Design of a drivetrain for a FS race car

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Bachelor End Project

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Appendices
Introduction

University Racing Eindhoven (URE) is a team of 50 students who competed in the Formula Student competition. The competition began in 1986 in the United States. The main goal was to bring automotive students and automotive company’s together. With first vehicles that looked like go karts, the competition improved over the years to very professional open wheel, single-seater racecars. Much new and mostly very innovative stuff made the cars go faster but also a lot more complex. In 1999 the competition came to Europe. It started with one race in England and nowadays there are three races in the UK, Germany and Italy.

Students get a lot of freedom in the design of their car. The most important points a Formula Student car has to fulfill are, a fully operational suspension system, an 4 stroke engine with a maximum of 610cc and an inlet restrictor with a diameter of 20mm. For the drive train there are no major restrictions. Cars are equipped with four wheel drive, CVT’s etcetera.

URE started this year with the build of there fifth car in six years. Every year the new car became better than the previous car. Nowadays the team belongs to the sub top of the European Formula Student teams.

In the past, the reliability of the car was the main problem, also in the driveline. That is why the main aspect of this bachelor final project is reliability. After that comes weight of the parts and the manufacturing/purchasing costs. The project involves all the parts of the drive train, from engine to rear wheels. Every chapter contains a specific part of the drive train. After examining the drive train of the team’s previous car, the URE04, the chapters follow the new drive train from final drive to wheel hubs.
1. **Model driveline**

As a start for the design of the drive train for the URE 05 the drive train of the URE 04 is investigated. The layout of both drive trains is similar. The crankshaft is connected to the clutch with a gear ratio of 1.93 (79/41). The gearbox is a sequential six speed gearbox, the first gear has a ratio of 2.79 (39/14). The car is chain driven and has a final drive ratio of 3.82 (42/11). The differential is of the torque sensing type and has a torque bias ratio of 3:1. This means that the differential can transfer 75% of the torque to the wheel with the most traction. The drive shafts have unequal length because of the offset of the differential to the middle of the car. The unequal length creates a different in stiffness between the left and right driveshaft. The drive shafts are equipped with homokinetic couplings to cope with the wheel travel. Figure 1.1 gives a schematic overview of the layout of the drive train.

![Figure 1.1; schematic overview of the driveline](image-url)

To make a good estimation of the torque in the driveline a schematic model of the drive train is made (appendix A). The model is simplified and the inertia and stiffness’s are recalculated with respect to the different gear ratios (appendix b). The inertia of the car becomes:

\[
J_{\text{total}} = 0.10 \text{ kg} \cdot \text{m}^2
\]

\[
k_{\text{total}} = 16.66 \text{ Nm/rad}
\]

\[
\omega = 12.58 \text{ rad/s}
\]

\[
J_{\text{engine}} \text{ and } J_{\text{car}} \text{ both have an inertia of } 0.05 \text{ k} \cdot \text{m}^2 \text{g}.
\]
For $J_{\text{car}}$ the new resonant frequency can be calculated. Because have almost the same inertia the simplified model can be cut in half. The stiffness must then be multiplied by 2.

$$\omega = \sqrt{\frac{K_{\text{total}} \cdot 2}{J_{\text{engine}}}} = \sqrt{16.66 \cdot 2/0.05} = 26 \text{ rad/s}$$

In the past drive train failure always occurred during take off from 0 km/h. This is the situation that will be investigated. The typical model parameters for this situation are:
- The crankshaft is rotating with 6500 rpm,
- All parts of the drive train after the clutch have no rotational speed/energy before the clutch closes
- Due to launch control the rotational speed of the crankshaft never drops below 6500 rpm when clutch closes

The rotational speed of the driveshaft at the moment the clutch closes is the engine speed converted to rad/s.

$$v = \frac{6500}{60} \cdot 2\pi = 681 \text{ rad/s}$$

The angular acceleration can be calculated by:

$$\alpha = \omega \cdot v = 26 \cdot 681 = 17698 \text{ rad/s}^2$$

With the angular acceleration and total inertia the torque can be determined.

$$T = \alpha \cdot J = 17698 \cdot 0.10 = 1770 \text{ Nm}$$

The torque is divided into two driveshafts, which both gets half of the torque. This means the torque in one driveshaft is:

$$T_{\text{driveshaft}} = \frac{1770}{2} = 885 \text{ Nm}$$

For the drive train a safety factor of two is applied. So the total torque after the final drive is:

$$T_{\text{safety}} = 1770 \cdot 2 = 3540 \text{ Nm}$$
This means that each driveshaft has to be designed to cope with 1770Nm of torque. The homokinetic couplings (which are purchased) can cope with a maximum torque of 1700Nm.
2. Final drive

The final drive is the reduction between the gearbox and the differential. Because it’s not possible to adapt the gearbox ratios the top speed of the car is prescribed by the final drive.

- 2.1 Design goals.
  o Possibility of removing first gear.
  o Calculate final drive ratio.
  o Calculate type chain/sprockets.

- 2.2 Possibility of removing first gear.

The following gear ratios are used in the Suzuki GSX R600 engine

<table>
<thead>
<tr>
<th>Gear</th>
<th>Ratio</th>
</tr>
</thead>
<tbody>
<tr>
<td>1st</td>
<td>39/14 = 2.79</td>
</tr>
<tr>
<td>2nd</td>
<td>32/16 = 2.00</td>
</tr>
<tr>
<td>3rd</td>
<td>32/20 = 1.60</td>
</tr>
<tr>
<td>4th</td>
<td>30/22 = 1.36</td>
</tr>
<tr>
<td>5th</td>
<td>29/24 = 1.21</td>
</tr>
<tr>
<td>6th</td>
<td>25/23 = 1.09</td>
</tr>
</tbody>
</table>

Table 2.1: Gearbox ratios

Note: in the instruction manual of motorbike the 4th ratio is wrong, the ratio’s in table 2.1 are counted on the gearbox sprockets.

