FE analyses of Pin on Disc tests to analyze hybrid CVT behavior

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FE analyses of Pin on Disc tests
to analyze hybrid CVT behavior

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<thead>
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<th>Symbol</th>
<th>Description</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>(A)</td>
<td>area</td>
<td>([mm^2])</td>
</tr>
<tr>
<td>(d)</td>
<td>displacement</td>
<td>([m])</td>
</tr>
<tr>
<td>(E)</td>
<td>energy</td>
<td>([N])</td>
</tr>
<tr>
<td>(F_{\text{app}})</td>
<td>applied force</td>
<td>([N])</td>
</tr>
<tr>
<td>(F_{\text{CoF}})</td>
<td>friction force</td>
<td>([N])</td>
</tr>
<tr>
<td>(F_{\text{net}})</td>
<td>netto force</td>
<td>([N])</td>
</tr>
<tr>
<td>(F_T)</td>
<td>normal force</td>
<td>([N])</td>
</tr>
<tr>
<td>(N)</td>
<td>normal force</td>
<td>([N])</td>
</tr>
<tr>
<td>(P)</td>
<td>power</td>
<td>([W])</td>
</tr>
<tr>
<td>(P_r)</td>
<td>pressure</td>
<td>([N/mm^2])</td>
</tr>
<tr>
<td>(r)</td>
<td>radius</td>
<td>([m])</td>
</tr>
<tr>
<td>(t)</td>
<td>time</td>
<td>([s])</td>
</tr>
<tr>
<td>(T)</td>
<td>torque</td>
<td>([Nm])</td>
</tr>
<tr>
<td>(v)</td>
<td>speed</td>
<td>([m/s])</td>
</tr>
</tbody>
</table>

### Greek symbols

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>(\mu)</td>
<td>coefficient of friction</td>
<td>([-])</td>
</tr>
<tr>
<td>(\mu_d)</td>
<td>dynamic coefficient of friction</td>
<td>([-])</td>
</tr>
<tr>
<td>(\mu_s)</td>
<td>static coefficient of friction</td>
<td>([-])</td>
</tr>
<tr>
<td>(\omega)</td>
<td>radial velocity</td>
<td>([\text{rad/s}])</td>
</tr>
</tbody>
</table>
Chapter 1

Introduction

At the University of Doshisha there is a dry hybrid v-belt test rig. A dry hybrid V-belt is also known as the rubber CVT-belt. The belt is of the pull belt type. It is build up out of 2 rippled rubber belts with aramide reinforcement cords at the center of the belts in the longitudinal direction. Between these belts the blocks are positioned. The ripples keep each block at a fixed position. The blocks are made of a core of an aluminum alloy covered with a phenolic resin coating, see figure 1.1. The CVT belt is made by BANDO [2]. The field of application of these belts is the so called Kei-car. A Kei-car is a small car with a relative small engine. The belt is capable of transmitting relative small powers. These belts have to last for approximate 100,000 [Km].

![Figure 1.1: The hybrid V-belt of BANDO.](image)

The present CVT system using the hybrid V-belt has a problem "noise". The V-belt is cooled with fresh air taken from the atmosphere. An open inlet and exit exist where noise is emitted. If these holes are closed, another problem occurs, i.e., heat. The hybrid V-belts are fabricated with some synthetic materials, polymers and elastomers which cannot survive at elevated temperature. In order to simultaneously solve both problems, the belts are going to be lubricated with oil since the today's technology has invented rubber belts which can survive under the fossil oil lubricant condition. However, another problem appeared. The transmittable torque decreased almost 80% as compared with the dry condition. The CoF (Coefficient of Friction) between the belt and pulleys decreased enormously. With TDF-fluid slightly more power could be transmitted than with ATF-fluid.

In order to raise the CoF a surface treatment was applied at the pulley surfaces. The pulleys were sandblasted and micro-sandblasted. Micro-sandblasting is desired over normal sandblasting, because the wear of the belt is smaller for micro-sandblasting. In combination with the ATF-fluid there are no big changes. On the other side, with TDF-fluid there is. The transmittable torque is not at the desired level, but closer to the desired level.
Research is done on the CoF of these working conditions with PoD-tests (Pin on Disc). In this report the outcomes of this research is presented and compared with the FE-analysis (Finite Element) of the PoD-tests. There may be a relation between the pressure distribution between the pulley and belt and the transmittable torque. Probably the two materials of the pin are debet to this different pressure distribution.

In order to make conclusions on and the relation between the transmittable torque and the pressure distribution the fine meshes are first adapted in the rigid pulley CVT model. Then the ideal pressure distribution is known on the tops of both arms of the block. Next it will be implemented in the model with rigid pulleys as well, but these are able to tilt. The next step is to apply this fine mesh into the model with deformable pulleys. And at last in the model with deformable and tilting pulleys. In this way a complete and clear view can be presented about the influence of the pressure distribution on the transmittable torque. This part has not been completed, because there was no time left to do it. This has to be done in the future.

Also a small modification is made for the V-belt in order to solve the main problems with the noise. A modification will be made and the noise level and transmittable torques are compared of the treated and untreated belt. In order to solve the noise problem. The idea of removing a part of the edge of the rubber belt can be used to compare the transmittable torque with the pressure distribution in the PoD-tests. The complete edge will be removed in order to be able to do this.
Chapter 2

Pin on Disc Simulations

When surfaces are in contact they usually transmit shear as well as normal forces across their interface. There is generally a relationship between these two force components. This relationship, known as the friction between the contacting bodies, is usually expressed in terms of the stresses at the interface of the bodies. With a PoD test rig the CoF between two materials can be determined. In this case the test disc will consist of the pulley material, with treated and untreated surfaces and the pin consist of a part of the block that is used in the rubber V-belt. The configuration of this setup is given in figure 2.1.

![Figure 2.1: The PoD test machine.](image)

The pin is made out of the upper arm of a block and only half of it will be used, see figure 1.1 and 2.1. A special clamp will hold this block in the PoD test machine. The normal force applied in the PoD-test is 100 [N]. In the CVT test rig the normal force on a single block is between 0 en 300 [N]. The disc will rotate with 1 [rpm]. The radius on which the pin will be acting on the disc is 18.4 [mm] and the sampling time is 1 [Hz]. The total time of one run is 30 minutes. The slip, which occurs in this test is infinite and is much larger than the slip in CVT test rig. In the CVT
test rig the slip between pulley and belt should be zero. These tests can be done with and without lubrication. When the tests are finished, the results are compared with those of the simulations in ABACUS. If they are not corresponding, the ABACUS-model has to be adjusted till the results of the simulations are almost corresponding with the real PoD tests. Then a link can be made with the previous made FE-model in ABACUS of the CVT [1].

