Feasibility study of BST for truck application

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Chapter 1

Introduction

In this report the feasibility of the use of a retarder for a Brake Shift Transmission (BST) in a truck is discussed. The reference transmission is a ZF AStronic twelve speed transmission with retarder. The BST concept enables the vehicle of providing the wheels with torque during shift moments. This improves the gear shift quality. In conventional manual, and in automated manual transmissions as the AStronic the torque is interrupted during shifting. The BST system makes use of a brake and a planetary gear to handle the powerflow. Because a lot of trucks already have a retarder integrated in the transmission, the use of the retarder as brake in the BST application can be interesting. Because of the high powers involved, a retarder is ideal for supporting high brake forces. A comparison is made between different retarders in chapter 2. The specifications of the truck and transmission are studied in chapter 3. In chapter 4 a model of the system is made and the model is optimized for different criteria. Also suggestions are made for topologies of the system into the Astronic gearbox.
Chapter 2

Retarders in general

2.1 Introduction to the use of retarders

Retarders are used for generating a braking torque opposite to a rotational motion to slow a vehicle down. They are additional to the conventional friction brakes. Conventional friction brakes are capable of generating very high braking torques and absorbing energy at high rates for short periods. The disadvantage of these brakes is that they wear out because of the friction. Another disadvantage is the risk of fading. When a heavy truck is running downhill, large braking power is needed for a long period of time to stop the vehicle. Conventional friction brakes like drum brakes or disk brakes only have a limited amount of heat capacity and heat dissipation. To meet the braking requirement and to reduce wear in conventional brakes retarders are used.

Retarders are based on electro-magnetic eddy current braking or hydrodynamic braking. In this paper the eddy current brake and the hydrodynamic brake will be discussed. Because the brake has to be used in the BST concept engine brakes are not discussed.

The retarders can be positioned at different locations of the drive train. Retarders located at the engine side of the gearbox are called primary retarders and retarders at the drive shaft side of the gearbox are called secondary retarders. Primary retarders deliver a gear dependent braking torque which increases as transmission ratios become higher, and gears lower. They can therefore generate higher braking torques at low speeds. Disadvantages are possible engine stall, additional loads on the transmission and the interruption of braking power flow during shifting. Primary retarders are often integrated in to the transmission and combined with the torque convertor. The braking torque of secondary retarders is only a function of the vehicle speed. High braking torques are available for a wide velocity range. At high speeds large power can be consumed. To increase the braking torques at lower speed an additional step up gear can be used to increase the velocity of the retarder. This can also decrease the size of the retarder. A ratio of 1:2 is common.

The braking power of the retarder is controlled by hand at the steering column. Also automatic controls are available that are actuated by the braking pedal. The system can be integrated with ABS systems and can even be used as a 'cruise control' for downhill driving.
2.2 Hydrodynamic retarder

2.2.1 Working principle

The concept to use fluid as a working medium to oppose rotary motion and absorb energy is used for a long time. Froude invented a dynamometer in 1877 to measure torques. The hydrodynamic retarder consists of a rotor connected to the rotating shaft and a stator fixed to the retarder housing. The rotor is turning and generates a centrifugal force on the fluid. This forces the fluid to travel in a radial outwards flow. At the same time the fluid has a tangential speed due to the rotation of the rotor. When the fluid is centrifuged outwards in to the cavities of the stator (which stands still), the tangential speed is abruptly stopped, and the fluid flows back to the center of the turbine to enter the rotor again. This arresting and accelerating of the tangential velocity component of the fluid velocity and the momentum of the flow generates a braking torque. Because there is no work produced all energy will be turned in to heat by heating up the oil. The relationship of the retarder torque $T_r$ to the diameter $D$, the speed $w$, the oil density $\rho$ and the performance $\lambda$ is shown in the general formula 2.1

$$ T_r = \lambda \rho w^2 D^5 $$

(2.1)

