. The velocity profile that corresponds to this deformed shape (calculated as explained in section 4.1) leads to a maximal shear deformation which is 20% larger than the shear deformation for the quasi-static situation (symmetric deformed shape). At this point, it should be noted that the behaviour of rubber is highly non-linear and that this increase in shear does not necessarily lead to a similar increase in longitudinal force. Actually, for the shear values found in the calculations, the longitudinal contact force increases only very slightly.

In order to assess the validity of the above result, the resultant longitudinal force for a travelling velocity of 1.5 km/h has been calculated with the multi-body model presented in chapter 2. This longitudinal force is 323 N, which means that the resultant longitudinal force predicted by the multi-body model is 65% larger when the tyre is travelling at 80 km/h than in the quasi-static situation (1.5 km/h).

It is clear that the two models lead to completely different results. The DELFT-TYRE model of TNO used in the multi-body calculation is based on experimental data and has been extensively validated. Therefore, it must be concluded that the modelling approach based on a non-symmetric variant of the Koutny model is not appropriate to evaluate the increase of the longitudinal contact force with the travelling velocity. However, it should also be noted that the resultant forces obtained with the DELFT-TYRE model can not be extrapolated to estimate the variations of the contact force distribution due to the travelling velocity and give only a qualitative estimation of these force variations.

4.4 Estimation of the effect of turn-slip on the contact forces

As explained in section 2.2, the test set-up at STUVAtac has very specific characteristics which are significantly different from a real vehicle riding on a road. The tyres at the STUVA set-up are forced to ride in a circle (Figure 1(a)) which causes the tyre to rotate around a vertical axis through the centre of the contact patch (z-axis). This effect is known as turn-slip and the result is an increase of, mainly, the transversal forces and a net moment around the z-axis. In Figure 29 an illustration of the slip velocities at the contact area (blue arrows) is given and the resultant forces acting on the road are plotted (red arrows).
Driving direction

Figure 29: Illustration of the turn-slip velocity field on the contact patch and of the resultant longitudinal and transversal forces.

The net resultant longitudinal force is directed opposite to the driving direction and the net resultant transversal force is directed towards the centre of the test track. An rough estimation of these forces has been determined with the multi-body model of the STUVA set-up described in section 2.3 and is given in Table 5 for a travelling velocity of 80 km/h. The resultant contact forces in the absence of turn-slip are also shown for comparison.

Table 5: Resultant contact forces with and without turn-slip. Travelling velocity 80 km/h.

<table>
<thead>
<tr>
<th></th>
<th>With turn-slip</th>
<th>Without turn-slip</th>
</tr>
</thead>
<tbody>
<tr>
<td>Longitudinal force [N]</td>
<td>-586</td>
<td>-535</td>
</tr>
<tr>
<td>Transversal force [N]</td>
<td>-6337</td>
<td>-4</td>
</tr>
<tr>
<td>Moment around z-axis [N.m]</td>
<td>-1550</td>
<td>0</td>
</tr>
</tbody>
</table>

The results in Table 5 show that the main consequence of turn-slip is a resultant transversal force and a net moment around the z-axis. In order to evaluate the relative importance of this effect with respect to the forces determined in the previous section, an estimation of the local increase (at the contact between one block and one sensor location) in contact stresses due to turn-slip is needed. A number of assumptions have been made:

- The resultant longitudinal and transversal forces are due to two forces acting in opposite directions and giving a net moment around the z-axis (see Figure 30).
- These forces are the result of parabolic stress distributions across the length and width of the contact patch.

A schematic view of the resultant contact forces acting on the road is given in Figure 30. The positive x direction is the travelling direction and the positive y direction points to the outside of the circle. The longitudinal and transversal forces are $F_l=586$ N and $F_t=6337$ N respectively. The contact length and width can also be calculated with the multi-body model and are $L_c=0.292$ m and $W_c=0.324$ m respectively. It can easily be shown that the forces depicted in Figure 30 give a resultant moment around the z-axis equal to the result in Table 5 ($M_z=-1550$ N.m).
The contact stress distribution that corresponds to the resultant force in Figure 30 has been estimated under the assumption of parabolic stress distributions. The resulting longitudinal and transversal contact stress distributions have been depicted in Figure 31 (a) and (b) respectively.

