Design of a geared neutral transmission test rig

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Internal traineeship

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Eindhoven, March 2004
Table of contents

Introduction 2

Chapter 1: Working principles 3

Chapter 2: Test rig components 8

Chapter 3: Placement of epicyclic gear 13

Chapter 4: Simulation of the test rig 16

Conclusions and recommendations 18

Appendix A: List of symbols 19

Appendix B: List of components 20

Appendix C: Component drawings 22

Appendix D: Strength calculations 25
Introduction

All vehicles which are driven by internal combustion engines have essentially the same problem when taking off. The internal combustion engine has to rotate to operate while the vehicle is standing still. Still, torque on the wheels is needed to take off. The most common solutions to this problem are in the form of a clutch or a torque converter.

A normal clutch essentially consists of a set of circular friction plates on which the axial closing force can be controlled. The greater this force, the greater the torque which can be transferred through the friction plates. If the exerted torque on the clutch is larger than the maximal friction torque, the clutch slips and the transferred torque is equal to the friction torque.

At higher vehicle speeds the axial force is larger than needed for the torque to be transferred, then the clutch is fully closed, the friction plates are at equal speed.

A torque converter is a hydrodynamic coupling which can be simplified to two disks which are rotating close to each other. Between these disks is a fluid, mostly oil. One disk is connected to the engine, the other to the outgoing axle. The amount of torque transferred depends on the speed of the engine.

An uncommon, relatively new, option to solve the problem of vehicle stand still and takeoff is a so called 'geared neutral' transmission. This type of transmission combines a variator, a fixed gear and a planetary gear set. By using the capability of the planetary gear set to subtract rotational speeds and the continuously variable ratio of the variator, the outgoing axle can stand still and even rotate backwards. This means that the vehicle can stand still, and drive backwards while the ingoing axle of the transmission, the engine, is turning and no axles are disconnected.

The problem with this kind of transmission is that the vehicle only stands still at one exact speed ratio of the variator. Due to errors on this speed ratio, the exact ratio of the transmission will not be zero, the transmission will constantly switch between forward and reverse. Therefore, variator ratio control is not suitable for vehicle stand still. Another approach is to brake the outgoing axle of the transmission and controlling the variator output torque by controlling slip of the variator. Any errors in variator speed ratio will then be compensated by slip.

This last approach has not been proven yet though. To do this, a test rig is desired. This report describes the design of such a test rig.
Chapter 1: Working principles

Configuration
As mentioned in the introduction, a geared neutral transmission has three main components; the variator, an epicyclic gear and a fixed ratio gear.

The figure below shows schematically how the components are joined. The primary wheel of the fixed ratio gear (FR) and the primary pulley of the variator (VAR) are directly linked to the ingoing shaft of the transmission.

The secondary pulley of the variator is connected to the sun wheel of the epicyclic gear (EG), the secondary wheel of the fixed ratio gear to the planet wheel carrier. The annulus of the epicyclic gear is connected to the outgoing shaft of the transmission.

Figure 1.1: Schematic representation of a geared neutral transmission
**Ratio relationships**

The relation between the angular speeds of an epicyclic gear is:

\[ \omega_{sun} = \omega_{pla} + P(\omega_{pla} - \omega_{ann}) \]  

(1)

P is known as the planetary ratio. For the whole transmission this equation results in:

\[ \frac{\omega_{out}}{\omega_{in}} = \frac{1}{P}(1 + P)F - r_{var}\]  

(2)

The last equation shows when \( r_{var} \) is equal to \( (1+P)F \), the ratio of the whole transmission, \( r_{GNT} \), is equal to zero. This means that the outgoing shaft of the transmission is standing still, and therefore the vehicle as well, while the ingoing shaft can have a random number of revolutions.

If \( r_{var} \) is larger than \( (1+P)F \), \( r_{GNT} \) is negative. The vehicle is then driving backwards.