The torque is dependent on $w^2$ and $D^5$. $D$ is an important factor. The performance coefficient
 normally is a function of the speed ratio $\nu = \frac{\omega_{a}}{\omega_{r}}$. In this case the speed ratio obviously is zero because the stator stands still. The coefficient $\lambda$ is now a function of the rotor speed, the blade angle and the fill level. Of course also $\rho$ is a function of the fill level because the density of air differs a lot from the density of the fluid, generally oil. The retarder is controlled by changing the amount of liquid in the circuit. The retarder fill level controls the braking torque of the retarder. Of course the retarder can produce no braking torque when the vehicle is at rest. The retarder fill level is controlled by pressure. The oil is retained in the retarder by this control pressure. The pressure on the retarder outlet is approximately proportional to the retarder torque. Because the rotor works as a pump it builds up pressure and the fluid wants to escape. A high value for the fluid density $\rho$ would be desirable to generate high torques. Considering figure 2.2 it is clear that the curve for 100% fill coincides with the equation for the retarder torque. For certain higher values of $\omega$ the generated torque no longer follows this curve. The reason for this is that the pressure at the retarder outlet is limited. When the speed increases to a certain value the pressure generated by the retarder is too high and there will no longer be a 100% fill.

2.2.2 Cooling

The continuous braking power is limited by the cooling capacity of the system rather than by the retarder itself. Often the cooling system is shared with the engine cooling system. During driving downhill the engine produces less power and needs less cooling capacity. This can be counteracted by an exhaust brake which generates heat in the engine. This heat also has to be cooled by the engine cooling system. The retarder is sometimes integrated into the transmission and shares oil and oil pump with the transmission. This decreases the weight of the retarder. Cooling capacity is currently limited to around 300 kW. For short periods of time higher braking powers can be developed. To get additional cooling also liquid to air heat exchangers can be installed. The temperature rise across the retarder can be calculated as from the following equation.

$$Q_f = \frac{P}{C_p \rho \Delta T}$$

(2.2)

Where $Q_f$ represents the fluid flow through the retarder, $P$ the power consumed, $C_p$ the specific heat of the fluid and $\Delta T$ the temperature rise across the retarder. A high specific
heat is of course desirable. Water has a high specific heat and a high density. Disadvantage is that water has a low boiling temperature, and a bad corrosion resistance and lubrication.

2.2.3 Power losses

To control the retarder an oil booster has to be controlled. Also a hydraulic pump supplies oil pressure. Fan losses occur when the retarder is not being used but still generates a small amount of torque. This is the result of the air circulating between the rotor and stator. There are devises to reduce these losses by engaging slides between rotor and stator.

2.2.4 Control of the retarder

The braking power is controlled by the oil fill level of the retarder. As mentioned before the retarder torque is approximately proportional to the outlet pressure of the retarder. This is of course for the area in figure 2.2 where the retarder is not filled 100%. Often a cylinder filled with oil and pressurized with air is used to regulate the pressure at the discharge port of the rotor (impellor). In this way the retarder can be filled fast to ensure a short response time.

2.3 Electrical retarders

2.3.1 Working principle

The electric retarder can develop negative powers reaching 500 kW during short times. This braking power can be enough to satisfy all braking requirements even without the use of friction brakes. It is estimated that an electrical retarder can deliver 80% of the duty which otherwise would have been delivered by the service brakes. These advantages are equal to the hydrokinetic retarder.

The principle of the electric retarder is based on the creation of eddy currents in a metal disc rotating between electro-magnets, which setup a force opposing the rotation of the disk. As long as the electro-magnets are not energized, the disk will rotate freely. When the electro-magnets are energized, the rotation of the disk is retarded and the energy absorbed is transferred into heat. The electro - magnets are powered with current from the batteries of the vehicle. The braking torque varies proportional to the value of the current through the coils of the electro magnets. In this way the torque is easy to control by changing the current. The eddy current electric retarder for vehicles is often mounted directly on the driveshaft. The reason for this is that the cooling fins can blow the air freely and spread the heat that is generated.

The electric retarder is relatively simple to produce yet it uses a complex electro-magnetic phenomena. The calculations made of these systems are often empirical or done with finite element programs.

The simple principle of why these eddy-currents work as a brake will be explained. If a current flows through a conductor, a magnetic field is produced with an intensity proportional to the current. In a coil for example the magnetic intensity equals:

$$H_m = \frac{N_i}{l_m} [A/M]$$  \hspace{1cm} (2.3)
Figure 2.4: Principle of electric braking

Figure 2.5: Electrical disk brake
The flux density $B = \mu_0 H$, the flux then equals:

$$\Phi_m = B_mA_m = \frac{NiAm\mu_m}{l_m} \quad (2.4)$$

The flux can be calculated as a function of the current.