There are two reasons to benefit from removing the first gear. As shown in figure 2.1 there is a large gap between first and second gear. The gear change pattern of the gearbox is


The gearbox is one of the sequential type which means that to change gears you need to shift up or down and then go back to the neutral position. It is not possible to change two gears in one time. An internal spring in the gearbox makes sure that the gear lever goes back to neutral position after a gear change.

The problem is the neutral gear, it only needs a half stroke of the gearlever to go to neutral. On previous cars of University Racing Eindhoven an electric motor and a potmeter where used to control the actuator to go to neutral. This system is almost as heavy as a pneumatic system. A pneumatic system is much harder to control to go to neutral. The problem was the millage of the electric system.

With the removal of first gear, neutral becomes a complete stroke down of the gearlever which makes the system easier to control.
2.3 Calculate final drive ratio.

In figure 2.1 are the speeds plotted against the engine rpm with a final drive ratio of 11/43. This is the setup of the URE04 car.

The top speed of the car with this setup is about 160 km/h. A typical top speed during a race is about 125 km/h. The first gear seemed to be a bit short depending on the acceleration results. 12000 rpm is the best point to shift gear because the power drops after 12000 rpm. This means an rpm drop of about 3000 rpm in first gear. Depending on the power curve this means also a great power drop.

With the two design goals in mind, removing first gear and a top speed of about 125 km/h a new final drive ratio is calculated, see appendix C, the Matlab-file “eindoverbreining.m”. This Matlab file calculates the speeds in each gear, the diameters of the sprockets, and the number of links in the chain. This last is particularly important because the number of links used is about half the number of links used by a motorcycle. With the exact number, a chain can be ordered which can be cut into two chains.
With a final drive ratio of 11/50 the top speed in 6\textsuperscript{th} gear and 14000 rpm is 140 km/h. In practice this rpm in 6\textsuperscript{th} gear is hard to reach. The speed of 125 km/h is at 12000 rpm. The first gear is about 10km/h longer then with the final drive ratio 11/43.

The rear sprocket has a larger outer diameter then one used previous year. This means that the centerline distance between the front and rear sprocket had to be increased. The result of 11/50 final drive ratio with the old centerline distance is seen in figure 2.3.
Figure 2.3; 1 = front sprocket, 2 = rear sprocket

Because the gap between the two sprockets is very narrow, any disalignment of the sprockets would result in excessive wear of the chain. To make the alignment of the sprockets less critical the engine is moved 8 centimeter forward. The relocation of the engine was done in consult with the designer of the chassis.

- **2.4 Calculate type chain/sprockets.**

Motorcycle chains and sprockets come in different sizes. The size is based on the roller-width, roller-diameter and pitch of the chain. The original chain for this engine is a 525 chain. In table 2.2 different motorcycle chains are listed.

<table>
<thead>
<tr>
<th>Chain no.</th>
<th>Pitch</th>
<th>Roller diameter</th>
<th>Roller width</th>
<th>Sprocket thickness</th>
</tr>
</thead>
<tbody>
<tr>
<td>420</td>
<td>1/2”</td>
<td>5/16”</td>
<td>1/4”</td>
<td>0.227”</td>
</tr>
<tr>
<td>425</td>
<td>1/2”</td>
<td>5/16”</td>
<td>5/16”</td>
<td>0.284”</td>
</tr>
<tr>
<td>428</td>
<td>1/2”</td>
<td>0.335”</td>
<td>5/16”</td>
<td>0.284”</td>
</tr>
<tr>
<td>520</td>
<td>5/8”</td>
<td>0.400”</td>
<td>1/4”</td>
<td>0.227”</td>
</tr>
<tr>
<td>525</td>
<td>5/8”</td>
<td>0.400”</td>
<td>5/16”</td>
<td>0.284”</td>
</tr>
<tr>
<td>530</td>
<td>5/8”</td>
<td>0.400”</td>
<td>3/8”</td>
<td>0.343”</td>
</tr>
<tr>
<td>630</td>
<td>3/4”</td>
<td>15/32”</td>
<td>3/8”</td>
<td>0.343”</td>
</tr>
</tbody>
</table>

Table 2.2; motorcycle chain dimensions (http://www.gizmology.net/sprockets.htm)
With the calculated torque of 3540 Nm (see chapter 1) and the rear sprocket diameter of 0.25m the force in the chain can be calculated.

\[ F_{\text{chain}} = \frac{T}{r} = \frac{3540}{(0.25/2)} = 28500N \]

A standard 520 chain has a max chain force of 27000N. By choosing this chain, the chain will be the weakest link in the drive train. The advantage of this will be that a chain is relatively cheap to the other drive train parts and easy to repair. Also 520 chains and sprockets are widely available.
3. Differential

The differential is the connection between the rear sprocket of the final drive and the drive shafts. A differential has also the function of allowing the drive shafts to rotate with different speeds. This is important when taking a corner.

- 3.1 Design goals.
  - Choice of a differential
  - Design of a chain tensioner
  - Design of the differential mounts

- 3.2 Choice of a differential.

For the differential, weight is an important issue. Most differentials exceed more than 7 kilograms and are designed to cope with much more torque than in a formula student driveline. There are two main types of differentials used in the formula student competition. The Torsen differential from an Audi Quattro and a Salisbury type differential custom made by a company in Germany called Drexler.

The first differential is used on our two previous cars. There are three disadvantages about the Torsen differential:
  - Almost no adaptations in torque bias ratio
  - When one wheel loses grip, all the torque goes to that wheel
  - Weighs 3.3 kg without housing

The most important advantages are
  - Cheap
  - Short delivery time
  - Reliable, even without oil

The Salisbury differential is specially designed for the formula student competition and is run by several teams. Advantages are
  - Very wide adaptations in torque bias
  - Weighs 2.6 kg including housing
  - Different characteristic for accelerating and braking
  - Oil sealed

The disadvantages are:
  - Expensive
  - High Running cost
  - Very long delivery time

Because all the disadvantages of Salisbury differential can be overcome this differential is chosen.