The contact stress distributions in Figure 31 have been used to estimate the average longitudinal and transversal forces that act on each tread block as the block travels along the contact patch. The forces “seen” by a sensor of 0.01 m width have been calculated for each tread block (see Figure 32).

The resultant longitudinal and transversal forces “seen” by each of the four sensors along the contact patch are given in Figure 33 (a) and (b) respectively. Comparing Figure 33(a) to Figure 24 it can be concluded that turn-slip does not increase the longitudinal force significantly. The main effect of turn-slip on the longitudinal forces is the shift of the resultant force to the inner side of the contact patch, which makes the forces on this side somewhat higher and the forces on the outer side somewhat lower than in Figure 24.
The comparison of the results in Figure 33(b) with Figure 24 shows that turn-slip has a significant effect on the transversal forces. The magnitude of the additional transversal force is of the same order as the measured force (in absence of turn-slip) and the force distribution is no longer symmetric along the contact patch. Therefore, this additional force should be considered when comparing the lifetime calculations to the experimental results from the STUVA set-up. However, it should be kept in mind that the forces given in Figure 33 are an estimation based on resultant point forces from the multi-body model of the STUVA set-up (section 2.3). A detailed model of the contact area would be needed in order to provide a more accurate prediction of the contact stresses due to turn-slip.

**Figure 32:** Positions where the turn-slip forces have been estimated (strips of 0.01 m width).

**Figure 33:** Estimated contact force on a sensor location \((10^{-4} \, \text{m}^2)\) due to turn-slip \([\text{N}]\).
(a) Longitudinal stress, (b) Transversal stress.
4.5 Conclusions and recommendations

In this chapter the models for the determination of the 3D contact forces have been described and the main results have been summarised. The following conclusions can be derived:

- The 2D FE model overestimates the longitudinal contact forces, which leads to the conclusion that the transversal deformation of the tread block has a significant influence on the longitudinal contact forces.

- The contact forces predicted by the 3D FE model of the tread block in quasi static (low rolling velocity) conditions show a reasonably good correlation with the experimentally determined contact forces. It can therefore be concluded that the proposed approach is appropriate to predict the contact forces in quasi static conditions. However, there are several factors that influence the predicted contact forces, such as the material models and parameters, which should be carefully studied in future projects.

- The dynamic simulations for a travelling speed of 80 km/h show that the inertia and, to a lesser extent, damping effects can be neglected and that the contact forces are mainly due to the static shear deformation of the tread block.

- Regarding the prediction of the contact forces at 80 km/h, the moment equilibrium of the resultant vertical and longitudinal forces calculated with the multi-body model described in chapter 2 leads to an estimate of the offset of the resultant vertical force which is less than 2% of the contact length. This is in agreement with results from the literature [7]. However, if this offset is used to calculate the new shape of the deformed tyre based on the Koutny model, the obtained velocity profile leads to a maximal shear deformation which is only 20% larger than the maximal shear deformation for 1.5 km/h. Due to the nonlinear behaviour of rubber, the expected increase in longitudinal force is much smaller. This contradicts the results of the multi-body model, which predicts an increase of 65% of the resultant longitudinal force if the velocity increases from 1.5 to 80 km/h. Therefore it is concluded that the deformed tyre shape obtained from the Koutny model is not appropriate to determine the velocity boundary condition for the tread block. At this point, further research is needed in order to produce reliable predictions of the 3D contact forces at 80 km/h.

- The main effect of turn-slip is a significant increase and a non-symmetric variation of the transversal force along the contact patch.

The results described in this chapter show that there are many aspects that should be carefully studied in order to improve the prediction of the tyre/road contact forces. Some of those aspects are:

- Accurate information about the material properties of the particular rubber compound of the tread under consideration is needed. This information can be gathered in dedicated experiments performed on samples of this rubber. Added to this, the choice of material model and element type should also be investigated.

- The viscoelastic properties of rubber should be included in the model and their influence on the predicted (longitudinal) contact forces should be assessed.

- A FE model of the tyre should be built in order to determine the deformed shape of the belt. This (exact) deformed shape could be used instead of the approximated Koutry model to determine the velocity profile of a point on the belt as it travels around the tyre. This could be used as boundary condition for a tread section (several blocks) that travels around the tyre belt.