Faraday's law: induced voltage in a coil due to the time rate of change of flux times the amount of windings

$$e(t) = N \frac{d}{dt}(\Phi(t)) \quad (2.5)$$

This means that a constant current through a coil will impose no voltage on the coil. This can be understood because the coil then has no electric resistance. From Faraday's law can be learned that a changing flux will induce a voltage.

In case of the eddy current brake, the coils perpendicular to the rotating disk will produce a flux as a result of the current through these coils. These flux lines pass through the disk. As long as the disk rotates, the flux on a fixed point on the disk will change. This time rate change of flux will induce a voltage in the disk. Alternating eddy currents are created within the disk with strength proportional to the flux. The eddy currents wind themselves around the flux lines penetrating the disk, and will impose an apposing flux. Because of this apposite flux the braking force will be produced.

Figure 2.6 shows the torque as a function of the speed of the disk and the flux. It is typical for an a-synchronous machine, also called induction machine. Obviously the torque developed at stand still is zero, because no eddy currents can be created.

### 2.3.2 Cooling

The braking power that is produced by the eddy currents will be turned into heat. This heat has to be transported to the surrounding air by radiation and convection. Often the disk is equipped with fans to ventilate the heat to the air. The value of dissipated energy can be calculated by the following expression.

$$Q = MC_p \Delta T \quad (2.6)$$

9
The disc can reach temperatures of 400 degrees Celsius. Thermal stability will be achieved because the mass of air passing through the fan M will increase at higher speed, and the cooling capacity will increase with higher temperature differences. The torque will decrease with a rising temperature of the disk to a value that is half of the torque generated by a cold disk. The electric resistance increases and the magnetic permeability decreases at a rising temperature.

A disadvantage of this electric retarder is that the heat produced in the disk has to be cooled by the centrifugal fan on the disk. This means that it is difficult to mount this retarder in a closed environment near to the engine and gearbox. Maybe solutions can be found to solve this problem. One could think of transporting the hot air away from the retarder by an air channel fitted with additional fans. Water could also be used to cool the disk and transport the heat away, but the flow and cooling capacity should be great enough to prevent the water of boiling. Another option could be to transport the power electrical to another place. This would mean that another machine would have to be designed. An a-synchronous electrical generator would be used to generate the braking currents. The currents are not used to heat up the disk, but are transported to a heat exchanger through electrical wires. There are some disadvantages of this system. A-synchronous machines of this size fitted with coils on both stator and rotor would have a lot of weight and are expensive. Also the problem arises of transporting these large powers though electrical wires to a heat exchanger outside the vehicle. High-speed machines can be used to generate high voltages around 1000 Volts, but even then 400 Amperes are needed to transport the power of 400 kW. An average power line can only handle $10A/mm^2$. This means very heavy power lines would be needed.

2.3.3 Control of the retarder

Braking power is controlled by the amount of current through the coils. This is done by a set of relay boxes. The power is retracted from the vehicle battery. A cut-off system is coupled to prevent activation of the retarder at very low vehicle speeds when the driver presses the brake pedal.

2.3.4 Power losses

The power losses when the retarder is not in use (fan and bearing losses) are relatively small. When the retarder is in use it will need power for the battery to power the coils. The maximum power use is around 3.5 kW, but the average power consumption lies much lower. The average power consumption lies beneath 0.25 and 0.35 kW. A normal truck battery is able of providing this power.