The Salisbury type differential works like a normal differential with additional friction plates. A normal differential divides the torque equally over the two drive shafts. When one wheel loses grip all the torque goes to that wheel. The difference between a normal differential and a Torsen differential is that the Torsen differential has a torque bias ratio
of 3:1. It can send 75% of the torque to the wheel with the most traction as long as both wheels have traction.

Figure 3.1 shows the internals of a Salisbury differential. The drive shaft is connected to the side gear (green). The ground ramp (bleu) is driven by the engine. On the other side of the cam (black) is another ground ramp (not shown).

The hole in the ground ramp for the cam has a camfer, when the ground ramp is driven the camfer pushes the ground ramp to the outside. With this movement clutch plates are pressed together which creates friction. How hard the clutch plates are pressed together depends on the shape of the camfer. Half of the clutch plates are connected to the side gear and the other half to the ground ramp. The friction creates a connection between the side gear and the housing, and so between the engine and the wheels. When there is deceleration the cam pushes to the other side of the hole in the ground ramp, see figure 3.2. This side has another camfer and so another characteristic.
3.3 Design of a chain tensioner.

In chapter 2 a 520 chain was chosen for the final drive. Because the length of the chain is not stepless, a chain tensioner has to be designed. To have the chain under the right tension prevents the chain from excessive wear and from the possibility to hop of the sprockets during driving.

The pitch of a 520 chain is 5/8” or 15.875mm. A chain is build alternately out of male and female links. So the chain tensioner needs to expand two times the pitch (31.75mm) to make a connecting link fit, see figure 3.3.

The design of the chain tensioner consist of two excentric disks witch each house one of the differential bearings. The disk can rotate in the differential mount. The centerline of the differential bearing has an offset of 1cm to the centerline of the chain tensioner disk.
By rotating the chain tensioner disk 180 degrees the distance between engine and differential, or front and rear sprocket, vary by 2cm. If the top and bottom line of the chain runs parallel this means a difference in chain length of 4cm. This is a little more then the 31.75mm required to bridge two times the pitch.

In appendix D the bearing forces on differential bearings are calculated. The chain tensioner disk is made out of aluminum 6082-T6. This material is commonly used at our manufacturer. For material properties see appendix E. The maximum stress stays below the yield stress of the material, see figure 3.4.

- **3.4 Design of the differential mounts.**

The differential mounts have two functions, namely:
- Support for the differential bearings
- Two of the four engine mounts to the frame.

One of the disadvantages of the Drexler differential was the long delivery time. The differential will be delivered at the end of April 2009. The estimated date that the URE05 has its roll out is the 10th of March. This means that another differential has to build in to the car till the Drexler differential arrives. A previous car of URE, the URE03, is fitted with a Torsen differential. At first the URE05 will be fitted with differential of the URE03.
For the design of the car, the agreement was made that the x direction is the length, the y direction is the width and the z direction is the height of the car. The front engine mounts fix the engine in the x, y and z direction and rotation around the x axis. The differential mounts (eq. the rear engine mounts) must fix the rotation around the y and z axis. The differential mounts will be attached to the rear bulkhead. This bulkhead is stiff in the y and z direction but relatively weak in x direction.

The last requirement for the differential mounts is the location of the differential. As seen in figure 2.3 the front- and rear sprocket can not overlap each other. Also it is best for the homokinetic couplings standard not to be under an angle. This means that the centerline of the differential must lie on the centerline of the rear wheels.

Figure 3.5 shows how the differential and the differential mounts fit in the assembly of the car. Shown are the engine on the left and the rear bulkhead on the right.

During the finite element analysis (FEA) two things are important. The stress may not exceed the maximum yield stress and the displacement in the y direction of the rear sprocket may not cause the chain to run of the sprocket.

Figure 3.6 shows the result of the FEA. In the analysis the differential mounts are designed in detail. The engine and differential with rear sprocket are designed as a solid to apply the constrains and forces on the right position. As an assumption for the weight
of the engine (including intake, exhaust, etc), 800N is taken. The forces on the engine are:

<table>
<thead>
<tr>
<th>Direction</th>
<th>Acceleration</th>
<th>Force</th>
<th>Magnitude [N]</th>
</tr>
</thead>
<tbody>
<tr>
<td>X</td>
<td>1.5G</td>
<td>$F_x$</td>
<td>$1.5 \times 10 \times 800 = 1200$</td>
</tr>
<tr>
<td>Y</td>
<td>1.5G</td>
<td>$F_y$</td>
<td>$1.5 \times 10 \times 800 = 1200$</td>
</tr>
<tr>
<td>Z</td>
<td>3.5G</td>
<td>$F_z$</td>
<td>$3.5 \times 10 \times 800 = 2800$</td>
</tr>
<tr>
<td></td>
<td></td>
<td>$F_{chain}$</td>
<td>28500</td>
</tr>
</tbody>
</table>

Table 3.1; forces on the engine

$F_{chain}$ is applied on the rear sprocket and the reaction force is attached to the engine in the opposite direction. The other three forces are applied on the engine. The differential mounts are also made out of aluminum 6082-T6.

![Figure 3.6; stresses in the differential mounts](image-url)
For the alignment of the chain the forces in the y direction are important. The manufacturer prescribes the alignment of the chain as in figure 3.7.

![Figure 3.7; chain alignment](image)

The center to center distance ($L_{hoh}$) between front and rear sprocket is 221mm. This means the front edge of the rear sprocket can not have more then 0.7mm displacement in the y direction, see figure 3.8.

![Figure 3.8; displacement in y direction of differential mounts](image)
To fit the URE03 Torsen differential in the car some adjustments must be made to the differential mounts. In figure 3.9, spacer 1 (yellow) needs to be removed. The right triangle (green) must be attached to the other side of the right differential mount (brown). The parts (red) that connect the differential mounts to the rear bulkhead must be rotated 180 degrees. The top (blue) and bottom (not shown) plate needs to be remade.