If \( P \) is chosen larger as one, which is normal for geared neutral configurations, the difference between minimal and maximal ratio of the transmission, compared to a normal variator transmission, becomes smaller. This difference is equal to the ratio difference of the variator divided by \( P \). Due to this drop in ratio range, the maximal ratio of a geared neutral transmission is too small for high vehicle speeds. This problem is solved by adding two clutches; one between the primary sides of variator and fixed ratio gear, and a second one within the epicyclic gear. The clutches are shown in the figure below.

![Figure 1.2: Schematic representation of a geared neutral transmission with two extra clutches](image)
At higher vehicle speeds clutch C1 is opened and clutch C2 is closed. In this situation the variator is used directly, as in a normal CVT transmission. In the figure below, the ratio of the variator is represented as a function of the ratio of the whole transmission. The ratio range of the variator as well as the value of $P$ is an example.

![Graph showing ratio of the variator at different transmission ratios](Image)

Figure 1.3: Ratio of the variator at different transmission ratios
Torque relationships

The torques in an epicyclic gear set are related to each other by the following formula:

\[ T_{\text{sun}} : T_{\text{plg}} : T_{\text{ann}} = -1 : (1 + P) : P \]  

(3)

For the whole transmission this equation results in:

\[ T_{\text{out}} = T_{\text{ann}} = T_{\text{in}} \frac{P}{(1 + P)F - r_{\text{var}}} \]  

(4)

In this equation no losses of the transmission due to friction and slip are taken into account. The formula shows when \( r_{\text{VAR}} \) is equal to \( (1 + P)F \), when the vehicle is standing still, the value of the outgoing torque of the transmission is infinite and is no longer dependant of the ingoing torque. This is only partly true because equation 3 states that the outgoing torque, \( T_{\text{ann}} \), is equal to \( P T_{\text{sun}} \). The torque on the sun gear is the torque on the secondary shaft of the variator, which is limited. The maximal torque of the transmission is therefore determined by the maximal allowable torque of the variator as well as the planetary ratio \( P \).

Variator control

When the vehicle is standing still, the ratio \( r_{\text{GNT}} \) of the transmission must be exactly equal to zero. For the ratio of the variator this means that it must be exactly equal to \( (1 + P)F \). However, when controlling the speed ratio of the variator there will always be a small error. Due to this error the transmission continuously switches between forward and reverse. Combined with play in the driveline this leads to a nervous neutral condition. If the vehicle is forced to stand still, for example by braking the wheels of the vehicle, the small control error of the variator speed ratio can lead to very high torques within the transmission. This is because the variator is forced to have a ratio of \( (1 + P)F \). Therefore, controlling the ratio of the variator is not an option for a vehicle standing still.

As mentioned, the ratio of the variator has to be exactly equal to \( (1 + P)F \) when the vehicle is standing still. Because this is impossible by variator speed control only, another approach is needed. This approach focuses on the outgoing torque of the transmission rather than speed ratio. With this approach the vehicle is held in place by braking the wheels, this is also done with vehicles which have a torque converter. When controlling the output torque of the transmission, the torques within the transmission are limited. They are proportional to the output torque because of the epicyclic gear. Play in the driveline is no longer a problem because the outgoing torque of the transmission is always in the same direction.
Controlling outgoing torque of the transmission is done by controlling the outgoing torque of the variator. With a V-belt type variator this is possible by allowing the belt to slip on the pulleys. The outgoing torque of the variator is then equal to the maximal torque which can be transferred through friction. This maximal torque is given by the following equation:

\[ T = \frac{2F_{\text{clamp}} \mu R}{\cos(\lambda)} \]  

In this equation \( F_{\text{clamp}} \) is the axial clamping force on the pulleys. By adjusting this value the outgoing torque of the variator can be controlled. The amount of slip of the variator is limited however. Too much slip leads to damage to belt and pulleys. Too little slip and a small change in the amount of slip leads to an unacceptable change in the outgoing torque of the variator. Therefore, the amount of slip has to be controlled as well as the axial clamping force.