2.4 Comparison

The electrical and hydraulic retarders are compared at a few criteria shown in table 2.1. The best advantage of the hydraulic retarder is the possibility to be integrated into the transmission. This is ideal for the combination with the BST system which also has to be integrated into the transmission. Electrical retarders are often fitted at the driveshaft to cool with surrounding air. The controllability has to meet certain requirements to use the system for BST purpose.
<table>
<thead>
<tr>
<th>Criteria</th>
<th><strong>Electrical retarder</strong></th>
<th><strong>Hydraulic retarder</strong></th>
</tr>
</thead>
<tbody>
<tr>
<td>Cost</td>
<td>Relatively cheap, sold separate</td>
<td>More expensive, but often available as option</td>
</tr>
<tr>
<td>Weight</td>
<td>200 - 300 kg</td>
<td>Dependent on design</td>
</tr>
<tr>
<td>Installation</td>
<td>Easy because of simple design; Not very big (350mm) diameter; Installed at driveshaft</td>
<td>Often already installed in transmission with shared engine cooling system</td>
</tr>
<tr>
<td>Power</td>
<td>Around 500 kW</td>
<td>Around 500 kW</td>
</tr>
<tr>
<td>Torque</td>
<td>High torques (3000 Nm) already from 1000 rpm, Maximum torque can drop at higher speed</td>
<td>High maximum torques (1500 Nm) starting at 750 rpm</td>
</tr>
<tr>
<td>Controllability</td>
<td>Good by applying current to the system</td>
<td>Reasonable by pressurizing the system oil is forced to the rotor</td>
</tr>
<tr>
<td>Power losses</td>
<td>Almost zero (bearing losses); Small energization losses</td>
<td>Small fan losses from air turbulence in the rotor</td>
</tr>
<tr>
<td>Speed</td>
<td>Around 4000 rpm</td>
<td>Around 3500 rpm</td>
</tr>
<tr>
<td>Cooling</td>
<td>Forced air cooling (not in closed environment)</td>
<td>Shared with engine cooling (engine overheating has to be prevented)</td>
</tr>
</tbody>
</table>

Table 2.1: Comparison on different criteria
Figure 2.7: ZF Intarder with braking graph. The vehicle speed is proportional to the retarder speed $\omega_r$ with $r_{FD} = 3.4$ and step up ratio 2. The torque is measured at the driveshaft which is twice the retarder torque.

2.5 The ZF intarder

The ZF intarder is a secondary hydrodynamic retarder consistent of a rotor with step up gear and stator integrated into a ZF AS-tronic transmission. The retarder system consists of the intarder, a heat exchanger and an electronic control unit. It is directly connected to the driveshaft with a step up gear as can be seen in figure 2.7. Therefor the braking torque can be shown as a function of the vehicle speed also in figure 2.7. With a maximum braking torque of 3200 Nm the intarder is able to deliver a braking power of 500 kW. This power is transported to the engine cooling system by the heat exchanger. The oil circuit of the intarder is shared with the transmission oil. The intarder can be operated in different ways. Manually the driver can choose from five different braking levels. In 'Bremsomat' mode the desired road speed is maintained by continuous variation of the braking torque. This can also be combined with cruise control ('Tempomat') mode. The retarder will never interfere with safety systems like ABS and ASR to ensure that the brake and driveline management systems work together at all times. The minimum speed at which the retarder can handle maximum torque is approximately 120 [rad/s], the maximum retarder speed is limited to approximately 3500 rpm = 366 [rad/s]. The moment of inertia is estimated from figure 2.7 using equation 2.7 for cylindrical objects. Exact data wasn’t available.

$$J = \frac{1}{2} \rho h R^4 \pi$$  \hspace{1cm} (2.7)

The retarder inertia is estimated as the sum of the shaft inertia and a disk of the same size as the rotor which is approximately $J_r = 0.125kgm^2$.

The retarder torque is controlled with a booster and several volume- and pressure valves. The booster is pneumatic pressurized to release an amount of oil into the system in a short time. The volume control valves are then controlled to apply the exact amount of oil pressure to realize the required braking torque. The oil pressure is proportional to the filling grade of
the retarder. After the booster is used for several times, it can take a while before it can be used again. See also [5], Appendix 1 of M. Pesgens. The response time is approximately 0.3 seconds.

The controllability of the intarder can be improved by using proportional actuation of the booster. An oil pump separate from the transmission can and the loss of the valves to the cooling system can lead to a shorter response time. Recent developments at ZF have already led to the improvement of the controllability of the intarder.
Chapter 3

Specifications of the truck and transmission

In this chapter the specifications of a Daf XF truck with ZF-Astronic transmission are reviewed. One of the reasons was that the project would be discussed with Daf trucks. Another reason is that the Daf XF is sold with an Astronic automated transmission, which is needed for the Brake Shift Transmission system.

3.1 Daf XF 95

The new DAF XF 95 truck is chosen for this project. The truck is available with different engine types. All of the engines are build up of a 12.6 liter 6 cylinder in line turbo intercooling engine with UPEC fuel injection. This engine can produce different kinds of power and torque characteristics. One of them can be seen in figure 3.1.