2.1 Theory behind the PoD test machine

Before modeling an ABAQUS-model one has to understand what the theory is behind the PoD-test. Therefore the Coulomb friction model theory will be explained. The real CoF consists of two separated CoF's, namely the static and dynamic CoF. They have both their own region in which they are valid. When there is no slip (no movement between the contacting surfaces), the static CoF is valid. When the slip is infinite the dynamic CoF is valid, see figure 2.2. In between there is transition region. The transition from static to dynamic friction results in a small startup error. Then there is no linear behavior. After that the dynamic CoF becomes constant. In case of a fixed normal force, the reaction force of the static CoF is not constant. It is dependent on the magnitude of the applied force and can be calculated with equation 2.1, it has a linear behavior. The transition from the static to the dynamic friction has in the beginning a small startup error and then becomes constant and can be determined with the same equation, equation 2.1. The net force, which is available for the movement can be calculated with equation 2.2 [3].

\[ F_{CoF} = N \cdot \mu \]  
\[ F_{net} = F_{app} - F_{CoF} \]

In figure 2.3 the forces that are acting in the PoD-test are presented. At the top of the pin, a force is applied, normal force \( N \). The disc is rotating with a fixed speed \( \omega \). Due to the normal force, the angular speed and the CoF a reaction force will occur. This reaction force is the friction force \( F_{CoF} \).

The force that is applied at the disc surface, \( F_{app} \), is the opposite of this force and is \(-F_{CoF}\). The torque force, \( F_T \), is the same as the applied force, \( F_{app} \), equation 2.2. This is force is larger than the friction force and can be compared with the applied force, equation 2.3. The force that is acting on the disc is dependent of the radius on which the pin is acting, equation 2.4. The power needed to drive the disc is the torque multiplied with the the angular velocity, equation 2.5.
Chapter 2. Pin on Disc Simulations

Figure 2.3: Forces acting in a Pin on Disc test.

\[ F_T = F_{app} \]  
\[ T = F_T \cdot r \]  
\[ P = T \cdot \omega \]  

2.2 Real pin wear

Real pin wear can be divided in four phases. In phase one the pin bends and only a small part of the pin has real contact with the disc. There has been no wear. In the second phase the pin has been worn a little bit and a larger part of the pin surface has contact with the disc. In the third phase, the surface has been wearing more. The complete surface of the pin has contact with the disc, but there is not yet steady state behavior. After a while the pin has been wearing more and the surfaces are polishing each other. This phase is called the steady wear phase, and the pressure is more or less constant in time. These four phases are represented in figure 2.4. The arrow in the figure denotes the direction of movement of the disc. During the POD tests a little bit of pin will be wear off the pin. The pressure distribution shown in this figure is the behavior that occurs when one material is used for the pin. The pressure distribution due to the two different materials of the block will be different.

Figure 2.4: Behavior of the pin in wear process: (1) First contact, (2) edge-like contact, (3) full contact phase, (4) steady wear phase.
Real pin wear can be modeled within ABAQUS. The reason that this is not done is because two simulation licenses have to be shared with all the laboratory members of the CVT and Bamboo lab. Everybody has to do simulations and simulations of this type of behavior will take a large amount of time on the computers and that is simply not possible. Only the first phase will be modeled, [6] and [7]. In the real CVT-model wear is not modeled as well. In order to be able to compare the data of these two models, wear should not be included in the POD simulation tests.
Chapter 3

The ABAQUS-Model

ABAQUS (version 6.5-4) is a finite element program, which is used to do the simulations of the PoD-tests. There are three different stages in this program, to make, calculate and analyse the model. In the preprocessing stage the physical model is made. ABAQUS/CAE is used to do this. The real simulation is done in ABAQUS/standard. This runs as a background process and does the real numerical calculations. After the simulation is finished the results can be monitored in the postprocessing phase. ABAQUS/CAE is also used to view, plot and process the data. The program can draw the data output, but to be able to process it in an easier and more structured way, MATLAB (version 7.1) is used.

3.1 The first model

It is possible to make an exact copy of the configuration of the PoD test, but this will require a very long CPU-time (Central Processing Unit time). This is not necessary to obtain good results. A simplified model will be made. The first model that is made contains as less elements as possible in order to make the simulation run very fast. Difficulties with the settings like materials, boundary conditions and loads can be solved easily. The main goal is to get the simulation running. After this step, the model will be refined in order to get more accurate results. A proper friction formulation will be chosen and the number of elements will be increased to be able to check the real pressure distribution between pin and disc, due to the different materials of the pin.

3.1.1 The part module

There are several ways to make or load parts into the part module. In ABAQUS there is a drawing feature, which makes it possible to draw the parts. Because the shapes of the parts are very easy to draw, this feature is used. Modeling is done in millimeters, so the other quantities have to be of the same order.

Modeling the pin

In figure 3.1 the real pin design is shown. The pin and clamp holder can be modeled in exactly that way in ABAQUS. This results in a very complicated structure and strange structured elements (in case of the used C3D8 elements, see section mesh). Again the CPU-time will be too large. It is better to only model the part of the pin that is extending the clamp and make proper boundary conditions for the top surface. The pin is extending about 5 millimeters. The tip is made square, with the same area as in the CVT-model that is available [1]. When a connection is made between the models, the contact surface area dimensions are the same, with similar contact behavior and probably pressure distribution. In this way it is easier to compare the two models.

First an intersection of the pin is drawn. It is drawn at the position in the co-ordinate system, where it is situated in the assembly. Then only one translation is needed to get the pin at its right position. The dimensions of the pin used in the FE-analysis are shown in table 3.1. In table 3.2,
The pin and disc dimensions.

The properties of this part are shown. After extrusion of the intersection the part is intersected in four equal parts. By intersecting the pin, extra nodes appear that are necessary to be able to apply the BC’s at the top surface.

The disc:
- Outer radius: 25.9 mm
- Inner radius: 10.9 mm
- Height: 5 mm
- Thickness (height): 7 mm
- Thickness (depth): 2.56576 mm

Table 3.1: Dimensions of the pin and disc in the ABAQUS-model.

<table>
<thead>
<tr>
<th>The disc:</th>
<th>The pin:</th>
</tr>
</thead>
<tbody>
<tr>
<td>Outer radius</td>
<td>Width</td>
</tr>
<tr>
<td>Inner radius</td>
<td>Height</td>
</tr>
<tr>
<td>Thickness (height)</td>
<td>Thickness (depth)</td>
</tr>
<tr>
<td>25.9 mm</td>
<td>2.95 mm</td>
</tr>
<tr>
<td>10.9 mm</td>
<td>5 mm</td>
</tr>
<tr>
<td>7 mm</td>
<td>2.56576 mm</td>
</tr>
</tbody>
</table>

Modeling the test disc

Figure 3.1 is also used to determine the dimensions of the disc. The test disc is a circular disc with flat sides at a part of the outer radius. These flat sides are used to clamp the disc species onto the the PoD test machine. When the disc is designed in its original way in ABACUS, these flat sides results in in homogenous elements, in case of the used square elements. By making the disc circular all the elements have the same shape. At the center a hole is made for the same reasons. Both discs are presented in figure 3.2.

The center of the pin acts at a radius of 18.4 mm at the disc. This is exactly the middle between the inside and outside radius of the disc. The track on which the pin can run is 15 millimeters wide. Out of the first results follows that this is more then wide enough to prevent stresses and deformations at the sides of the disc, since the Young’s modulus of the disc is much larger than the Young’s modulus of the pin. The thickness of the disc is also sufficient. The
Chapter 3. The ABAQUS-Model

19

3.1 The ABAQUS-Model

(a) Real disc.