**Advantages and disadvantages**

The advantages of a geared neutral transmission over conventional CVT transmissions are:

- No separate starting device needed
- No Drive-Neutral-Reverse set needed
- Hill-performance determined by variator rather than engine
- Better acceleration, the vehicle can launch at maximal engine power performance
- Ratio can be adjusted between reverse and forward continuously

The disadvantages of this concept are:

- Epicyclic gear as well as a fixed ratio gear needed
- Two extra clutches necessary for high vehicle speeds
- Fixed ratio gear transfers more than engine output power
Chapter 2: Test rig components

To be able to make a good choice for the components for the test rig, a list of demands has been made:

- Torque and slip control applicable
- Transmission must have a geared neutral point
- Specifications of a real vehicle must be approached
- Torques and speeds have to be measured
- Maximal numbers of revolutions around 3000 rpm
- Maximal torques on the variator around variator specification
- Safe for user and bystander
- As simple as possible and low-cost

Variator

A control of the amount of slip and torque of the variator has to be possible, as mentioned in the list of demands. Firstly, this means the amount of slip has to be measured. To do this, the difference between actual and theoretical ratio has to be determined. The actual ratio can be measured by measuring the primary and secondary speed. The theoretical ratio can be derived from the axial position of the pulley halves.

Secondly it is necessary for a slip control to be able to influence the amount of slip. In chapter one is mentioned that the maximal outgoing torque, which also determines slip, is dependant of the axial clamping force. This force should therefore be controllable.

In the Power Trains section, where the geared neutral transmission should be built, much research is done on a certain V-belt transmission. This is a transmission built by Jatco, type CK2. The axial clamping force on the pulleys can be controlled. Recently, a sensor for measuring the axial position of the pulley halves within this transmission has been developed. Within this transmission the speeds of the primary and the secondary pulley are measured by Hall sensors. Therefore the amount of slip of this variator can be determined.
The properties of this variator make it very suitable, therefore the CK2-transmission is chosen to be modified to a geared neutral transmission. Below is a picture of the CK2-transmission.

Figure 2.1: The CK2-transmission by Jatco

**Epicyclic gear**

The epicyclic gear has to answer to two demands; it has to be able to withstand the desired torques and it has to be possible to drive the sun gear, carrier and annulus. Because epicyclic gears which can be driven at all three parts are rare and a suitable epicyclic gear is used within the CK2-transmission, the choice has been made to modify this epicyclic gear. It is able to withstand the torques because it is matched to the variator.

The planetary ratio $P$ of this epicyclic gear is equal to $\frac{z_{\text{ann}}}{z_{\text{sun}}} = \frac{82}{56} \approx 1.46$. 
Fixed ratio gear

Now variator and epicyclic gear are chosen, a choice has to be made for the fixed ratio gear. First, the ratio of the gear is important. This determines the minimal and maximal ratio of the transmission. In the table below the minimal and maximal ratio of the transmission are shown as a function of the fixed gear ratio $F$.

<table>
<thead>
<tr>
<th>$F$</th>
<th>$r_{GNT \text{ min}}$</th>
<th>$r_{GNT \text{ max}}$</th>
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<tr>
<td>0.1</td>
<td>-1.37</td>
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<td>0.2</td>
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<td>0.6</td>
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<td>0.7</td>
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<td>0.9</td>
<td>-0.02</td>
<td>1.23</td>
</tr>
<tr>
<td>1.0</td>
<td>0.15</td>
<td>1.40</td>
</tr>
</tbody>
</table>

Table 2.1: Minimal and maximal ratios at different values of $F$

In the table above, it is clear that a ratio smaller than 0.1 or larger than 1.0 is not suitable. The transmission can only be in reverse or forward respectively. A fixed ratio between 0.5 and 0.8 gives the most realistic ratio range of the geared neutral transmission.

In accordance with the list of demands, the variator has to be loaded heavily. The maximal torque on the secondary pulley of the variator is about 220 Nm, this leads to a torque of 548 Nm on the carrier. The maximal speed of the ingoing shaft should be around 3000 rpm. When using a ratio $F$ of 0.7, the maximal power through the fixed ratio gear is about 120 kW.

The fixed ratio gear has to be placed more or less parallel to the variator, therefore the distance between the primary and secondary shaft is equal to that of the variator, this is 168 mm.