- Regarding the validation of the model, more suitable experimental data for comparison to FE calculations should be generated. A measurement system could be developed where the position of each sensor with respect to the tread is exactly known, which would allow for a meaningful comparison with FE data.
5 Conclusions and recommendations

5.1 Conclusions

In this report the research carried out at the Dynamics & Control group of the Eindhoven University of Technology has been presented. Three main topics have been discussed: dynamic analysis of the test set-up at STUVAtec, investigation of the applicability of Fuji Prescale films to tyre/road contact measurements and Finite Element (FE) modelling of the tyre/road contact. The main conclusions of this work are:

- Large contact force variations can occur at the STUVA test set-up due to the dynamic interaction between the load-axle-tyre system and the height variations of the track. However precise knowledge of the dynamic properties of the test set-up at STUVAtec and of the height profile of the track is needed in order to give an accurate prediction of the contact forces. In a more general context this means that the influence of the vehicle dynamics and of the long wavelength height fluctuations on the contact forces should be taken into account.

- The experimentally determined contact force from the TireView database cannot directly be related to a fixed position on the tyre tread, which makes it cumbersome to use this data for validation purposes. Therefore it is recommended to search for alternative measurement techniques in order to obtain more suitable data for FE validation. However the experiments carried out in this project to determine the contact pressure distribution, although promising, have not provided more suitable data.

- The contact forces predicted by the 3D FE model of the tread block in quasi static (low rolling velocity) conditions show a reasonably good correlation with the experimentally determined contact forces. It can therefore be concluded that the proposed approach is appropriate to predict the contact forces in quasi static conditions. However, there are several factors that influence the predicted contact forces, such as the material models and parameters, element type, tread geometry and velocity profile, which should be carefully studied in future projects.

- The dynamic simulations for a travelling speed of 80 km/h show that inertia effects can be neglected and that damping effects have a small influence on the longitudinal contact force with respect to the travelling velocity of 1.5 km/h. Therefore it can be concluded that the longitudinal contact forces are mainly due to the static shear deformation of the tread block. However, it should be noted that the viscoelastic properties of rubber can not be neglected and should also be included in the model.

- Regarding the prediction of the contact forces at 80 km/h, the velocity profile obtained with the modified Koutny model leads to a maximal shear deformation which is only 20% larger than the maximal shear deformation for 1.5 km/h. Due to the nonlinear behaviour of rubber, the expected increase in longitudinal force is much smaller. This contradicts the results of the multi-body model presented in chapter 2, which predicts an increase of 65% of the resultant longitudinal force if the velocity increases from 1.5 to 80 km/h. Therefore it is concluded that the deformed tyre shape obtained from the Koutny model is not appropriate to determine the velocity boundary condition for the tread block. At this point, further research is needed in order to produce reliable predictions of the 3D contact forces at 80 km/h.

- The influence of turn-slip on the 3D contact forces has been estimated based on the resultant contact forces obtained from the multi-body model described in chapter 2. The main effect of turn-slip is a significant increase and a non-symmetric variation of the transversal force along the contact patch.

5.2 Recommendations

As mentioned in the introduction, there are important factors that affect the contact forces between tyre and road and, due to the limited time-span, have not been included in the project description. There are many challenging aspects which should be thoroughly investigated over a larger time span (4-5 years). Typically, this should be done through a number of dedicated projects.
The model of the tyre/road interaction should be extended and validated further to include all relevant tyre dynamics. Furthermore, the model must be suitable to be used for vehicle dynamics analyses. A multi-body model of a truck with trailer including all the relevant dynamic effects has already been developed at the Dynamics & Control Group. This multi-body model together with the extended tyre model would allow for the calculation of the contact forces in realistic conditions like, for example, during cornering, braking, driving over potholes, etc. The model would also help to understand the role of the interaction between the dynamics of the vehicle and road unevenness in the generation of road surface wear. Developing and validating a complete vehicle-tyre-road interaction model could be the topic of a continuation project.

It has already been said that it is quite difficult to determine contact stresses of a tyre on a road surface experimentally. Especially, the shear stresses are difficult to measure. In general, the measurement problem can be considered from two sides:

1. The road surface is instrumented to measure the contact stresses.
2. The tyre is instrumented to measure the contact stresses.