Figure 3.9; differential mounts with Drexler differential (left), with URE03 Torsen differential (right)
4. Drive shafts

The driveshafts are the link between the differential and the rear wheel hubs. They also act as a torsion spring that filters torque peaks out of the driveline. Because of the wheel travel, from -30mm to +30mm, each driveshaft is equipped with two homokinetic couplings, in this case tripods.

- 4.1 Design goals.
  - design of the driveshaft
  - design of the tripod housings
  - axial fixation of the drive shafts
  - lubrication and dust caps for tripod joints

- 4.2 Design of the driveshaft.

The main question for the drive shafts is to design or to buy. A company in the USA sells complete formula student drivelines. The main reason to buy these drive train parts is manufacturing time. Drive shafts with splines are very labor intensive. The length can only be determined in the last phase of the design process of the car. This phase is also the busiest phase as it comes to manufacturing parts, the choice was made to purchase the drive shafts.

- 4.3 Design of the tripod housings.

On the differential side of the driveshafts the tripod housing is manufactured out of one piece with the stub axels. On the wheel hub side the tripod housing is manufactured out of one piece with the rear wheel hub. See chapter 7.

- 4.4 Axial fixation of the drive shafts.

Because of the wheel travel of the wheel hubs, the distance between the tripod housing on the differential side and the tripod on the wheel hub side is varying. This varying distance is fixed by a spring inside the driveshaft. The driveshaft has a hole of 10.5mm drilled all the way through. The hole is filled over the length of the axle with PVC filler rod. Between the filler rod and one end cap the spring is fitted, see figure 4.1 and 4.2.
Figure 4.1: Driveshaft assembly

Figure 4.2: Exploded view of driveshaft assembly
- 4.5 Lubrication and dust caps for tripod joints.

GKN Tripod grease is recommended by the manufacturer for lubrication and to prevent overheating caused by friction. Dust caps seal the gaps between the driveshaft and the tripod housing to keep the grid out and the grease in. see figure 4.3.

Figure 4.3; dust cap and dust cap in assembly
5. Wheel hub front

The front wheel hub connects the wheel to the upright. The main difference between the front and the rear wheel hub is the drive torque. Because the car is rear wheel driven there are no drive torques going to the front wheel hubs.

- 5.1 Design goals.
  o Reliability.
  o Strength calculations (FEA).
  o Low mass because of unsprung mass.
  o Apply feature for speed measurement.

- 5.2 Reliability.

There are four forces who affect the front wheel hub, a force in the x direction (accelerating or braking), in the y direction (cornering), the z direction (wheel travel, bump) and the brake torque.

The magnitudes of the forces in the y and z direction are calculated during the multi-link suspension design. For four situations a simulation was made, acceleration & braking, bump, cornering and slalom. The forces in x direction and the brake force are calculated in appendix x.

The maximum forces and brake torque are listed in table 5.1.

<table>
<thead>
<tr>
<th>Direction</th>
<th>Force</th>
<th>Situation</th>
</tr>
</thead>
<tbody>
<tr>
<td>$F_{\text{brake}}$</td>
<td>3800 N</td>
<td>Braking</td>
</tr>
<tr>
<td>X</td>
<td>1550 N</td>
<td>Acc/Brake</td>
</tr>
<tr>
<td>Y</td>
<td>2500 N</td>
<td>Corner</td>
</tr>
<tr>
<td>Z</td>
<td>3750 N</td>
<td>Bump</td>
</tr>
</tbody>
</table>

Table 5.1: Forces on wheel hub

The wheel hub is restricted to several dimensions namely:
1. inner diameter of the wheel bearing (65mm)
2. spacing between the two wheel bearings
3. position of the brake disc flange
4. et value of the wheel (position of the wheel flange)

The important parts of a wheel hub are figured (figure 5.1) and listed (table 5.2) below.
The axial movement is fixed by a c-clip in groove no. 5. Any axial play can be determent by inserting a spacer with a specified thickness between the circlip and the inner wheel bearing.

5.3 Strength calculations (FEA).

With the forces and restricted dimensions known a hub can be designed. The main dimensions that can be varied are the wall thickness of the hub, the thickness’s of the flanges and the radii between the flanges and the hub. The FEA analysis is done in two parts. The first analysis represents the forces in the x, y and z direction applied on the wheel flange. The second analysis represents the brake forces. The reason why these two analyses are done separate is because when the brake torque is applied the wheel flange becomes the fixed world.
The wheel flange needs to have a minimum thickness of 8mm because of the wheel nuts. These are pressed in the wheel hub and have special grooves to clamp in the wheel flange.
The brake disc is mounted to the wheel hub with four bolts. In the fem analysis the brake forces are attached to one brake-disc mounting point. This is done to simulate a worst case scenario. If three bolts would vibrate lose during driving, the wheel hub stays intact. This is done because it is cheaper and easier to replace a brake disc then a wheel hub. The wheel hubs are made out of CrNiMo, for material properties see appendix G. This material is chosen because it is in store at the manufacturer and it is good resistant against fatigue. In the worst case scenario the maximum stress is 290 MPa. This is almost a factor three lower then yield stress of the material. Focused on reliability this is a good safety factor.

- **5.4 Low mass because of unsprung mass.**

Overall the stresses in the wheel hub stay far beyond the maximal allowable stresses of the material, this could result in a lighter wheel hub. The thickness of the hub is 1.5mm, just for the stresses it could be thinner but it would be vulnerable lying around the workshop. Also to prevent to much play between the wheel hub and the wheel bearing, the wheel hub needs a small tap with a hammer for assembling. To create a lighter hub a smaller wheel bearing could be chosen. Unfortunately the wheel bearings and upright design where not adjustable. The overall weight of one front wheel hub is 650 gram.
5.5 Apply feature for speed measurement.