(b) Simplified disc.

Figure 3.2: The test discs in ABAQUS, with and without a center hole and the arrangement of the elements with equivalent mesh size.

dimensions of the pin and the disc used for the ABAQUS-model are presented in table 3.1. A sketch is made of the intersection of the body. This intersection is revolved around the y-axis for 360°. The settings for the disc are presented in table 3.2. Next the disc is divided in four sections of 90°. The model will not make a full circular movement, it will only rotate at most for 45°. Within this region steady state behavior occurs and more data is not needed. This is also done in order to get a short CPU-time. A fine mesh will only be made for the partition of the disc which has physical contact with the pin surface. To be able to control the movements of the disc a RP (Reference Point) is constructed in the part module. Later on in the interaction and load module several constraints are assigned to this RP. It is positioned 25 [mm] below the bottom surface of the disc. In this way the program runs more stable.

<table>
<thead>
<tr>
<th>Part Name</th>
<th>Modeling Space</th>
<th>Type</th>
<th>Base Feature</th>
</tr>
</thead>
<tbody>
<tr>
<td>Pin</td>
<td>3D</td>
<td>Deformable</td>
<td>Solid</td>
</tr>
<tr>
<td>Disc</td>
<td>3D</td>
<td>Deformable</td>
<td>Solid</td>
</tr>
</tbody>
</table>

Table 3.2: Part properties.

3.1.2 Property module

In the property module the different material properties and body sections are assigned to specified regions of the pin and the disc. The disc consists of one material and one section. The pin consist of two materials. A core of an aluminium alloy and a resin coating covering the aluminum alloy. In the first model the pin has as only one section and consist only of the aluminum alloy. In table 3.3 the different material properties are presented.

<table>
<thead>
<tr>
<th>Section Name</th>
<th>Material Behavior</th>
<th>Type</th>
<th>Young’s Modulus [N/mm²]</th>
<th>Poisson’s Ratio [-]</th>
<th>Section Type</th>
</tr>
</thead>
<tbody>
<tr>
<td>Pin Core</td>
<td>Mechanical, Elastic</td>
<td>Isotropic</td>
<td>185000</td>
<td>0.33</td>
<td>Solid/Homogenous</td>
</tr>
<tr>
<td>Disc</td>
<td>Mechanical, Elastic</td>
<td>Isotropic</td>
<td>210000</td>
<td>0.30</td>
<td>Solid/Homogenous</td>
</tr>
</tbody>
</table>

Table 3.3: Material properties.
3.1.3 Assembly module

A new co-ordinate system is created in this part. The pin and disc will be loaded in this co-ordinate system and will be assigned to this co-ordinate system. First the disc part is loaded and positioned at the center of the co-ordinate system. It is rotated counterclockwise for $15^\circ$. This rotation is made to be sure that the pin touches the fine mesh part of the disc when it starts rotating. The pin itself is moved backwards in the $3^{rd}$-direction for 1.28288 [mm], which is the thickness of the pin, to align its center line, at the line of the radius of the disc.

In the assembly module it is also possible to create sets of lines, nodes and surfaces. These are useful to make it easier to select lines, nodes and surfaces when refining the mesh or assigning surfaces when the number of them arises and it becomes hard to select them separately each time. With really fine meshes it is almost impossible to select every single object.

3.1.4 Step module

The step module is used to create different steps for analyzing. It is useful to make a single steps for each change in the model. In every step can be prescribed what the output parameters have to be. Also the length of the step, the actual simulation time, can be set. As well as the start, minimum and maximum value of a single increment and the maximum number of increments.

In the initial step the model is checked or it is modeled in a correct way and if the output requests that are requested for each step are possible. In the first step the force is applied at the top surface of the pin. A ramp function is used to apply the force. In the second step this force is maintained and the disc starts to rotate with a prescribed speed. The length of this step determines the length of the path at the disc. The settings are presented in table 3.4.

<table>
<thead>
<tr>
<th>Step</th>
<th>Step name</th>
<th>Procedure</th>
<th>Time [s]</th>
<th>Maximum number of increments</th>
<th>Increment size</th>
</tr>
</thead>
<tbody>
<tr>
<td>0</td>
<td>Initial</td>
<td>Initial</td>
<td>N/A</td>
<td>N/A</td>
<td>N/A</td>
</tr>
<tr>
<td>1</td>
<td>Apply Pin Force</td>
<td>Static, General</td>
<td>1</td>
<td>10</td>
<td>0.1, 0.1, 1</td>
</tr>
<tr>
<td>2</td>
<td>Rotate Disc</td>
<td>Static, General</td>
<td>5</td>
<td>100</td>
<td>0.1, 0.001, 1</td>
</tr>
</tbody>
</table>

Table 3.4: Step properties.

3.1.5 Interaction module

Interaction of the surfaces

In order to let the model run well a friction formulation is prescribed for the contact between the pin and the disc surfaces. There are a few models to prescribe the friction between the pin and disc. In the first model a penalty of 0.15 [-] will prescribe the friction, because this is the average CoF out of the results of the PoD-tests. In the beginning the friction in both directions ($1^{st}$ and $3^{rd}$-direction) is the same. For the next model, a thorough research on the different friction models will be given. In table 3.5 the initial friction conditions are given.

<table>
<thead>
<tr>
<th>Interaction</th>
<th>Contact property</th>
<th>Friction formulation</th>
<th>CoF</th>
</tr>
</thead>
<tbody>
<tr>
<td>Pin surface/disc surface</td>
<td>Tangential behavior</td>
<td>Penalty</td>
<td>0.15</td>
</tr>
</tbody>
</table>

Table 3.5: Interaction properties.

Disc rotation

In order to be able to control the movements of the disc a coupling has to be prescribed. With the RP assigned in the part module it is possible to control the disc by one single point, see
Table 3.6. The RP is set as the control point and the bottom surface of the disc is the controlled region. The complete surface is selected because when only the outer region is selected the results will be influenced by a the bending of the disc, which is neglected because it is not appearing. Now every single point on the surface has the same BC's instead of only the outer region. A kinematic coupling is chosen instead of a distributing coupling. In case of a kinematic coupling all the DOF (Degrees Of Freedom) at the coupling nodes are eliminated and the coupling nodes will be constrained to move with the rigid body motion of the reference node. In case of a distributing coupling, the DOF are not eliminated, but are enforced in a average sense. This means that the resulting forces at the coupling nodes are equivalent to the forces and moments at the reference node and the force and moment equilibrium of the distributed loads about the reference point is maintained, [5].

<table>
<thead>
<tr>
<th>Constraint</th>
<th>Constraint type</th>
</tr>
</thead>
<tbody>
<tr>
<td>Disc / RP</td>
<td>Coupling</td>
</tr>
</tbody>
</table>

Table 3.6: Constraint properties.