In both cases the “surface” that is used for measuring is somehow modified, i.e. it differs from the not instrumented surface and is therefore less realistic, which means that the contact stresses will be different. One way to overcome this problem is to build a contact stress measurement device that is as close to reality as possible. In case of the Road Surface Pressure Transducer Array one can try to use more sensors or to make the top of the pins different. One could for example think of putting small blocks with the texture and friction properties of a certain asphalt type on top of the pins. A limitation however is that there must always be gaps between the individual measurement cells or pins to be able to assign a force/pressure to a certain area. Another way to overcome the problem that the surfaces are modified is to accept this and to use computer models of the tyre that are validated with the measurements (on these non ideal surfaces) to predict the forces on different asphalt / road surface types. In this latter approach it is important to have a controlled environment, for example a setup in a laboratory environment that can be used to measure the tyre contact stresses at various controlled conditions (different vertical loads, sideslip angles, brake forces, etc.).

An important difference between instrumenting a tyre and a road surface is that a tyre can be mounted on a vehicle that can be driven at various road surfaces. Measurement data can be captured over the whole surface. When a sensor is mounted in the road surface, only a snapshot of the tyre contact stresses when driving over the sensor can be made. In this context the test-track at STUVAtec could be improved to include a monitoring system that would measure the forces and/or accelerations at the tyre hub and the contact forces at discrete location on the road surface. In this way it would be possible to reconstruct the time history of the contact force distribution along the full length of the track. However, in order to carry out experiments on real roads with tyres under real vehicles instrumenting the tyre seems to be more desirable. The developments in the field of Smart Tyres make it possible to monitor tyre interior pressure and temperature using wireless systems. However further research is needed in order to develop wireless measurement systems with sensors on the tyre tread that can measure the contact forces between tyre and road. It is therefore highly recommended to explore both the possibility of instrumenting the road and the possibility of instrumenting the tyre.
6 References


3. Fernando E.G., Musani D., Park D. and Liu W., “Evaluation of Effects of Tyre Size and Inflation Pressure on Tyre Contact Stresses and Pavement Response”, Research Report 0-4361-1, Texas Transportation Institute, Texas A&M University, College Station, TX, 2005.


7 Appendix A: Data related to the STUVA test set-up

A.1 Technical specifications of the STUVA test set-up

STUVAtec Mathias-Brüggen-Str. 41
Studiengesellschaft für
unterirdische Verkehrs-anlagen mbH Köln, den 25.08.2006
Köln, 50827 Köln
0000-KEAP-032

Test circuit

(1) air spring
  2 pieces per wheel
  type: SP 2 B 12 R
  manufacturer: Phoenix AG, Hamburg
  volume: 5,200 cm³ (per air spring)
  pressure: max. 7 bar
  resonance frequency: 1.8 Hz

(2) shock absorber
  2 pieces per wheel
  type: EM 106891
  manufacturer: Fichtel & Sachs AG, Schweinfurt

(3) tyre
  type: XZA; XZY
  manufacturer: Michelin
  size: 445/65 R 22.5
  pressure: 7.5 bar

(4) weight
  a) wheel: 1,000 kg per wheel (approx.)
  b) axle: 8,000 kg
A.2 Simplified 2D model of the load-axle-tyre system

The 2D model of the STUVA set-up is depicted in Figure 34. In this model the tyres are modelled as a mass-spring system of mass $m_t$ and stiffness $t_k$. The suspension is modelled as a spring-damper system of constants $k$ and $b$ respectively. The axle and the load are modelled with their mass and inertia moments: $M_1$, $J_1$ for the axle and $M_2$, $J_2$ for the load.

In Figure 34 $r_1$ and $r_2$ are the road inputs given by the height profile of the test track. The dynamic equations of this system are:

$$
\begin{bmatrix}
0 & 0 & 0 & 0 \\
0 & 0 & 0 & 0 \\
0 & 0 & 0 & 0 \\
0 & 0 & 0 & 0
\end{bmatrix}
\begin{bmatrix}
\dot{y}_1 \\
\dot{y}_2 \\
\dot{y}_3 \\
\dot{y}_4
\end{bmatrix}
+ 
\begin{bmatrix}
b & 0 & -b & b \frac{L}{2} \\
b & 0 & -b & -b \frac{L}{2} \\
-k & -k & k & -k \frac{L}{2} \\
2k & 0 & 0 & 0
\end{bmatrix}
\begin{bmatrix}
\ddot{y}_1 \\
\ddot{y}_2 \\
\ddot{y}_3 \\
\ddot{y}_4
\end{bmatrix}
= 
\begin{bmatrix}
k + k_1 & 0 & -k & -k \frac{L}{2} \\
0 & k + k_2 & -k & -k \frac{L}{2} \\
-k & -k & 2k & 0 \\
k \frac{L}{2} & -k \frac{L}{2} & -k \frac{L}{2} & 0
\end{bmatrix}
\begin{bmatrix}
y_1 \\
y_2 \\
y_3 \\
y_4
\end{bmatrix}
+ 
\begin{bmatrix}
r_1 \\
r_2
\end{bmatrix}
$$

The parameter values used in the calculations are summarised below:

<table>
<thead>
<tr>
<th>$m_t$</th>
<th>$k_1$</th>
<th>$k$</th>
<th>$b$</th>
<th>$M_1$</th>
<th>$J_1$</th>
<th>$M_2$</th>
<th>$J_2$</th>
</tr>
</thead>
<tbody>
<tr>
<td>1e3 kg</td>
<td>1e6 N/m</td>
<td>5.5e5 N/m</td>
<td>1.2e4 Ns/m</td>
<td>5e3 kg</td>
<td>4.17e4 kg.m$^2$</td>
<td>3e3kg</td>
<td>4.17e3 kg.m$^2$</td>
</tr>
</tbody>
</table>

An eigenvalue analysis of the above system has been performed in order to determine the natural frequencies and corresponding modeshapes. The natural frequencies are summarised in the table below:
The estimated natural frequencies have been provided by the engineers at STUVAtec. The modeshapes that correspond to the above frequencies are plotted in Figure 35. The dotted lines indicate the position of the system at rest.

\[
\begin{array}{|c|c|c|}
\hline
\text{Mode number} & \text{Natural frequency SimMechanics (Hz)} & \text{Estimated natural frequency (Hz)} \\
\hline
1 & 1.47 & 1.5 \\
2 & 2.89 & - \\
3 & 6.37 & - \\
4 & 6.79 & 6.2 \\
\hline
\end{array}
\]

It has already been mentioned in chapter 2 that the first mode has a significant influence on the force variations. It can be clearly seen in Figure 35(a) that large vertical displacements of the axle+load occur at the first resonance frequency of 1.47 Hz.
A.3 SimMechanics model of the test set-up

The SimMechanics model of the STUVA set-up has been developed by Dr. ir. Antoine Schmeitz, who has participated in the LOT project until October 2007.
A.4 Original base ground of the STUVA set-up

Figure 36: Original base ground. (a) Measured profile (mm), (b) Vertical force variations (kN).
Appendix B: Additional information on measured 3D contact forces

B.1 TireView and the VRSPTA system

Figure 37: TireView Main Menu.

Figure 38: Schematic view of the VRSPTA system.
B.2 Fuji Prescale Films

A Fuji Pre-scale Film consists of microcapsules filled with color forming material. An applied pressure, depending on the value of this pressure, will break these microcapsules. When a microcapsule is broken, the color forming material is released. This material reacts with a color developing material and the magenta color will be formed. The pre-scale films are designed with a so called Particle Size Control Technology. This means the microcapsules are layered and therefore the capsules will react to various degrees of pressure. The pressure sensitive films release the color forming material in a density corresponding to a specific level of applied pressure. Because of the Particle Size Control Technology, there exist seven different applicable pressure ranges for the Fuji Pre-scale Films (see Figure 39).

![Figure 39: Applicable pressure ranges for Fuji Pre-scale Films.](image)

The films are available in two possible formats. One format is based on two sheets. The other is based on a single sheet. The film types MS (Medium pressure) and HS (High pressure) are single sheet layered types. The other Fuji film types are two sheet layered types.

The single sheet film type consists of a polyester base, with a coated color developing layer on it. The micro encapsulated color forming material is layered on top of the film as can be seen in Figure 40(a). The double layered Pre-scale film is composed of an A film (made of a polyester base) which is coated with microcapsules filled with color forming material. The C film (also made of a polyester base) is coated with a color developing material. To use these films, make sure the coated surfaces are facing each other. Otherwise there will be no footprint (see Figure 40(b)).