The wheel speed sensors are used to compare the wheel speeds of each of the four wheels. For example, the wheel speeds between the front and back wheels is used for launch control. Launch control limits the amount of wheel slip during take off. The sensor is sorted out by the electronics department of the team. For the specifications about the sensor see appendix H. To make the wheel speed sensor work, holes of at least 8mm diameter needs to be drilled in the wheel hub. For this wheel hub holes of 9mm diameter are chosen so that when the sensor vibrates a little it still is able to see the holes. The holes are drilled behind the circlip so there are no forces that go through his section of the wheel hub. The mounting of the sensor will be done by the designer of the uprights.

Figure 5.4; final design wheel hub front
6. Wheel hub rear

As mentioned in chapter 6 (the front wheel hub), the main difference between front and rear is the driving torque. All other design goals are the same for the front and rear wheel hub.

- 6.1 Design goals.
  o Same as front wheel hub
  o Connection tripod housing to rear wheel hub
  o FEA of the rear wheel hub with drive torque

- 6.2 Design goals the same as front wheel hub.

For this design goals see chapter 5.

- 6.3 Connecting tripod housing to rear wheel hub

The first solution for this problem was to design a flange between the wheel hub and the tripod housing, see figure 6.1. three bolts connect the flange to the wheel hub and six bolts connects the tripod housing to the flange. The tripod housing can be purchased.

Figure 6.1; tripod housing, flange and wheel hub rear
The second solution is to manufacture the rear wheel hub and the tripod housing out of one piece, see figure 6.2. This solution makes the manufacturing of the wheel hub more complex but saves more than 500 grams. Therefore the second solution is chosen.

Figure 6.2; wheel hub rear

The dimensions of the front and rear wheel hubs are almost the same. An additional fem analysis is made to calculate the stresses in the hub applied by the driving torque, see figure 6.3. The rear wheel hubs are made out of the same material as the front wheel hub (CrNiMo, see appendix G). To create a worst case scenario, the force from the tripod to the tripod housing is applied on one ear of the tripod housing. The shape of the tripod housing is a copy of the internals of a tripod housing sold by the manufacturer, see appendix I.
The second solution creates two new design goals.

- Axial fixation of the driveshaft inside the wheel hub
- Implementation of dust cap and sealing inside the hub

### 6.4 Axial fixation of the driveshaft inside the wheel hub.

An axial stop for the driveshaft needs to be integrated in the wheel hub to prevent the tripod from popping out of the tripod housing. A nylon cup mounted on a PVC cup holder ensures the right axial location of the tripod, number 1 and 2 in figure 6.4. Numbers 1 and 2 are mounted by circlip number 3.
As seen in figure 6.2, a gear ring is implemented in the wheel hub for the wheel speed sensor. The grease for the tripod can escape through the holes. A PVC bush (figure 6.4, number 5) is pressed into the wheel hub. A dust cap will be mounted over the open end of the PVC bush.

### Table 6.1: parts in rear wheel hub assembly

<table>
<thead>
<tr>
<th>Number</th>
<th>Function</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Nylon cup</td>
</tr>
<tr>
<td>2</td>
<td>PVC cup holder</td>
</tr>
<tr>
<td>3</td>
<td>Circlip</td>
</tr>
<tr>
<td>4</td>
<td>Tripod housing</td>
</tr>
<tr>
<td>5</td>
<td>Dust cap holder</td>
</tr>
</tbody>
</table>

- **6.5 Implementation of dust cap and sealing inside the hub.**

As seen in figure 6.2, a gear ring is implemented in the wheel hub for the wheel speed sensor. The grease for the tripod can escape through the holes. A PVC bush (figure 6.4, number 5) is pressed into the wheel hub. A dust cap will be mounted over the open end of the PVC bush.
7. Conclusion

The main design aspect for the drive train was reliability. For the design of all the drive train parts a safety factor of at least two is used. If the drive train would fail, the chain will be the weakest link. The chain is easy and relatively cheap to repair and widely available. The moment the chain breaks the engine and rear wheels will be decoupled.

The second design aspect was the weight of the parts. Both the front and rear wheel hubs are lighter than previous year. The drive shafts and final drive kept the same weight as last year. In case of the Drexler differential, the differential is lighter. Only the differential mounts gained extra weight. This is because the differential mounts also function as engine mounts. Therefore the mounts are larger than last year.

Third important design aspect was the costs of the produced and purchased parts. The differential mounts are designed to be laser or water cut, which is cheaper than milling. The wheel hubs are the only parts that need CNC machining. If the hubs would be designed to manufacture them in a conventional way they would gain weight. For the wheel hubs this is particular important because they not only affect the total car weight but also the unsprung mass of the car. For suspension reason the weight of the unsprung mass needs to be as low as possible. In case of the driveshaft, if manufacturing time allows they still can be manufactured.

8. Recommendations

- During my bachelor final project a master thesis was running on the implementation of a constant velocity transmission (CVT) in a Formula Student racecar. Part of this thesis was measure the torque in the drive shaft with strain gauges. When the results of these measurements are complete, the calculated torque with the schematic model of the drive line can be checked.
- To reduce the unsprung mass even more, the possibility to design aluminum wheel hub can be investigated.
- Some teams run successfully a belt final drive instead of a chain final drive. This could reduce the weight of the final drive.
9. Source list

- **Literature**

[1] Dictaat Constructieprincipes 1 (nr 4007), 2000


- **Internet**


[3] www.regina.it Chain


Appendix A

Schematic model of the URE04 driveline

Massatraagheden aandrijflijn

J1 = motor + ingaand tandwiel primary gear
J2 = uitgaand tandwiel primary gear + koppeling + ingaande tandwielen bak
J3 = uitgaande tandwielen bak + voortandwiel
J4 = achtertandwiel + differnetieel + homokineet + homokineethuis
J5 = j6 = velg & band
J7= auto