3.1.6 Load module
boundary conditions

To be able to simulate the clamping of the clamp holder it is important that the BC's (Boundary Condition) are chosen in a proper way. The BC's should not generated extra stresses in the pin and it should be able to apply an force on the top. Using a coupling as for the disc is not possible because then the material is not able to move due to its poison ratio. Therefore the center node movements in the 1st and 3rd direction are suppressed and movement in the 2nd direction is free. The rotations around this vector is free as well. They are suppressed by the boundary conditions at the central partition lines. At the line in the direction of the rotation only the 1st direction is fixed and all the others are free. For the center line from the inner to the outer radius the 3rd direction is fixed. All these BC's on the top surface together make sure that the surface is able to move towards the disc in the 2nd and that the surfaces of the pin and disc stay aligned. The BC's make it also possible that the material can act to its poison ratio. So the material is able to deform. See figure 3.3.(a) and table 3.7.

Figure 3.3: The boundary conditions of the pin and the disc.

The previous prescribed constraint of the disc is used in the load module. An angular velocity of 1 [rpm] is prescribed for the RP around the 2nd-direction axis. The other movements, rotations
and translation are suppressed. This should be done because there is no backlash at the disc shaft and in the clamp holder part as well. In figure 3.3.(b) the boundary condition of the disc is shown. The purple surfaces is the controlled surface. The red dot below the disc is the RP.

<table>
<thead>
<tr>
<th>Boundary condition</th>
<th>DOF’s</th>
<th>Initial step</th>
<th>Step 1</th>
<th>Step 2</th>
</tr>
</thead>
<tbody>
<tr>
<td>Pin Central node (dot)</td>
<td>U1</td>
<td>Inactive</td>
<td></td>
<td>Propagated</td>
</tr>
<tr>
<td></td>
<td>U2</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>U3</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>UR1</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>UR2</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>UR3</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Pin Front and back node (filled square)</td>
<td>U1</td>
<td>Inactive</td>
<td></td>
<td>Propagated</td>
</tr>
<tr>
<td></td>
<td>U2</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>U3</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>UR1</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>UR2</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>UR3</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Pin Left and right node (empty square)</td>
<td>U1</td>
<td>Inactive</td>
<td></td>
<td>Propagated</td>
</tr>
<tr>
<td></td>
<td>U2</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>U3</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>UR1</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>UR2</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>UR3</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Disc</td>
<td>V1</td>
<td>Inactive</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>V2</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>V3</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>VR1</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>VR2</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>VR3</td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

\[ V = \text{speed in [mm/s]} \]
\[ R = \text{translation in [mm]} \]
\[ \times = \text{fixed} \]
\[ \circ = \text{free movement} \]
\[ VR = \text{radial speed in [rad/s]} \]

Table 3.7: Boundary conditions.

load

There are two ways in which a force can be prescribed on the top surface. The first one is to make a force vector to act at a node. The second option is to let a pressure act on a surface. The second one is the best. The first option results in a non homogenous force distribution in the upper part of the pin. When a pressure is applied at the top surface the size of the surface is of no importance because the quantity is prescribed in force per unit area and the pressure is distributed equally over the complete top surface. The value out of table 3.8 is calculated out of equation 3.1.

In figure 3.3.(a) the applied pressure is only acting on the highest surface in the figure to keep the figure clear. In the first step of applying the pressure it can be seen that the pressure distribution is done in the right way, because the von mises stresses at the top surface have all the same color. This means that the stresses have the same value.

\[ Pr = \frac{N}{A} \]

\[ Pr = \frac{100}{2.95 \cdot 2.56576} = 13.21 \ [N/mm^2] \]
3.1.7 Mesh module

As already mentioned the used elements are of the type C3D8, see table 3.9. It is a continuum solid element. It has 8 nodes, at each corner 1. This is the best option, since the model is made in such a way that the elements are almost square in the contact region. This is needed in order to get good results. The element must be able to “feel” what is happening at the other side of the element. When the brick is long and narrow, this is not possible, and the obtained results will be not very reliable. An other option is the C3D6-element. This is a 6-nodes linear triangular prism. Verhoeven [1] has already researched this option and came to the conclusion that the C3D8-elements were the best option to do analyses with. Because for further research the outcome of this research is going to be implemented in Verhoeven’s model it is useful to use the same kind of elements. The comparison is easier when they are kept of the same type.

To be able to get the desired amount of elements on each edge the seed edge feature is used. The edges are selected as sets edges with the same amount of desired elements on each edge. The pin consist of 16 elements in total. 2 in the width and dept direction and 4 in the direction of the height. The elements are almost square.

The 1st part of the disc with the fine mesh has in the radial direction 7 elements, 4 in the direction of the height and 14 in the tangential direction. These elements have physical contact with the pin. The elements of this part are almost square, except of a small angle. The elements of this part of the disc have almost the same size as the pin elements. Then the calculations will run smoother and give more accurate results, because every node of the pin has contact with a corresponding node on the disc surface.

In order to reduce the number of elements the other 3 sections of the disc will contain less elements. They have no physical contact with the pin. Therefore it is possible to apply a lower number of elements. Because all the nodes of the each elements has to be connected to each other, the number of elements in the radial and height direction they have the same value (resp. 7 and 4). The number of elements in the tangential direction has a lower value, 9 elements. These elements have a long cuboid form.

<table>
<thead>
<tr>
<th>Part name</th>
<th>Element type</th>
<th>Element type description</th>
</tr>
</thead>
<tbody>
<tr>
<td>Disc</td>
<td>C3D8</td>
<td>8-node linear brick</td>
</tr>
<tr>
<td>Pin</td>
<td>C3D8</td>
<td>8-node linear brick</td>
</tr>
</tbody>
</table>

3.1.8 Job module

After finishing the physical modeling, the job can be submitted for analysis. The program will checked if all settings are correct. Also will be checked if the requested output variables are available for the model. The first jobs run on a computer with a small processor, so the memory the program is allowed to use is set on 256 Mb. Even though it is not a complicated large model so it runs fast. During analysis, the analysis can be checked with the monitor function. The progression of the analysis is shown. The settings for this module are shown in table 3.10.

3.1.9 Visualization module

In the step module the parameters have to be selected for the results. The parameters in table 3.11 are of importance in this analyses. The values that are computed in ABAQUS are verified
3.2. RESULTS OF THE FIRST MODEL

Looking at the animation of the simulation makes clear that the boundary conditions of the pin are chosen properly. There are no extra stresses generated around the boundary conditions at the top pin. In previous simulations, it was very clear when the BC’s were wrong. Stresses due to the BC’s could be clearly seen. When a plot is made of the nodes at the lower surface of the disc part of the 2nd-direction, no meaningful displacement occurs (order $1 \cdot 10^{-15}$ in the contact area). This means that the BC of the disc is chosen in the right way as well.

In the first step the pressure is nicely equally distributed, because the model has one color at each increment, the von Mises stresses are of the same magnitude in every point in the pin. This
is what is desired. A top view of the disc shows that there is symmetric spot of stresses in the disc around the pin. That it is not completely round is due to square form of the pin. An intersection of the disc shows that the thickness of the disc is chosen properly. Large stresses occur only in the upper part (the upper 3.5 millimeters) of the disc.