![Figure 40: The two working principles of the Fuji Prescale Films.](image)
Appendix C: Data related to the FE model

C.1 Rubber material properties [8]

Table 1: Thermomechanical continuum input parameters of rubber block

<table>
<thead>
<tr>
<th>Characteristics</th>
<th>Units</th>
<th>Rubber</th>
</tr>
</thead>
<tbody>
<tr>
<td>Mooney-Rivlin coefficient $C_{10}$</td>
<td>N/mm²</td>
<td>0.258</td>
</tr>
<tr>
<td>Mooney-Rivlin coefficient $C_{01}$</td>
<td>N/mm²</td>
<td>0.459</td>
</tr>
<tr>
<td>WLF-shift factor $d_1$</td>
<td>–</td>
<td>43.1</td>
</tr>
<tr>
<td>WLF-shift factor $d_2$</td>
<td>–</td>
<td>509.4</td>
</tr>
<tr>
<td>Mass density $\rho$</td>
<td>kg/m³</td>
<td>1207</td>
</tr>
<tr>
<td>Thermal conductivity $\lambda$</td>
<td>W/(mK)</td>
<td>0.284</td>
</tr>
<tr>
<td>Heat capacity $c$</td>
<td>kJ/(kgK)</td>
<td>1.45</td>
</tr>
<tr>
<td>Thermal expansion coefficient $\alpha_T$</td>
<td>1/K</td>
<td>$189.4 \cdot 10^{-6}$</td>
</tr>
</tbody>
</table>

C.2 Sketch of the block for the 3D FE model

The dimensions in the above sketch are given in mm and the depth of the block is 11 mm.
C.3 Description of the Koutny model

The curvature of the tyre belt can be represented using a simple geometric model known as the Koutny model [9]. The Koutny model comprises three tangential circular arcs. One arc corresponds to the curvature in the upper part of the tyre ($R_{\text{upp}}$ in Figure 41) and two identical arcs ($R_{\text{trans}}$ in Figure 41) correspond to the curvature of the tyre when entering and leaving the contact patch. Knowing the effective rolling radius, the length of the contact patch and the invariable length of the belt, the values of $R_{\text{upp}}$ and $R_{\text{trans}}$ can be calculated.

![Figure 41: Koutny model [9].](image)

The SimMechanics model of the STUVA set-up has been used to determine the contact length and effective rolling radius of the Goodyear tyre 425/65R22.5 G286A 165KL TL for a load of 50 kN and an inflation pressure of 8.5 bar. These values and the corresponding values of $R_{\text{upp}}$ and $R_{\text{trans}}$ are summarised in the table below (all values are given in meters).

<table>
<thead>
<tr>
<th>Contact Length</th>
<th>Effective rolling radius</th>
<th>$R_{\text{upp}}$</th>
<th>$R_{\text{trans}}$</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.292</td>
<td>0.526</td>
<td>0.559</td>
<td>0.307</td>
</tr>
</tbody>
</table>

The data from Table 5 can be used to determine the velocity of a tread block as it travels along the belt. Given the travelling velocity of the tyre, 80 km/h, the rotational velocity can be found to be 0.79 rad/s. The variation of the tyre radius along the circumference can be determined from the Koutny model and combining this with the knowledge of the rotational velocity leads to the velocity profile given in Figure 20(b).
Appendix D: Progress reports

D.1 Progress report LOT: 01/09/06 to 01/11/06

1. Introduction

In this report the work done at the TU Eindhoven between the 1st of September 2006 and the 1st of November 2006 is summarized.

As a result of the discussion about the STUVA test set-up at the kick-off meeting of 23rd Augustus 2006, the TU Eindhoven agreed to build a multi-body model of the test set-up in order to analyze the influence of the track layout on the contact forces. Since this was not included in the original timetable, an update of the timetable is included in section 0 (the changes are marked in red). Due to this additional task there is a small delay on the tasks Model rubber profile & contact model and Validation: Normal forces.

2. Dynamic analysis of the STUVA test set-up

A multi-body model of the STUVA test set-up has been built based on the information provided by STUVA. This model has been used to analyze the influence of track non-uniformities and unevenness on the contact forces. The results of this analysis indicate that the discontinuities between track sections have a relatively little importance compared to the unevenness of the track. Large contact force variations (above 10 kN) can occur for relatively small net height differences (40 mm) along the track.