Stijfheden aandrijflijn

k1 = ingaande as versnellingsbak
k2 = uitgaande as versnellingsbak
k3 = as achtertandwiel
k4 = steekas links + aandrijfas links + wielnaaf + band
k5 = steekas rechts + aandrijfas rechts + wielnaaf + band

Overbrengingen

i1 = primary gear
i2 = versnellingsbak
i3 = final drive

Spelingen

s1 = speling versnellingsbak
s2 = speling differentieel
Appendix B

The m-file; “model_aandrijflijn” (Dutch)

% model aandrijflijn URE04

% benodigde gegevens auto
m=320  % massa auto [kg]
rb=20.5/2 * 0.0254  % straal band [m]

% massatraagheden aandrijflijn [kg*m^2]
j1=0.05  % motor + ingaand tandwiel primary gear
j2=4.624e-3  % uitgaand tandwiel primary gear + koppeling + ingaande tandwielen bak
j3=1.3414e-3  % uitgaande tandwielen bak + voortandwiel
j4=9.1125e-4  % achtertandwiel + differnetieel + % homokineet + homokineethuis
j5=0.2372  % velg & band
j6=0.2372  % velg & band
j7=m*rb^2  % auto

% stijfheden aandrijflijn [Nm.rad]
k1=3.60e4  % ingaande as versnellingsbak
k2=1.53e4  % uitgaande as versnellingsbak
k3=1.74e6  % as achtertandwiel
k4=2.26e4  % steekas links
k5=5.70e3  % aandrijfas links
k6=4.59e5  % wielnaaf
k7=3.72e4  % band
k8=4.44e4  % steekas rechts
k9=3.90e3  % aandrijfas rechts

% stijfheden aandrijflijn opgeteld [Nm.rad]
K1=k1
K2=k2
K3=k3
K4=(1/k4+1/k5+1/k6+1/k7)^(-1)
K5=(1/k8+1/k9+1/k6+1/k7)^(-1)

% overbrengingen
i1=79/41  % primary gear
i2=39/14  % versnellingsbak
i3=42/11  % final drive

% vereenvoudiging model
% massatraagheden
A=j1
B=j2/i1^2
C=j3/(i1*i2)^2
D=j4/(i1*i2*i3)^2
E=j5/(i1*i2*i3)^2
F=j6/(i1*i2*i3)^2
G=(2*j5+j7)/(i1*i2*i3)^2
% stijfhen
c1=K1/i1^2
c2=K2/(i1*i2)^2
c3=K3/(i1*i2*i3)^2
c4=K4/(i1*i2*i3)^2
c5=K5/(i1*i2*i3)^2

% vereenvoudigd model
% aannme diff is starre verbinding
jtot=\text{A+B+C+D+E+F+G}
c\text{tot}=\left(\frac{1}{c1+1/c2+1/c3+1/(c4+c5)}\right)^{-1}
we=\sqrt{c\text{tot}/j\text{tot}} \text{ % eigenfrequentie [rad/s]}

Answer m-file

jtot =

0.1052

c\text{tot} =

16.6571

we =

12.5836
Appendix C

The Matlab-file “eindoverbrening.m” (dutch).

```matlab
% eindoverbrenging.m
% uitreken van sneleheden per versnelling
% tandwieldiameters en lengte van de keting

close all
clear all
clc

% overbrengingen
% z = aantal tanden
z_voor = 11
z_achter = 50
i_eind = z_achter / z_voor;

% k2 motorblok (waarden in werkplaatshandboek kloppen niet, tanden geteld
% op assen)
i_primair = 79/41; % k2
i_versnelling = [39/14; 32/16; 32/20; 30/22; 29/24; 25/23]; % van k2
versnellingsbak
i_diff = 1;
i_totaal = i_primair * i_versnelling * i_eind * i_diff;
reductie = 1 ./ i_totaal;

% straal band = r
banddiameter_inch = 20.5;
banddiameter = (banddiameter_inch * 2.54) / 100;
r = banddiameter / 2;

% toerental motor
rpm = [1000: 500: 14000];

% hoeksneldheid motoras = omega
omega = (rpm/60) * 2*pi;

% snelheid auto
v1 = reductie * omega * r ; % in m/s
v = v1 * 3.6; % in km/h

figure
plot (rpm,v)
grid on
axis tight
title ('i final = 11/50')
xlabel('engine speed [rpm]')
ylabel('speed v[km/h]')
```
% steekcirkeldiameter tandwiel
% steekcirkeldiameter = d [m]
% s = steek ketting [m]
s = 0.015875;
d_voor = s / (sin(pi / z_voor))
d_achter = s / (sin(pi / z_achter))

% lengte ketting
% afstand tussen de middelpunten van de tandwielen = L_hoh [m]
% minimale afstand tussen tandwielen = L_hoh_min [m]
L_hoh = 0.201;
x = 2 * (L_hoh / s) + ((z_voor + z_achter) / 2) + ((z_achter - z_voor) / (2 * pi))^2 * (s / L_hoh);
aantal_schakels = ceil (x) % x naar boven afronden tot heel getal
lengte_ketting = aantal_schakels * s

Answers m-file

Figure 2.1 and;
aantal_schakels =

59

lengte_ketting =

0.9366
Appendix D

Bering forces on differential bearings

In chapter 2 the chain force is calculated, namely 28500N
The momentum balance is shown below.

\[ F_{\text{chain}} = 28500 \text{N} \]

\[ L_1 = 0.042 \text{m} \quad 2 \quad L_2 = 0.160 \text{m} \quad F_{\text{right}} = \]

\[ F_{\text{left}} = \]

L1 is the distance from rear sprocket to the left differential bearing.
L2 is the distance between the differential bearings.