In the second step the von Mises stresses for the pin are similar in what can be found in [6]. Because real wear is not simulated it can be seen that due to the force at the top surface, high stresses occur at the front of the pin and low stresses at the back. It look as if the model is modeled in a right way, but to make sure it is a good model, the values have to be compared to analytical calculated values.

Also in the second step can be seen that at the inside of the pin (near the disc center) the stresses are higher than at the outside of the pin. When the disc rotates and the normal force is pushing at the top of pin, the pin tends to stick, together with the movement it causes a little bending of the pin. The front of the pin is pushed 'into' the disc and at the back it lifts up a little according to the front. Because the circular movement of the disc the stresses at the inside of the pin are higher then at the outside of the pin.

The disc stresses in the second step are floating a little bit. Again due to the rough mesh. The stresses at the front of the pin are higher then at the end of the pin. And the stresses near the center of the pin are higher than at the outer radius. According to what has been explained for the pin this is according to each other.

To make sure that the model is correct the different values of the quantities have to be checked. The values that will be checked are summarized below. An approach of what the values should be, are calculated with the equations presented in chapter 1. The prescribed values should have a certain value at a certain point in the simulation.

1. The contact pressure in the last increment of the first step should be equally to 13.21 [N/mm²]
2. The magnitude of the reaction forces should be around $F_{Cof} = N \cdot \mu = 100 \cdot 0.15 = 15$ [N], from equation 2.1
3. The rotational displacement of the nodes should be equally to what is prescribed, $\omega = 2 \cdot \pi / 60$ [s]

### 3.2.1 Contact pressure

To make sure that the contact pressure has the right values, the pressure has to be presented in a time, pressure figure. In the figure 3.5.(a) the bottom surface of the pin with the ABAQUS node numbers are displayed. These are numbers corresponding with the legend of the left figure 3.5.(b). What can be seen is that at t = 1 the value is 13.21 [N/mm²], what is desired.

### 3.2.2 Displacement

The displacement of the pin is also known. Because it is a square feature, the mean displacement at the center line on the disc is calculated. The displacement is calculated with 3.2 and is shown below. This is corresponding what can be seen in figure 3.6.(a).

$$d = r \cdot t \cdot v$$

$$d = 0.0184 \cdot 5 \cdot 2 \cdot \frac{\pi}{60} = 9.63 \text{ [mm]}$$

### 3.2.3 Frictional dissipation

The energy which is lost in heat due to the friction is called the frictional dissipation. The frictional dissipation is calculated with equation 3.3. In the equation below the real value is presented. In figure 3.6.(b). It appears that all the values are corresponding and that the PoD simulations are done in a right way and that the results are reliable.
3.2. RESULTS OF THE FIRST MODEL

(a) Von mises stresses and the node numbers of the bottom surface of the pin.  
(b) The pressure distribution at all nodes of the bottom surface of the pin.

Figure 3.5: The first pin on disc model.

(a) The displacement of the disc.  
(b) Frictional energy dissipation in the contact area.

Figure 3.6: The first pin on disc model results.

\[ E = N \cdot \mu \cdot d \]  \hspace{1cm} (3.3)

\[ E = 100 \cdot 0.15 \cdot \left(0.0184 \cdot 5 \cdot \frac{2 \cdot \pi}{60}\right) = 0.145 \, [J] \]

3.2.4 3-D pressure plot

To be able to make a clear comparison of the future models a 3-D plot of the pressure distribution between the pin and disc is made. Therefore the pressure at time step 6 is taken and plotted. The x and y-axis are the axis along the pin surface and the z-axis represents the pressure distribution. On the left side of the x-axis is the disc center. In the back of the plot the front of the pin is positioned (there occurs the highest pressure). This is done to make the figure clear. If all the other plots of the distance, dissipated energy and reaction forces are similar as for the first model, only this plot will be given for the comparison between the models. This plot presents the most important
(a) Reaction forces on the lower surface of the disc in the tangential direction.

(b) Reaction forces on the lower surface of the disc in the transverse direction.

Figure 3.7: The first pin on disc model results.

In figure 3.8 the pressure distribution of the first FE-analysis is shown. Clearly can be seen that near the center of the disc the pressure is higher ($y = 0$) and at the front ($x = 2.56$) the pressure is higher than at the back ($x = 0$). This is what is expected.

Figure 3.8: The pressure distribution in the last time step between the pin and disc.
Chapter 4

Refining the PoD models

Now a good model is designed, the model can be refined in order to generate more accurate results on the pressure distribution. The real wall thickness of the phenolic resin is between 0.4 and 0.45 millimeters. When the real wall thickness is reached further refining of the model can be done, but will be of no use in order of accuracy. Further refining will only be necessary for real real wear modeling and even then only for the contacting nodes.

Of every refining step two models are made. One with only the aluminium core material and one where the surface elements are of the resin material, see figure A.1. By making two models of each refining step the influence of the different materials on the pressure distribution can be shown clearly. The changes in order to the previous model will be presented for each model.

Again a check is made of the angular velocity of the disc and the frictional dissipation between the pin and the disc. If they are similar these results will not be shown but a reference will be made to the figures of the first model, because they contain no new information.

4.1 The first refined mesh models

Model 2 and 3 are the first models with a more refined mesh. In the direction of the height of the pin, 7 elements are used, and in the other two directions 4. In this way the pin elements are almost square, what is desirable. The disc mesh is refined as well. The mesh of the disc is presented in the appendix, see figure A.2 and table A.1. The size of the mesh of the disc is the same for the models 2, 3, 4 and 5. The mesh is not further refined for model 4 and 5 to keep the simulation time short.

4.1.1 Changes in the part module

For each element a different intersection is made. This is done for two reasons. In the subsections about the property and load module will be explained why. Normally a mesh size or a seed edge is given and then the desired mesh size can be generated.

Another intersection is made for the wall material at the bottom of the node. Probably is the pressure distribution between the pin and disc dependent on the different material properties of the pin. The wall material thickness thus the resin cover for model 3 is about 0.7 [mm].

4.1.2 Changes in the property module

The only change in the property module is that the material properties of the pin resin are changed for model 3. In table 4.1, the material properties for the resin are presented. The intersections made in the part module enables the possibility to select different regions for the pin core and resin material. In figure A.1 a clear view of the different materials is given. The contacting surface partition consists of the resin material. Except the top four partitions at the center of the top surface. In real life this is the case as well. These partitions consist of aluminum.
## 4.1. THE FIRST REFINED MESH MODELS

### Table 4.1: Material properties.

<table>
<thead>
<tr>
<th>Section Name</th>
<th>Material Behavior</th>
<th>Type</th>
<th>Young’s Modulus [N/mm²]</th>
<th>Poisson’s Ratio [-]</th>
<th>Section Type</th>
</tr>
</thead>
<tbody>
<tr>
<td>Pin Core</td>
<td>Mechanical, Elastic</td>
<td>Isotropic</td>
<td>18500</td>
<td>0.33</td>
<td>Solid/Homogenous</td>
</tr>
<tr>
<td>Pin Resin</td>
<td>Mechanical, Elastic</td>
<td>Isotropic</td>
<td>12300</td>
<td>0.33</td>
<td>Solid/Homogenous</td>
</tr>
<tr>
<td>Disc</td>
<td>Mechanical, Elastic</td>
<td>Isotropic</td>
<td>210000</td>
<td>0.33</td>
<td>Solid/Homogenous</td>
</tr>
</tbody>
</table>

### 4.1.3 Changes in the assembly module

Nothing changed in the assembly module.