A working document with the main results was sent to the related partners the last week of September and guidelines for the manufacturing of the track sections were derived together with the TU Delft. The practical consequences of these guidelines were discussed during the meeting of 12 October 2006 at Heijmans and the manufacturing strategy was established. In view of the potentially large influence of the track unevenness on the contact forces, it was also decided to measure the track profile after the track sections are installed and to perform a new analysis in order to determine the contact force variations in the actual test situation.

In the meeting of 12 October 2006 it was also decided to measure the profile of the base ground of the track of the STUVA set-up. The measured profile sent by the RWTH Aachen on the 25th of October 2006 has been analyzed with the multi-body model of the set-up. The results show large contact force variations. The impact of these findings has still to be discussed with the TU Delft.

In order to have a better view of the STUVA set-up a visit to the set-up has taken place at the 3rd November 2006. During this visit it became clear that the information provided by STUVA is based on estimations and not on actual measurements of the dynamic behavior of the set-up. This introduces additional uncertainties for the prediction of the variation of the contact forces. The significance of these uncertainties has to be evaluated together with the TU Delft.

3. Model of rubber profile/contact modeling

A 2D Finite Element (FE) model of a rubber block has been build and boundary conditions have been defined according to the loading conditions of 850 kPa inflation pressure and 50 kN load. The dimensions of the block are 70mmx15mmx40mm (length x height x depth) and have been derived from the geometry of the tyre chosen for this project (Goodyear 425/65R22.5 G286).

The material model used for the rubber compound is the Mooney-Rivlin model. At this point, the material properties of the rubber have been taken from the literature. A Coulomb friction model has been used for the longitudinal contact forces with a unit friction coefficient.
As a first step a stationary situation is considered and the block is pressed against a flat surface with a uniform load. After equilibrium is reached a displacement is applied at the top of the block to study the influence of slip on the contact force distribution. The preliminary results show that large peaks arise at the leading edge for both the normal and the longitudinal forces.

In addition to the above described work, the TireView software (from the Texas A&M University) has been installed and the 3D contact force distribution for the Goodyear 425/65R22.5 tyre has been calculated for the desired load and inflation pressure.

In the coming weeks the friction model will be further developed and the model will be extended to 3D. The predicted normal forces will be compared to the information from TireView.

4. Updated timetable

<table>
<thead>
<tr>
<th>Model feature</th>
<th>Sep 06</th>
<th>Oct 06</th>
<th>Nov 06</th>
<th>Dec 06</th>
<th>Jan 07</th>
<th>Feb 07</th>
<th>Mar 07</th>
<th>Apr 07</th>
<th>May 07</th>
<th>June 07</th>
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<tr>
<td>Model rubber profile &amp; contact model</td>
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<td>✔️</td>
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<tr>
<td>Validation: Normal forces. Stationary</td>
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<tr>
<td>Validation: 3D forces. Stationary</td>
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<td></td>
<td></td>
<td>✔️</td>
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<tr>
<td>3D contact stress distribution for 4/8 mm asphalt</td>
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<td>✔️</td>
<td>✔️</td>
<td>✔️</td>
</tr>
</tbody>
</table>
1. Introduction

In this report the work done at the TU Eindhoven between the 1st of November 2006 and the 1st of May 2007 is summarized.

2. Dynamic analysis of the STUVA test set-up

In January 2007 the heights of the 16 test plates and the profile of the base ground has been sent by the RWTH Aachen to the TU Eindhoven. Several different plate/base ground combinations have been simulated in order to find the optimal plate configuration that will originate the lowest contact force variations. Based on these calculations a definitive advice on the optimal track lay-out has been issued on the 17th of January 2007.

After the plates were laid the final track profile has been measured by RWTH Aachen and sent to TU Eindhoven on the 4th of February 2007. The simulations with this profile show that the average force variations on each plate are below 10% of the nominal load. According to the TU Delft this is acceptable.

3. 3D Contact forces

A 3D Finite Element (FE) model of one block of the tread profile of the Goodyear tyre 425/65R22.5 G286A 165KL TL has been built. The material model used for the rubber compound is the Mooney-Rivlin model. At this point, the material properties of the rubber have been taken from the literature. A Coulomb friction model has been used for the longitudinal contact forces with a unit friction coefficient. The boundary conditions have been defined in two steps. In the first step the block travels across the contact patch in quasi-static (v=1.5 km/h) conditions. These results can be compared to the data from the TireView software to fit the model parameters.