There are basically three possibilities for the fixed ratio gear: three conventional gears, a belt drive and a chain.

Because conventional gears can transfer much power per volume and need little maintenance, this is the first considered option. If only two gears are used, it would be difficult to realize the fixed shaft distance. Furthermore, the direction of rotation of the carrier would change. By using three wheels both these problems are solved. The middle gear can be placed in such a way that it touches both other gears. The main disadvantage of gears is the need for them to be lubricated by oil.

The second option which has been looked at is a belt drive. The main advantages of a belt are its low noise and that no lubrication is needed. A company which is specialized in belt drives and chains was asked to determine the minimal width of a belt which answers the demands. After some research the company reported it was not possible to transfer the amount of power on such a short distance with a belt drive.
The possibility to use a chain as a fixed ratio gear in the transmission has been researched by the same company. They found a chain which satisfies the specifications. The short distance and the high amount of power is still a problem, therefore the lifespan of the chain is only 21 hours at 120 kW. However, this is sufficient for the test rig, it is not often used at its maximal power and the experiments are often short. Furthermore, the chain is a cheap solution.

From the options above, the chain is chosen for its easier lubrication and low costs.

**Other components**

There are some components which are necessary for the test rig and have not been chosen yet. These components are described below.

- Two flywheels, one to simulate the inertia of the engine and another to simulate the inertia of the vehicle.

- A controllable brake to simulate air resistance, rolling resistance and slopes. This brake should also stop the flywheel on the outgoing side of the transmission at the end of an experiment.

- Two torque sensors to measure in- and outgoing torques of the transmission. These should have a sufficient range and accuracy.

- A motor which drives the test rig. This motor should have sufficient power to drive all components. It has to be controllable as well. If the motor is placed in line with the primary shaft of the transmission, the dimensions of the engine are also important because of the small shaft distance of the variator.

- A platform on which all components have to be placed. This should offer enough space to all components.

Technical details of the components which are present within the Power Trains section and are suitable for the test rig are given in appendix B.
In the figure below an example is given for the configuration of the test rig. Components from appendix B are used.

Figure 2.3: Possible configuration test rig

1: CK2-transmission
2: Epicyclic gear
3: Electrical motor
4: Torque sensor
5: Flywheel on mount
6: Brakes on mount
Chapter 3: Placement of epicyclic gear

A main component of a geared neutral transmission is the epicyclic gear. As mentioned before, a Drive-Neutral-Reverse set from a CK2-transmission is chosen to be used. This set is displayed in the figure below:

![Figure 3.1 Epicyclic gear from Jatco CK2](image)

Normally, there is a clutch present within the epicyclic gear which blocks it. This clutch is actuated in the drive mode of the transmission. This clutch may be used to realize the mode switch of the geared neutral transmission, needed for higher vehicle speeds. However, testing the behavior in the geared neutral point is the main goal of the test rig. It is also hard to actuate this clutch, therefore the clutch is removed from the epicyclic gear.

The first intention was to place all components of the test rig separately on a platform. The advantage of this is that all components can be chosen more freely, because there is enough space between them. But aligning components, fixing them in place as well as cooling and lubricating the epicyclic gear brings complications.
From the CK2-transmission, only the variator and hydraulic unit are needed. The torque converter is completely obsolete. By removing the torque converter, enough space is made to position the epicyclic gear. In the figure below, the CK2-transmission is shown without the torque converter.