### 4.1.4 Changes in the step module

In the step module it occurs that in the second step at the first second (the 2\textsuperscript{nd} second) the behavior becomes more or less stable. A repetitive behavior is shown. To set the length for the next modeling session 5 seconds in this time step is sufficient. The start increment value is lowered as well as the minimum value. Also a maximum increment time size is set. Otherwise at the end of the simulation the steps taken by the simulation program are too large, see table 4.2.

<table>
<thead>
<tr>
<th>Step</th>
<th>Step name</th>
<th>Procedure</th>
<th>Time [s]</th>
<th>Maximum number of increments</th>
<th>Increment size</th>
</tr>
</thead>
<tbody>
<tr>
<td>0</td>
<td>Initial</td>
<td>Initial</td>
<td>N/A</td>
<td>N/A</td>
<td>N/A</td>
</tr>
<tr>
<td>1</td>
<td>Apply Pin Force</td>
<td>Static, General</td>
<td>1</td>
<td>10</td>
<td>0.01 1.0 0.125</td>
</tr>
<tr>
<td>2</td>
<td>Rotate Disc</td>
<td>Static, General</td>
<td>5</td>
<td>100</td>
<td>1.0 1.0 0.125</td>
</tr>
</tbody>
</table>

### 4.1.5 Changes in the interaction module

To be able to look for the real pressure distribution between the pin and the disc a more finer mesh is needed. But first the optimal friction model has to be implemented in the model to make sure that the pressure distribution which will occur is as close to real as possible. Therefore the first model will be used to fine tune this parameter. In the first model a simple penalty is implemented to simulate the friction. There are five different models with subroutines to simulate friction contact between surfaces. A closer look at all the possibilities is made. They are presented below. It is possible to change the friction properties in every step. In this case a lowered value of the CoF can be given to introduce the effect of lubrication between the bodies. In figure 4.1.(a) the desired friction behavior can be seen.

The friction formulation between the pin surface and disc surface which is very important and has probably a large influence on the pressure distribution between the pin and disc surface. The possible friction models in ABAQUS are:

- Frictionless
- Penalty
- Static-Kinetic Exponential Decay
- Rough
- Lagrange Multiplier (Standard only)
Chapter 4. Refining the PoD models

Sandblast pressure $P = 0.3$ [Mpa]

(a) The measured friction coefficient in the real PoD-test.

(b) Way of manufacturing pulley discs on large scale, the dotted line represents the movement of the nozzle and the angular movement of the disc is denoted by $\omega$.

Figure 4.1: Friction related data.

There are three options in the program that cannot be used in order to get the right friction model. The first one is the Frictionless option. It prescribes that when the surfaces contact that there will be no friction at all between the surfaces. PoD-testing is all about friction. Using this option will result in no (useful) data output. The next option that cannot be used is the rough option. This one prescribes that when the surface make contact, they will not be able to move in order to each other at all. This one can also not be used to obtain a good friction formulation. The last option that cannot be used is the Static-Kinetic Exponential Decay option. No anisotropic material behavior is available for this feature. At the moment this is not necessary, but maybe this behavior will be needed when the pulleys are produced on scale. While sandblasting, probably the pulleys will rotate while the nozzle moves in the radial direction (dotted line), see figure 4.1.(b). Due to this way of manufacturing it is possible that the CoF gets different values for the tangential and radial direction and this possibility should be kept open.

The Lagrange Multiplier-model should be avoided when possible. It has almost the same setting options as the penalty option, but using it will result in excessive longer CPU-times.

Eventually the Penalty option is used. In the first model an ordinary penalty was prescribed but this feature has more options, which can make it a good option to use. This feature is based on the Coulomb friction model. In case of a 3D-model it is possible to prescribed anisotropic friction behavior. It is also the friction formulation that used in the model in the original CVT-model in which the finer mesh-model will be implemented [4]. Only the peak out of figure 4.1 can not be simulated within one step. This peak is left out because in the PoD test it is of minor importance. In the real CVT model this is more important because the CVT works within the range of the no slip region. Then it should be simulated.

4.1.6 Changes in the load module

For each element a separate intersection is made. This is done in order to give all the nodes on the top pin surface the same BC’s as in the previous model for the centerlines. Otherwise extra stresses are generated and this is not desirable.

For the force on the top of elements all the top surfaces are selected (from four to eight). Because the unit is in Newton per square millimeter nothing has to the value of the pressure.
4.1.7 Changes in the mesh module

Again the seed feature is used to assign the number of elements along the edges. For the pin this means 4 elements in with and depth and 7 elements in the height direction. For the disc for the fine part 35 in tangential direction, for the rough part 15 and in the radial direction 15 elements and in the height 8, see figure A.1 and A.2 and table A.1

4.1.8 Changes in the job module

In the job module nothing has been changed

4.1.9 Changes in the visualization module

In the visualization module the same kind of behavior occurs. Only in a more refined way. If an intersection is made of the pin’s with resin material extra stresses can be seen between the two materials.

4.2 Results of the first refined mesh models

For both models the distance and frictional dissipation data are the same as the first model. Also the forces in both directions are similar. Because there are more elements contacting more detailed graphs are available for these models.

In the figures 4.2 the first differences in pressure distribution occur due to the different material properties of the pin. In figure 4.2.(a) the pin consists of only one material and the pressure represented in figure 4.2.(b) shows the pressure distribution due two the two different materials.

4.3 The second refined mesh models

Again a refinement of the pin mesh is made. The refinement of the pin mesh happens in the same way as prescribed in the previous section. The only difference is that the thickness of the wall material (resin) is now around 0.35 [mm]. To see the actual mesh sizes and which part of the pin contains which material see figure A.1. The mesh of the disc remains the same. Refining it further would result in too large CPU-times. After running the models the difference between the aluminum pin and the ’real’ pin becomes clear.
4.4 Results of the second refined mesh models

Similar behavior can be seen in comparison with the 2\textsuperscript{nd} and 3\textsuperscript{th} model. Because of the doubled number of elements a more detailed view on the pressure distribution, see figure 4.3.

![Pressure distribution for model 2](image1)

(a) The pressure distribution between the pin and disc for model 2.

![Pressure distribution for model 3](image2)

(b) The pressure distribution between the pin and disc for model 3.

Figure 4.3: The difference between the pressure distribution between the two models. On the left with a pin consisting of 1 material. On the right with a pin consisting of 2 materials.

4.5 The results of the 3\textsuperscript{th} and 5\textsuperscript{th} models compared

The more refined model, model 5 has a thinner wall than the model 3. A thinner wall close to the real one, shows that in the simulations a larger difference in pressure distribution appears due to the different wall thicknesses of the pin. If you compare figure 4.4.(a) and 4.4.(b) a more detailed and more accurate pressure distribution occurs. What the influences are on the transmittable torque needs further research.