Momentum balance at point 2
\[ F_{\text{chain}} \times L_1 = F_{\text{right}} \times L_2 \]
\[ F_{\text{right}} = 7500 \text{N} \]

\[ \Sigma F = 0 \]
\[ F_{\text{chain}} + F_{\text{right}} = F_{\text{left}} \]
\[ F_{\text{left}} = 36000 \text{N} \]
Appendix E

Material properties of aluminum 6082-T6 (www.matweb.com)

Aluminum 6082-T6

Categories: Metal; Nonferrous Metal; Aluminum Alloy; 6000 Series Aluminum Alloy

Material Notes: Material specs taken from SAPA / Indalex manual on extrusions.

Data points with the AA note have been provided by the Aluminum Association, Inc. and are NOT FOR DESIGN.

Composition Notes: Composition information provided by the Aluminum Association and is not for design.

Key Words: EU Numerical EN-AW-6082; EU Chemical AlSi1MgMn; AA6082; Sweden: SS-EN-AW-6082; Aluminium 6082-T6

Vendors: Click here to view all available suppliers for this material.

Please click here if you are a supplier and would like information on how to add your listing to this material.

Physical Properties

<table>
<thead>
<tr>
<th>Property</th>
<th>Metric</th>
<th>English</th>
<th>Comments</th>
</tr>
</thead>
<tbody>
<tr>
<td>Density</td>
<td>2.70 g/cc</td>
<td>0.0975 lb/in³</td>
<td>AA; Typical</td>
</tr>
</tbody>
</table>

Mechanical Properties

<table>
<thead>
<tr>
<th>Property</th>
<th>Metric</th>
<th>English</th>
<th>Comments</th>
</tr>
</thead>
<tbody>
<tr>
<td>Hardness, Vickers</td>
<td>95.0</td>
<td>95.0</td>
<td></td>
</tr>
<tr>
<td>Tensile Strength, Ultimate</td>
<td>290 MPa</td>
<td>42100 psi</td>
<td>wall thickness &lt; 5 mm</td>
</tr>
<tr>
<td></td>
<td>310 MPa</td>
<td>45000 psi</td>
<td>wall thickness &gt; 5 mm</td>
</tr>
<tr>
<td>Tensile Strength, Yield</td>
<td>250 MPa</td>
<td>36300 psi</td>
<td>wall thickness &lt; 5 mm</td>
</tr>
<tr>
<td></td>
<td>260 MPa</td>
<td>37700 psi</td>
<td>wall thickness &gt; 5 mm</td>
</tr>
<tr>
<td>Elongation at Break</td>
<td>10.0 %</td>
<td>10.0 %</td>
<td></td>
</tr>
</tbody>
</table>

Thermal Properties

<table>
<thead>
<tr>
<th>Property</th>
<th>Metric</th>
<th>English</th>
<th>Comments</th>
</tr>
</thead>
<tbody>
<tr>
<td>Thermal conductivity</td>
<td>170 W/m-K</td>
<td>1180 BTU-in/hr-ft²-°F</td>
<td></td>
</tr>
</tbody>
</table>

Material Components Properties

<table>
<thead>
<tr>
<th>Component</th>
<th>Metric</th>
<th>English</th>
<th>Comments</th>
</tr>
</thead>
<tbody>
<tr>
<td>Aluminum, Al</td>
<td>95.2 - 98.3 %</td>
<td>95.2 - 98.3 %</td>
<td>As remainder</td>
</tr>
<tr>
<td>Chromium, Cr</td>
<td>&lt;= 0.250 %</td>
<td>&lt;= 0.250 %</td>
<td></td>
</tr>
<tr>
<td>Copper, Cu</td>
<td>&lt;= 0.100 %</td>
<td>&lt;= 0.100 %</td>
<td></td>
</tr>
<tr>
<td>Iron, Fe</td>
<td>&lt;= 0.500 %</td>
<td>&lt;= 0.500 %</td>
<td></td>
</tr>
<tr>
<td>Magnesium, Mg</td>
<td>0.600 - 1.20 %</td>
<td>0.600 - 1.20 %</td>
<td></td>
</tr>
<tr>
<td>Manganese, Mn</td>
<td>0.400 - 1.00 %</td>
<td>0.400 - 1.00 %</td>
<td></td>
</tr>
<tr>
<td>Other, each</td>
<td>&lt;= 0.0500 %</td>
<td>&lt;= 0.0500 %</td>
<td></td>
</tr>
<tr>
<td>Other, total</td>
<td>&lt;= 0.150 %</td>
<td>&lt;= 0.150 %</td>
<td></td>
</tr>
<tr>
<td>Silicon, Si</td>
<td>0.700 - 1.30 %</td>
<td>0.700 - 1.30 %</td>
<td></td>
</tr>
<tr>
<td>Element</td>
<td>Ti</td>
<td>Zn</td>
<td></td>
</tr>
<tr>
<td>---------</td>
<td>-----</td>
<td>-----</td>
<td></td>
</tr>
<tr>
<td>Max.</td>
<td>&lt;= 0.100 %</td>
<td>&lt;= 0.200 %</td>
<td></td>
</tr>
</tbody>
</table>

**References** for this datasheet.

Some of the values displayed above may have been converted from their original units and/or rounded in order to display the information in a consistent format. Users requiring more precise data for scientific or engineering calculations can click on the property value to see the original value as well as raw conversions to equivalent units. We advise that you only use the original value or one of its raw conversions in your calculations to minimize rounding error. We also ask that you refer to MatWeb’s disclaimer and terms of use regarding this information. Click here to view all the property values for this datasheet as they were originally entered into MatWeb.
Appendix F

Calculation forces in x and brake direction

Figure x; schematic overview of wheel hub with forces in x direction

Calculations:
\[ g = 10 \text{m/s}^2 \]
\[ \mu_{\text{tire}} = 1.6 \]
\[ m_{\text{car}} = 320 \text{ kg (including driver)} \]
under full braking, 60% of the total weight on front wheels

\[ F_{x\text{total}} = \mu_{\text{tire}} \times m_{\text{car}} \times 60\% \times g \]
\[ F_{x\text{total}} = 3100 \text{N} \]

\[ F_{x\text{total}} \] is divided by two wheels, \( F_x = 1550 \text{N} \)

\[ F_{\text{brake}}; \text{momentum balance around wheel center} \]
\[ F_{\text{brake}} \times 105 = F_x \times 260 \]
\[ F_{\text{brake}} = 3800 \text{N} \]
## Appendix G