Figure 3.2: CK2-transmission without torque converter

For the lubrication and cooling of the epicyclic gear, a choice between grease and oil has to be made. Lubrication with grease is not an option however, because the heat generated within the epicyclic gear can not be discharged properly. Therefore, for the lubrication and cooling of the epicyclic gear the choice has been made to use the present oil circuit of the CK2-transmission. Within the shaft, which is normally driven by the turbine of the torque converter, an oil channel is present. This channel normally controls the lock-up of the torque converter. This channel will be used as feed pipe. For the discharge of the oil, the normal discharge channel which is on a larger diameter of the feed pipe is not suitable. Due to the centrifugal force on the oil, the epicyclic gear must be completely filled before oil can be taken away. This gives large viscous losses because of the different speeds within the epicyclic gear. Therefore, the oil should be discharged in the radial direction. To do this, the room in which the epicyclic gear sits is covered with a metal plate, to make it oil tight. The discharging oil now drops in the torque converter housing and can enter the carter of the transmission through a tube. The plate can also function as a support of the transmission, if thick enough. This is also done in another test rig within the Power Trains section.
Another demand is that the epicyclic gear has to be driven at three points. The sun wheel has to be driven by the variator. On the secondary shaft of the variator (the primary shaft in normal use) is the DNR-set. This is an epicyclic gear which makes it able for the CK2-transmission to shift between forward, neutral and reverse. This is locked for the geared neutral transmission. The shaft normally driven by the turbine of the torque converter is then directly linked to the secondary pulley of the variator. To connect this shaft to the sun wheel of the epicyclic gear a tube has to be made with the spline of the shaft on the inside and the spline of the sun gear on the outside.

The annulus has to be connected to the outgoing shaft of the geared neutral transmission. To do this a shaft has to be made which fits in the back side of the annulus and which is held in place by a shrink connection.

The planet carrier must be driven by the secondary wheel of the fixed ratio gear. Because the carrier is not very suitable to be driven, the core is placed in a new housing, which has to be produced. An extra piece is made on the housing to drive the pump of the CK2-transmission. When the ingoing axle of the transmission is rotating, the pump is driven through the fixed ratio gear.

The fixed ratio gear, which has to be placed more or less parallel to the variator, does not fit in the same room as the epicyclic gear. Therefore, an extra shaft has to be brought out, coaxial with the outgoing shaft of the transmission. The secondary wheel of the fixed ratio gear can then be connected to this shaft by bolts. The whole design is shown in the figure below:

![Diagram of transmission components](image)

1: Carrier housing
2: Annulus
3: Sun wheel
4: Spline tube
5: Outgoing shaft
6: Carrier lid
7: Plate on CK2

**Figure 3.3: Section of placement epicyclic gear**

Construction drawings of the components which have to be produced are in appendix C.
Chapter 4: Simulation of the test rig

To predict the response of the test rig on different input signals, a simulation is desired. Before a simulation can be built, a model of the test rig is needed. The test rig is simplified to the model below:

The figure shows the model consists of a flywheel on the left side ($J_{in}$) on which a torque ($T_{in}$) can be exerted, a variator (VAR), and a flywheel on the right side ($J_{out}$) on which a torque ($T_{out}$) can be exerted. The variator in this model represents the whole transmission, therefore it can have a negative ratio or a ratio of zero as well as a positive ratio. Losses within the drive train are not taken into account in this model.

The equation of motion of this model is the following:

$$\dot{\omega}_{in} = \frac{(T_{in} - r_{GNT}T_{out} - \dot{r}_{GNT}J_{out}\omega_{in}r_{GNT})}{J_{in} + J_{out}r_{GNT}^2} \quad (6)$$

This equation of motion has been implemented in Matlab-Simulink as follows:

![Figure 4.2: The base of the simulation model](image)
The chosen input signals of this simulation are engine torque, braking torque and ratio change of the variator. In another file, all parameters and initial conditions of the variables can be prescribed.

The output signals of the simulation are the number of revolutions of the ingoing shaft as well as the ratio of the transmission. With these two variables, combined with the brake torque, all torques and speeds within the transmission can be calculated. These variables are calculated within the Simulink model. In the figure below is an example of the result of a vehicle launch simulation.

![Figure 4.3: Results of a vehicle launch simulation](image-url)
Conclusions and recommendations

First, an inventory of the necessary components for the geared neutral transmission test rig has been made and the available components have been assessed. Secondly a design for the placement of the epicyclic gear within the CK2-transmission has been presented. The strength of critical components in this design has been checked and attention is given to manufacturing and assembly.

A choice is made for the fixed ratio gear which has yet to be placed.

Finally a simulation is built which predicts the torques and speeds of significant parts of the geared neutral transmission. With this simulation, other parts of the test rig can be chosen and values of test input parameters can be determined.