![Pressure distribution for model 2](image3)

(a) The pressure distribution between the pin and disc for model 2.

![Pressure distribution for model 3](image4)

(b) The pressure distribution between the pin and disc for model 3.

Figure 4.4: Effects of refining the mesh on the pressure distribution.
4.5. THE RESULTS OF THE 3TH AND 5TH MODELS COMPARED
Chapter 5

Microscopic friction behavior

To get a better understanding of the frictional behavior, a microscopic research has been done on the surface conditions of the contacting materials.

Measurements on the roughness of the block and disc materials has been done. Because it is hard to determine the roughness of the block at the contact area, because of the angle of the surfaces, the roughness of the resin on the top of the element is a good approximation for the roughness. In figure 5.1 the position of the measurements is shown. The micro sandblast treatment of this disc is done with a pressure 0.3 [MPa], perpendicular to the material. This roughness is easy to determine, because the test disc is flat.

![Figure 5.1: Positions where the surface roughnesses of the block and disc are measured.](image)

Three measurements have been done and a average value has been calculated for all the needed variables. In the table 5.1 the data is presented. In figure 5.2 the real surfaces are shown.

<table>
<thead>
<tr>
<th>Surface roughness block</th>
<th>Surface roughness disc Sandblast pressure 0.3 [MPa]</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>take 1</td>
</tr>
<tr>
<td>Length X [µm]</td>
<td></td>
</tr>
<tr>
<td>298.291</td>
<td>298.291</td>
</tr>
<tr>
<td>223.718</td>
<td>223.718</td>
</tr>
<tr>
<td>66733</td>
<td>66733</td>
</tr>
<tr>
<td>0.299</td>
<td>0.266</td>
</tr>
<tr>
<td>7.32</td>
<td>9.16</td>
</tr>
<tr>
<td>4.608</td>
<td>6.734</td>
</tr>
<tr>
<td>1.5</td>
<td>5.6</td>
</tr>
<tr>
<td>0.38</td>
<td>0.34</td>
</tr>
<tr>
<td>1.282</td>
<td>1.311</td>
</tr>
</tbody>
</table>

Table 5.1: Roughness data of the pin and disc.
Out of these data the variables needed for the FE analyses are obtained. First a model is made with smooth edges to get the settings right and the simulation running. Then a simplified 2-D model is made. The roughness of the peaks is for the resin material set on a width of 1 [µm] and the height 0.5 [µm] and for the disc material on a width of 1 [µm] and the height 10 [µm]. All the other parameters as speeds, forces, material properties are similar as in the PoD simulations. The size of the upper body is 50 by 50 [µm]. The lower body of 50 by 150 [µm].

In figure 5.3 the 2-D model is shown. The upper body consists of the disc material and the lower body consist of the phenolic resin. Movement of the upper body in the x-direction is suppressed. A pressure is applied at the top line of the upper body in the 2nd-direction. The boundary conditions on the edges allow the material only to move in the 2nd-direction. In the first step the edges of the lower body allow the material to move in the 2nd-direction. The lower edge is not moving in nor the 1st or 2nd direction. In the second step the lower body is moving with an equivalent speed of 1.93 [mm/s]. The pressure on the top edge of the disc material is maintained.

The first simulations with smooth (no rough surface) went well. After applying the peaks (the surface roughness) the simulations failed. The boundary conditions, sizes and settings are right. The problem could be solved by adjusting the accuracy of the coefficient of friction.

The next idea is to make floating boundary conditions. This means that there are two bodies that are of the same size. One body is fixed and the other moves in such a way that the surface floats and is repeating itself. The surface runs like a wave on the top surface. The idea is that side effects do not occur. Finally this should be applied for a 3-D structure. No time was left to do this. The data of table 5.1 in combination with the working conditions can also be used to determine weather and what kind of lubrication is present in the test rig. Line contact is present and the type of lubrications that can occur are:

- HL, the opposing surfaces are completely separated, like aquaplaning on a road
- EHL or EHD, the opposing surfaces are completely separated and the solid bodies deformed by a major load
- BL, the bodies are not entirely separated

This research has not been done anymore, because it would take too much time.
Figure 5.3: The first microscopic friction model.
Chapter 6

Belt modification

The main problems of the dry hybrid V-belt drive started with the rising of the temperature in the transmission box. Under normal, dry operating conditions the temperatures became too high and cooling should be applied. The first idea was to cool with air. This was done by drilling venting holes in the transmission box. Cooling with air is not sufficient and too much noise generated by the pulleys and belt. The transmission box was closed again and oil cooling was applied at the belt. Cooling was sufficient and the noise reduced. At this point the problems occurred with CoF. Due to the oil that is used to cool the belt, the CoF decreases dramatically. The first oil that was used to cool, was ATF-fluid (Automatic Transmission Fluid). By changing this type of fluid to TDF-fluid (Traction Drive Fluid out of the torodial CVT) a small increase of the CoF was obtained. But still the CoF is not at the desired level. The transmittable torque had still a too low value. After the surface treatment a difference occurred between the oils in favor of the TDF-fluid. The CoF is now larger but the lifetime of a V-belt is also shorter, because of the larger wear due to the surface treatments of the pulleys. The sandblasting treatment of 0.3 [MPa] is a good compromise between the lifetime of the belt and transmittable torque.

Figure 6.1: Main spots of noise production.

6.1 Noise production

Most of the noise in the test rig is generated at the points where the belt enters and leaves the pulleys, see figure 6.1. In order to reduce the noise a modification of the belt may solve this problem. The belt itself consists of 204 blocks. These are kept together with two rubber belts at each side of the blocks, see figure 1.1. The belt exceeds the blocks a little bit. At the top and bottom of the belt a ripple is present, which holds each block at its fixed position. It is possible that the ripple of the belt is generating most of the noise, because every ripple bumps with high
speed on the pulley by entering it, and slides off when leaving it. Creating a smooth edge may be a solution in order to solve this problem. By removing only a small edge of the belt, the ripple can be removed, see figure 6.2. Because only a small part of the belt is removed the transmittable torque will be lowered, this should not be too large. Maybe the problems with the noise and heating disappear due to this modification and is cooling with air again possible. Cooling with oil is not needed anymore and the problems with the decreased CoF and wear are solved. Removing this ripple may be a good idea, because it is applicable in the real fabrication process of the rubber belt. The belts already have chamfered sides in order to fit tightly against the blocks at the center.

Figure 6.2: Belt with removed edge.

6.2 Removing the belt ripple

Modification of the belt is not easy. When the belts are ordered the are already assembled when they arrive the laboratory. It is almost impossible to disassemble the belt by hand, because the blocks are fitted very tight on the rubber belts. It is possible to order separate blocks and belts and assemble them by hand but that is as well almost impossible. To be able to disassemble a belt, an used belt is ordered. This belt has run for 24 hours. It has a little backlash between the blocks and the belts. By ticking on the top of the blocks the rubber belt comes out itself. Make sure that all blocks get back on the same position on the belt when re-assembling the belt. This belt has run for some time and is a little worn. It is important to get all the blocks back at their original position. After disassembling the ripple can be removed by grinding off the edge. After re-assembling the belt it is ready to run in the CVT test rig. The level of noise can be measured and compared with the noise level of the untreated belt.