Material spec sheet CrNiMo ([www.matweb.com](http://www.matweb.com))

### Assab Steels 705M Machinery Steel

<table>
<thead>
<tr>
<th>Mechanical Properties</th>
<th>Metric</th>
<th>English</th>
<th>Comments</th>
</tr>
</thead>
<tbody>
<tr>
<td>Hardness, Brinell</td>
<td>275 - 335</td>
<td>275 - 335</td>
<td>Hardness supplied approx. 700°C tempering temperature. Quenching in oil ø 120. Testpiece hardened by oil quenching from 850°C.</td>
</tr>
<tr>
<td></td>
<td>240</td>
<td>240</td>
<td></td>
</tr>
<tr>
<td></td>
<td>320</td>
<td>320</td>
<td>500°C tempering temperature. Quenching in oil ø 120. Testpiece hardened by oil quenching from 850°C.</td>
</tr>
<tr>
<td>Hardness, Rockwell C</td>
<td>43.0 - 55.0</td>
<td>43.0 - 55.0</td>
<td>30 mm from the quenched end.</td>
</tr>
<tr>
<td></td>
<td>49.0 - 58.0</td>
<td>49.0 - 58.0</td>
<td>10 mm from the quenched end. tempering temperature 500°C. Hardened by oil quenching. tempering temperature 200°C. Hardened by oil quenching.</td>
</tr>
<tr>
<td></td>
<td>50.0</td>
<td>50.0</td>
<td></td>
</tr>
<tr>
<td>Tensile Strength at Break</td>
<td>790 MPa</td>
<td>115000 psi</td>
<td>R_m. 700°C tempering temperature. Quenching in oil ø 120. Testpiece hardened by oil quenching from 850°C.</td>
</tr>
<tr>
<td></td>
<td>1100 MPa</td>
<td>160000 psi</td>
<td>R_m. 500°C tempering temperature. Quenching in oil ø 120. Testpiece hardened by oil quenching from 850°C.</td>
</tr>
<tr>
<td>Tensile Strength, Ultimate</td>
<td>900 - 1100 MPa</td>
<td>131000 - 160000 psi</td>
<td>R_m. 700°C tempering temperature. Quenching in oil ø 120. Testpiece hardened by oil quenching from 850°C.</td>
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<tr>
<td>Tensile Strength, Yield</td>
<td>690 MPa</td>
<td>100000 psi</td>
<td>R_m. 700°C tempering temperature. Quenching in oil ø 120. Testpiece hardened by oil quenching from 850°C.</td>
</tr>
<tr>
<td></td>
<td>700 MPa</td>
<td>102000 psi</td>
<td>min. Re.</td>
</tr>
<tr>
<td></td>
<td>975 MPa</td>
<td>141000 psi</td>
<td>R_m. 500°C tempering temperature. Quenching in oil ø 120. Testpiece hardened by oil quenching from 850°C.</td>
</tr>
<tr>
<td>Elongation at Break</td>
<td>&gt;= 12.0 %</td>
<td>&gt;= 12.0 %</td>
<td>5XD. 500°C tempering temperature. Quenching in oil ø 120. Testpiece hardened by oil quenching from 850°C.</td>
</tr>
<tr>
<td></td>
<td>13.0 %</td>
<td>13.0 %</td>
<td></td>
</tr>
<tr>
<td></td>
<td>20.0 %</td>
<td>20.0 %</td>
<td>5XD. 700°C tempering temperature. Quenching in oil ø 120. Testpiece hardened by oil quenching from 850°C.</td>
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<tr>
<td>Reduction of Area</td>
<td>&gt;= 45.0 %</td>
<td>&gt;= 45.0 %</td>
<td>500°C tempering temperature.</td>
</tr>
<tr>
<td></td>
<td>42.5 %</td>
<td>42.5 %</td>
<td></td>
</tr>
</tbody>
</table>
Quenching in oil φ 120. Testpiece hardened by oil quenching from 850 °C.

63.0 % 63.0 %

700 °C tempering temperature.
Quenching in oil φ 120. Testpiece hardened by oil quenching from 850 °C.

Impact Test

<table>
<thead>
<tr>
<th>Material Components Properties</th>
<th>Metric</th>
<th>English</th>
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<tbody>
<tr>
<td>Carbon, C</td>
<td>0.360 %</td>
<td>0.360 %</td>
</tr>
<tr>
<td>Chromium, Cr</td>
<td>1.40 %</td>
<td>1.40 %</td>
</tr>
<tr>
<td>Iron, Fe</td>
<td>95.69 %</td>
<td>95.69 %</td>
</tr>
<tr>
<td>Manganese, Mn</td>
<td>0.700 %</td>
<td>0.700 %</td>
</tr>
<tr>
<td>Molybdenum, Mo</td>
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<td>0.200 %</td>
</tr>
<tr>
<td>Nickel, Ni</td>
<td>1.40 %</td>
<td>1.40 %</td>
</tr>
<tr>
<td>Silicon, Si</td>
<td>0.250 %</td>
<td>0.250 %</td>
</tr>
</tbody>
</table>

Some of the values displayed above may have been converted from their original units and/or rounded in order to display the information in a consistent format. Users requiring more precise data for scientific or engineering calculations can click on the property value to see the original value as well as raw conversions to equivalent units. We advise that you only use the original value or one of its raw conversions in your calculations to minimize rounding error. We also ask that you refer to MatWeb’s disclaimer and terms of use regarding this information. Click here to view all the property values for this datasheet as they were originally entered into MatWeb.
Appendix H

Wheel speed sensor spec sheet
The used wheel speed has type no. IFRM 06P1 701/1
Appendix I

Technical drawing stub axle (www.taylor-race.com)