In the second step the block travels at 80 km/h and the force distribution can be calculated. This task is not completed yet. Additionally, the variation of contact forces due to turn-slip has been addressed using the multi-body model developed for the STUVA test set-up. Turn-slip occurs specifically at this set-up because the tyres are forced to ride in a circle. The increase in turning moment and lateral forces due to this effect has been calculated and the contact forces will be modified using this information.

In the months February and March 2007 static contact pressure measurements have been carried out using FUJI Pre-scale films on a flat surface and on asphalt. Comparison of the two footprints shows that the effective contact area decreases dramatically on asphalt, as expected, and that the peak pressure values increase significantly.

4. General matters

In January 2007, Prof. Anghelanghe from the Politechnique University Bucharest has visited the TU Eindhoven. On of his main research topics is the experimental determination of 3D tyre/road contact forces and FE modeling of truck tyres.

In March 2007 the TU Eindhoven has attended the general LOT meeting and presented the general line of its contribution to the LOT project. The presentation can be found in the CDrom “LOT Meeting. March 22, 2007” issued by the TU Delft.

5. Updated time schedule

The expected time schedule presented in the meeting of 22 March has been delayed one month.

- 31 may 2007: time history of 3D contact forces
- 30 june 2007: report
Prediction of 3D contact forces
Progress TU Eindhoven

Dr. Ir. Ines Lopez

Dynamics and Control Group (D&C)
Department of Mechanical Engineering
Eindhoven University of Technology

Contents

• Status project
• Dynamic analysis of STUVA set-up
• 3D contact forces
• Time Schedule
Status project: Past work

- Preliminary results dynamic analysis STUVA set-up (September 2006).
- Antoine Schmeitz left TUe. (October 2006)
- Presentation of results dynamic analysis. Meeting M. Huurman, C. Gharabaghy, G. Bochove, I. Lopez. (October 2006)
- On site evaluation of test set-up. Visit C. Gharabaghy, I. Lopez to STUVA. (November 2006)
- Definitive advise on optimal track lay-out. (January 2007)
- Last calculations on definitive test-track. (February 2007)
- Static contact pressure measurements on flat surface. (February 2007)

04/07/2007

Status project: Ongoing work

- FE model of tire profile (March 2007)
- Fit model parameters to Tireview data (April 2007)
- Calculation of contact forces at 80 km/h (April 2007)
- Static contact pressure measurements on asphalt (April 2007)

04/07/2007
Dynamic analysis of STUVA set-up: Description

04/07/2007

Dynamic analysis of STUVA set-up: Simulations

04/07/2007
Dynamic analysis of STUVA set-up: Results

Final track profile: zmax-zmin=12 mm  Vertical load : 13% variation

04/07/2007

Dynamic analysis of STUVA set-up: Conclusion

Average vertical load per plate : <10% of nominal load

04/07/2007
3D Contact forces: Approach

- Measured data in quasi-static conditions (Tireview)
- Qualitative analysis based on
  - Finite Element model
  - MultiBody model
  - Literature
- Modify force distribution to include:
  - Effect of travelling velocity (80 km/h)
  - Effect of turn-slip
3D Contact forces: FE model

Contact pressure

04/07/2007

3D Contact forces: Boundary conditions

Quasi-static (v=1.5 km/h)  v=80 km/h

04/07/2007
3D Contact forces: Fit parameters (quasi-static)

INPUT

- Material data (lit.)
- Boundary conditions

Parameters → FEM → 3D Forces → Tireview

3D Contact forces: 80 km/h

INPUT

- Parameters
- Material data (lit.)
- Boundary conditions

FEM → 3D Forces
3D Contact forces: Turn-slip

Tires forced to drive on a circle

Slip velocities on contact patch

04/07/2007

3D Contact forces: Turn-slip

Approach:
- Estimate turn-slip moment (Mz) from MultiBody model.
- Modify force distribution to have the same resultant moment

Resultant forces acting on road

04/07/2007
Time Schedule

- 30 April 2007: Time history of 3D contact forces