6.3 Results of the belt modification

Before the belt is modified it is put in the CVT test rig and the level of noise is measured at a fixed position between the two pulleys. After modification at exact the same place the noise level is measured again. This is important to be able to get comparable data. In the tables 6.1 and 6.2 the results of the measurements of the noise level is presented for the belts. Not all the desired conditions could be measured, because of problems with the setup. It was not possible to let this shaft rotate at high rotational speeds and high clamping forces were not possible. In case of missing data a ×-mark is filled in. The sets of measuring data are not very reliable. The level of noise should be at least equal. The fact that there were a lot of students in the laboratory during the second run of the treated belt will not have been a benefit for the results. Another noise source, which generates a lot of noise is the hydraulic unit. A second measurement with both belts should be done with both runs on one day should be done.
Table 6.1: Measured noise level of the untreated belt.

<table>
<thead>
<tr>
<th>Belt type:</th>
<th>Speed in [rpm]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Un treated belt</td>
<td>74.5 81.5 84.5</td>
</tr>
<tr>
<td>Under Drive</td>
<td>73.5 77.7 79.8</td>
</tr>
<tr>
<td>Neutral</td>
<td>× × ×</td>
</tr>
<tr>
<td>Over Drive</td>
<td>× × ×</td>
</tr>
</tbody>
</table>

Table 6.2: Measured noise level of the belt with removed ripple.

Testing conditions:

\[ Q_{DN} = 3000 \ [N] \ T_{applied} = 0 [N] \]
6.3. RESULTS OF THE BELT MODIFICATION
Chapter 7

Implementation of the friction model in the CVT-model

In order to get a better, more detailed comprehension in the working of a CVT and the influence of the pressure distribution on the transmittable torque, the outcome of the FE-analyses of the PoD-model have to implemented in this model, since the PoD simulations worked out well. Three V-belt blocks will be modeled with a finer mesh and the two different materials so that the pressure distribution can be calculated on all arms. The blocks with the finer meshes has to rotate with the pulley once. The first step is the implementing of the model in the rigid shell model without pulley tilting. Then the influence in ideal working conditions can be examined. The pressure distribution on all arms can be examined and if this works out well it can be implemented in the more extended model with tilting and deformable pulleys. Afterwards conclusions can be drawn out of the importance of the pressure distribution and how this affects the efficiency of the CVT.

In this model the rubber belt is still sticking out in order to the block arms. The edge of the rubber belt is removed and is within the the arms. Otherwise there will be no comparable results with PoD results. If the belt is sticking out the belt touches the pulleys first and the friction of the rubber belt is much larger, so the influence of the transmittable torque by the arms cannot be analyzed. In figure 7.1 the configuration of the belt in these analysis is presented.

(a) On the left side the original block with belt. On the right side the adapted block is with retracted belt. (b) Position of the three fine mesh blocks in the CVT belt.

Figure 7.1: Adaptations for the CVT model.

7.1 The first CVT-model

In the first CVT-model the block consist only of aluminum, the same as in the POD simulations. How the model is build is not explained. Look for more details in [1]. The only thing that has been changed is that the rubber belt is not extending the blocks anymore. 0.515 [mm] of the rubber belts is removed over the complete edge as can be seen in figure 7.1.(a). To let the simulation run
well, an adaptation is made in the friction model. The contact formulation with the pulleys and belt are removed. Normally the slip between the blocks and pulleys has a margin region of 0.5 %. To let the simulation run it is set on 7 %. This means that the overall accuracy is lowered, but still acceptable.
Chapter 8

Conclusions and recommendations

A good and reliable PoD FE model has been built. The simulations of the PoD tests give good and reliable results. The results out of the FE model are verified with analytical calculations and the results are almost the same. The pressure distribution that has been calculated by FE programm is probably good representation of the pressure distribution in real life. The further refinements of the models give a good more thorough view on the pressure distribution.

Building a FE model for simulating the microscopic friction has not been succeeded. Problems occurred during the programming of the model. By the time that they could be solved, there was no time left to work out the model in a more detailed way. Proper suggestions have been made. In the future these can be worked out.

The modifications of the belt did not result in a lower noise production. The measurements showed that the level of noise was higher then before the belt modification. Maybe the measurements should take place early in the morning when the laboratory is less crowded and both runs of treated and untreated belt should take place on one day. This because the calibrations and surrounding noise is more or less the same. Also the noise of the hydraulic unit should be isolated.

Also because of lack of time the modification of the FE CVT model did not give comparable data. A first model has been made and runs well. But data that can be compared with the PoD tests is not available. A good CVT model has been made and is ready for use.

A complete overview of the problems is shown in figure 8.1. If all these problems have been simulated in ABAQUS and been tested experimentally a complete overview of the CVT belt is present. Then an optimization can be applied on the design of the belt, pulleys and working conditions.

Figure 8.1: A complete overview of the problems.
# Acronyms

<table>
<thead>
<tr>
<th>Acronym</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>ATF-fluid</td>
<td>Automatic Transmission Fluid</td>
</tr>
<tr>
<td>BL</td>
<td>Boundary Lubrication</td>
</tr>
<tr>
<td>CPU</td>
<td>Central Processing Unit</td>
</tr>
<tr>
<td>CVT</td>
<td>Continuously Variable Transmission</td>
</tr>
<tr>
<td>CoF</td>
<td>Coefficient of Friction</td>
</tr>
<tr>
<td>DOF</td>
<td>Degree Of Freedom</td>
</tr>
<tr>
<td>EHL or EHD</td>
<td>ElastoHydrodynamic Lubrication</td>
</tr>
<tr>
<td>FEM</td>
<td>Finite Element Method</td>
</tr>
<tr>
<td>HL</td>
<td>Hydrodynamic Lubrication</td>
</tr>
<tr>
<td>PoD</td>
<td>Pin on Disc</td>
</tr>
<tr>
<td>RP</td>
<td>Reference Point</td>
</tr>
<tr>
<td>TDF-fluid</td>
<td>Traction Drive Fluid</td>
</tr>
</tbody>
</table>
Bibliography


Appendix A

Pin mesh and disc mesh

In figure A.1 the actual sizes of the meshes of the 3 different models are given. In the last one the thickness of the wall is almost as in real. In table A.1 the number of elements along each line of the disc are presented.

Figure A.1: The pin mesh, number of elements and sizes.
Figure A.2: x, y and z represent the edges and are corresponding to number of elements along these edges in table B.1.

<table>
<thead>
<tr>
<th>Model number</th>
<th>Number of elements along the edge</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>x1 14 x2 9 y 4 z 7</td>
</tr>
<tr>
<td>2 and 3</td>
<td>x1 35 x2 15 y 8 z 15</td>
</tr>
<tr>
<td>4 and 5</td>
<td>x1 35 x2 15 y 8 z 15</td>
</tr>
</tbody>
</table>

Table A.1: Number of elements along the edges of the disc.