The XE 315 C is chosen, because of the limited torque of 1950 Nm. This because of torque limitations of the ZF Astronic transmission. The engine is able of producing power in the range of 1000 to 1900 rpm, but in practice for fuel efficient use 1500 rpm will be preferable as a maximum engine speed. The engine's maximum power is 315 kW at 1900 rpm. Note that the power increase from 1500 to 1900 rpm is only 5 kW.

Using the transmission data of the ZF Astronic a Traction diagram can be constructed that shows the force at the wheels for different gears as can be seen in figure 3.2. Table 3.1 shows data that is used to calculate driving resistance lines and Equation 3.2 is used to calculate the available traction force. The driving resistance lines are constructed using 3.1

The traction diagram shows that in lower gears a lot of traction force is available for acceleration, at higher ratios the traction force becomes very rapidiy. The hyperbolic line describes constant power. With a vehicle mass \( m_v \) of 20.000 kg the acceleration can easily be
Figure 3.1: torque curves and power curves
<table>
<thead>
<tr>
<th>parameter</th>
<th>value</th>
</tr>
</thead>
<tbody>
<tr>
<td>(i_{fd}) final drive ratio</td>
<td>3.4</td>
</tr>
<tr>
<td>(r_{dyn}) dynamic wheel radius</td>
<td>0.5</td>
</tr>
<tr>
<td>(\eta) total drive train efficiency</td>
<td>0.9</td>
</tr>
<tr>
<td>(m_v) vehicle mass</td>
<td>20000</td>
</tr>
<tr>
<td>(\rho) air density</td>
<td>1.199</td>
</tr>
<tr>
<td>(C_w) (C_w) value</td>
<td>0.75</td>
</tr>
<tr>
<td>A frontal area</td>
<td>10</td>
</tr>
<tr>
<td>(f_{rol}) rolling resistance parameter</td>
<td>0.015</td>
</tr>
<tr>
<td>(g) gravitational acceleration</td>
<td>9.81</td>
</tr>
</tbody>
</table>

Table 3.1: Verschillende parameters

<table>
<thead>
<tr>
<th>reduction</th>
<th>value</th>
<th>ratio</th>
<th>value</th>
</tr>
</thead>
<tbody>
<tr>
<td>(i_1)</td>
<td>15.858</td>
<td>(r_1)</td>
<td>0.063</td>
</tr>
<tr>
<td>(i_2)</td>
<td>12.326</td>
<td>(r_2)</td>
<td>0.081</td>
</tr>
<tr>
<td>(i_3)</td>
<td>9.565</td>
<td>(r_3)</td>
<td>0.104</td>
</tr>
<tr>
<td>(i_4)</td>
<td>7.435</td>
<td>(r_4)</td>
<td>0.134</td>
</tr>
<tr>
<td>(i_5)</td>
<td>5.873</td>
<td>(r_5)</td>
<td>0.170</td>
</tr>
<tr>
<td>(i_6)</td>
<td>4.565</td>
<td>(r_6)</td>
<td>0.219</td>
</tr>
<tr>
<td>(i_7)</td>
<td>3.474</td>
<td>(r_7)</td>
<td>0.287</td>
</tr>
<tr>
<td>(i_8)</td>
<td>2.700</td>
<td>(r_8)</td>
<td>0.370</td>
</tr>
<tr>
<td>(i_9)</td>
<td>2.095</td>
<td>(r_9)</td>
<td>0.477</td>
</tr>
<tr>
<td>(i_{10})</td>
<td>1.629</td>
<td>(r_{10})</td>
<td>0.614</td>
</tr>
<tr>
<td>(i_{11})</td>
<td>1.287</td>
<td>(r_{11})</td>
<td>0.777</td>
</tr>
<tr>
<td>(i_{12})</td>
<td>1.000</td>
<td>(r_{12})</td>
<td>1.000</td>
</tr>
</tbody>
</table>

Table 3.2: reductions en ratios
calculated:

\[ a(m/s^2) = \frac{220,000[N]}{20,000[kg]} = 11[m/s^2] \approx 1g \]

Of course this value can never be reached in practice because of limited wheel traction. A respectable value for acceleration would be around 1 - 2 m/s², meaning that the truck could reach 100 km/h in less than 15 seconds. The first gear ratio is chosen very low because of the demand to drive very slow with closed clutch in first gear (around 2 km/h).