However, the modification of the CK2-transmission is not finished. Some attention has to be given to mounting the fixed ratio gear, driving the primary pulley of the variator, modifying the hydraulic unit and adding an oil drain from the torque converter housing to the carter of the transmission.

Furthermore, the rest of the test rig has to be designed. Components have to be chosen and the positioning of them on the platform has to be looked at. Connecting the components is a issue as well.

On the other side, research has to be done on the theoretic part of the transmission. There is a simulation for variator slip control, but this has to be modified to model for a variator in a geared neutral transmission. Torque and slip control must be implemented to the test rig as well.

In the future it might be possible to modify the geared neutral transmission in such a way that shifting to high ratios is also possible, by adding two clutches.
## Appendix A: List of symbols

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>$c_b$</td>
<td>operation factor</td>
<td>-</td>
</tr>
<tr>
<td>$d$</td>
<td>shaft inner diameter</td>
<td>m</td>
</tr>
<tr>
<td>$D$</td>
<td>shaft outer diameter</td>
<td>m</td>
</tr>
<tr>
<td>$F$</td>
<td>fixed gear ratio</td>
<td>-</td>
</tr>
<tr>
<td>$F_{ax}$</td>
<td>axial force</td>
<td>N</td>
</tr>
<tr>
<td>$F_{clmp}$</td>
<td>axial clamping force on pulley</td>
<td>N</td>
</tr>
<tr>
<td>$J_{in}$</td>
<td>inertia flywheel engine side</td>
<td>kg m$^2$</td>
</tr>
<tr>
<td>$J_{out}$</td>
<td>inertia flywheel vehicle side</td>
<td>kg m$^2$</td>
</tr>
<tr>
<td>$P$</td>
<td>planetary gear ratio</td>
<td>-</td>
</tr>
<tr>
<td>$r_{GNT}$</td>
<td>ratio entire geared neutral transmission</td>
<td>-</td>
</tr>
<tr>
<td>$r_{VAR}$</td>
<td>ratio variator</td>
<td>-</td>
</tr>
<tr>
<td>$R$</td>
<td>running radius of push belt</td>
<td>m</td>
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<tr>
<td>$R_{p0,2}$</td>
<td>0,2 % yield limit</td>
<td>N/m$^2$</td>
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<td>radius friction surface</td>
<td>m</td>
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<tr>
<td>$T$</td>
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<td>torque on annulus</td>
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<tr>
<td>$z_{sun}$</td>
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<tr>
<td>$\omega_{pla}$</td>
<td>angular speed carrier</td>
<td>rad/s</td>
</tr>
</tbody>
</table>
Appendix B: List of components

**Variator**
The push belt variator has been built in a Jatco CK2-transmission, which is used in a Nissan Primera. The ratio range of the variator is 0.42-2.25. The maximal torque on the secondary shaft is about 220 Nm. The pulley sheave angle is 11 degrees.

**Epicyclic gear**
The epicyclic gear is normally used as a drive-neutral-reverse set within the Jatco CK2-transmission. The planetary ratio of this gear is $\frac{82}{56}$. The gear has an outer diameter of about 170mm and a overall thickness of 70 mm.

**Fixed ratio chain**
The chain to be used is a triplex chain with a pitch of 9.525. The overall width of this chain is 34.6 mm. The sprockets have 35 and 50 teeth respectively. The fixed ratio $F$ is therefore equal to 0.7.

The lifespan of this chain at 3000 rpm of the primary wheel and at 545 Nm on the secondary wheel is estimated to be 21 hours. At the same speed, but at a torque of 90 Nm this lifespan is estimated to 2500 hours.

**Flywheels**
A flywheel mounted in a support is available for the test rig. The inertia of this wheel is about 2.5 kg m$^2$. Three identical flywheels without a support are present, the inertia of each flywheel is about 0.5 kg m$^2$.

**Brake**
A support with two disk brakes is available within the section. These brakes are from a car and can easily support enough torque. The control of these brakes is not known yet though.