### 3.2 ZF Astronic transmission

The ZF Astronic 12AS2301 twelve speed transmission consists of a three speed section, a splitter group and a rear mounted range change group in planetary design. The splitter group and range change have synchromeshes. Speed matching (synchronization) is performed by the engine control unit (which controls the engine speed) and the transmission brake. During upshifts the engine is throttled back by the control unit to match the following gear. To reduce shift time this can even improved by an engine exhaust brake. Two countershafts are used to ensure a compact equal distribution of torque. The maximum input torque is approximately 1900 Nm. Reduction range is between 15.86 and 1.0 as shown in table 3.1. The transmission and powerflow diagram are shown in figure 3.3.

The spread of the ratio's is between 1 and 15.858 in twelve gears. This means a step per gear of 15.858/11 = 1.2856 or 28.56%.

A calculation can be made of the different ratios of the gears in the transmission shown in 3.4. The diameters \( D_1 \) to \( D_8 \) are diameters of the gears. \( D_a \) is the diameter of the annulus and \( D_s \) the diameter of the sun of the planetary gear.
Figure 3.3: Original transmission and powerflow diagram
The following values for $R_1$, $R_2$, $R_3$, $R_4$ and $R_p$ are calculated:

$$i_1 = 15.858 = \frac{D_A}{D_B} \left( \frac{D_A}{D_B} + 1 \right) = R_1R_4R_p$$
$$i_2 = 12.326 = \frac{D_A}{D_B} \left( \frac{D_A}{D_B} + 1 \right) = R_2R_4R_p$$
$$i_3 = 9.565 = \frac{D_A}{D_B} \left( \frac{D_A}{D_B} + 1 \right) = R_3R_4R_p$$
$$i_4 = 7.435 = \frac{D_A}{D_B} \left( \frac{D_A}{D_B} + 1 \right) = R_2R_3R_p$$
$$i_5 = 5.858 = \frac{D_A}{D_B} \left( \frac{D_A}{D_B} + 1 \right) = R_1R_2^{-1}R_p$$
$$i_6 = 4.565 = \left( \frac{D_A}{D_B} + 1 \right) = R_p$$
$$i_7 = 3.474 = \frac{D_A}{D_B} = R_1R_4$$
$$i_8 = 2.700 = \frac{D_A}{D_B} = R_2R_4$$
$$i_9 = 2.095 = \frac{D_A}{D_B} = R_1R_3$$
$$i_{10} = 1.629 = \frac{D_A}{D_B} = R_2R_3$$
$$i_{11} = 1.287 = \frac{D_A}{D_B} = R_1R_2^{-1}$$
$$i_{12} = 1.000 = 1$$

The following values for $R_1$, $R_2$, $R_3$, $R_4$ and $R_p$ are calculated:

<table>
<thead>
<tr>
<th>Gear</th>
<th>reduction ratio</th>
</tr>
</thead>
<tbody>
<tr>
<td>$R_1$</td>
<td>1.286</td>
</tr>
<tr>
<td>$R_2$</td>
<td>1.000</td>
</tr>
<tr>
<td>$R_3$</td>
<td>1.629</td>
</tr>
<tr>
<td>$R_4$</td>
<td>2.701</td>
</tr>
<tr>
<td>$R_p$</td>
<td>4.565</td>
</tr>
<tr>
<td>$R_r$</td>
<td>0.500</td>
</tr>
</tbody>
</table>

Figure 3.4: calculation of the ratios
Chapter 4

Design of the Brake Shift System

4.1 BST concept explanation

The BST system consists of an adapted driveline with engine, clutch, automated manual transmission (AMT), a differential (final drive) and a brake. It can be seen in figure 4.1. The brake is embodied as a retarder in the truck configuration. In a normal driveline the torque would travel through clutch, AMT and final drive. The upper section of the system with the planetary gear, reduction and retarder can be used to support the engine torque during shifting between gears. Because all torque travels through the planetary gear during shift interruptions, the AMT is released from its torque and can be shifted without using the clutch at all.