**Torque sensors**
Some torque sensors which are usually used in hydraulically driven test rigs are present. The torque capacity of these sensors in sufficient, but with a length of 350 mm and an outer diameter of 130 mm they are fairly big.
**Motor**
Two electrical motors have been looked at, one with a maximal power of 5 kW and another of 11 kW. The 5 kW motor is being used in another test rig, if this motor is chosen, it should be ordered. The 11 kW motor is available, but no control is present for it yet. Such a control has to be developed.

**Platform**
A platform is available within the section, this platform has a length of 3.8 m and a width of 0.95 m and is therefore quite long and narrow. The platform has only few component mounting points.
Appendix C: Component drawings

Carrier housing
Carrier lid
Spline tube

Outgoing shaft
Appendix D: Strength calculations

Before checking component strengths, maximal torques on the components have to be determined. Within the epicyclic gear, the torques are determined as following:
The engine which is normally fixed to the CK2-transmission has a maximal torque of 181 Nm. The torque converter amplifies this torque by a factor 1.8, the torque on the annulus of the drive neutral reverse set is then equal to 325.8 Nm. This leads to a torque on the sun wheel, which is connected to the secondary pulley of the variator, of 222.5 Nm. The carrier then is loaded by 548.3 Nm.

Determination of minimal tensile strength construction material
For a shaft which is loaded purely with a torque the following equations hold:

\[
\sigma_i = 0.8 \cdot R_{p0.2} \tag{7}
\]

\[
\tau_w = \frac{\sigma_i}{\sqrt{3} \cdot S_f} \tag{8}
\]

\[
W_w = \frac{\pi}{16} \cdot \frac{(D^4 - d^4)}{D} \tag{9}
\]

\[
T = W_w \cdot \tau_w \tag{10}
\]

Torque \( T \), outer diameter \( D \) and inner diameter \( d \) are known variables. When the safety factor \( S_f \) (2) is chosen, the minimal needed tensile strength of the construction can be determined. Together with this needed strength a material can be chosen.

The outgoing axle of the geared neutral transmission is a solid shaft with an outer diameter of 30 mm. This shaft is linked to the annulus of the epicyclic gear and is therefore loaded with a torque of 325.8 Nm. This leads to a minimal tensile strength of 266 N/mm\(^2\), most steels have a higher value than this.
The shaft which drives the carrier is placed coaxial with the outgoing shaft of the transmission. The critical point of this part is shown in the figure below.

![Diagram of carrier lid](image)

**Figure A4.1: Critical points of carrier lid**

The outer diameter $D$ of this shaft is equal to 80 mm, the inner diameter $d$ is 55 mm. The torque on this part is 548.3 Nm. The minimal tensile strength of the construction material of this part is 30.4 N/mm$^2$. All steels have a higher value than this.
Determination of number of mounting bolts

The torque on the different components of the carrier must be transferred by friction. The bolts which hold the different components in place are not allowed to be loaded on shear. They can only supply the axial force necessary for the friction. The allowable axial force needed in the bolts for them to develop enough friction is given by the equation below:

\[ F_{ax} = \frac{S_f \cdot c_b \cdot T \cdot \alpha_a}{\mu \cdot R_b} \]  

(10)

The bolts which fix the sprocket of the fixed ratio gear are mounted at a radius \((R_b)\) of 31 mm. The maximal torque \((T)\) on the carrier is 548.3 Nm. When the values of friction coefficient \(\mu\) (0.16), operation factor \(c_b\) (1.0), fixing factor \(\alpha_a\) (1.5) and safety factor \(S_f\) (2) are entered, the required axial force is 331.6 kN. The maximal allowable axial force on a M8 bolt of strength class 12.9 is 31.1 kN. Therefore 11 bolts would do, but due to symmetry a number of 12 is chosen.

The core of the old carrier is mounted on the new carrier housing at a radius of 60 mm. It is loaded by torque of 548.3 Nm as well. The required axial force \(F_{ax}\) is 171.3 kN. The maximal allowable axial force on a M6 bolt of strength class 12.9 is 15.3 kN. Therefore 12 bolts are needed.