The planetary gear is connected to the engine, the retarder and the output shaft of the transmission. In basic configuration the annulus is connected to the engine, the carrier to the load via the final drive and the sun to the retarder. The working of the system can be explained by looking at the speed characteristics of this planetary gear. Figure 4.2 shows the speed relations between sun (retarder), carrier (load) and annulus (engine). The engine speed increases before the shift from 1st to 2nd gear. During driving in fixed gears the retarder has no function. Losses are no greater than fan losses of conventional gearboxes with intertarder. The additional torque is supported by the gearbox, which is represented by the middle arrow in figure 4.2. As soon as the clutch is opened, there is no longer a fixed relation between engine and vehicle speed. In conventional situation the engine has to be throttled back to prevent the engine from increasing speed. This is no longer needed, because the engine's

![Figure 4.1: schematic BST system](image)
The specifications of the gearing have to meet certain requirements. In the example at 6th gear the retarders speed, which is proportional to \( \omega_s \), becomes zero. Beyond 6th gear even positive. The system can no longer work because the retarder would still provide torque in opposite direction, which is now the wrong direction. In practice it's not possible to use the retarder to support engine torque above 5th gear, because the maximum retarder torque is very limited at low speeds. See figure 2.2. The gear ratio of the driveline during shifting when torque travels through the planetary gear equals the ratio at which the retarders speed and \( \omega_s \) becomes zero, also called the ‘gear neutral ratio’ \( r_{GN} \). In this case the gear neutral ratio is equal to the 6th gear, which is much higher than first gear. This means that the torque at the wheels is lower. From figure 3.1 could be seen that there was a lot of excessive engine torque in lower gears. This can compensate the lower torque amplification. The available torque is also higher because of the inertia of the engine. A torque equal to \( T = J \omega \) is released from the engine’s kinetic energy. When \( r_{GN} \) is chosen higher the system can be used also for higher gears, but a price will be paid in losing torque during these shift moments. Below the 5th gear the system can even be used for powered down shifts. The retarder can support the torque, allowing the engine to speed up in order to match the lower gear. Unfortunately the system is useless above 5th gear. If the gear neutral ratio of the retarder could be shifted after the 5th gear to a higher gear, the system could be used again for the shifts above 5th gear.

### 4.2 Modeling of the system

The BST system is described in a model in figure 4.3. \( J_e \) is the total inertia of the engine, \( J_r \) is the inertia of the retarder, \( J_v \) the inertia of the total vehicle. \( T_e \) is the torque produced by the engine, \( T_a \) is the torque to the annulus of the planetary gear, \( T_k \) is the torque to the clutch and gearbox, \( T_s \) the torque at the sun of the planetary gear, \( T_r \) the retarder braking torque. The torque at the carrier of the planetary gear is \( T_c \), between \( r_c \) and \( J_c \) the torque is called \( T_2 \) and between the gearbox \( r_i \) and the final drive \( r_d \) the torque is \( T_1 \). The angular
engine speed is $\omega_e$. In most calculations the clutch is closed, and then $\omega_e = \omega_k$. The speeds of annulus, sun and carrier are $\omega_a, \omega_s$ and $\omega_c$. The ratios of the reductions can be defined:

$$r_c = \frac{\omega_a}{\omega_c}$$  \hspace{1cm} (4.1)
$$r_d = \frac{\omega_s}{\omega_c}$$  \hspace{1cm} (4.2)
$$r_i = \frac{\omega_t}{\omega_c}$$  \hspace{1cm} (4.3)

A basic planetary gear has three elements: a sun, carrier and annulus. Speed relations can be derived for the different elements leading to 4.4.

$$\omega_s = (z + 1)\omega_c - z\omega_a$$  \hspace{1cm} (4.4)

With $z = \frac{d_a}{d_s}$, the annulus diameter divided by the sun diameter. For torques in a planetary gear the following equations can be derived:

$$T_a = zT_s$$  \hspace{1cm} (4.5)
$$T_c = (z + 1)T_s$$  \hspace{1cm} (4.6)

Note that this means if one torque is known, the other two are known as well. $T_c$ is in the opposite direction of $T_s$ (figure 4.3). Three equations of motion can be derived for the inertias:

$$J_c\omega_c = T_c - \frac{T_s}{r_c} - T_k$$  \hspace{1cm} (4.7)
$$J_m\omega_m = T_b - T_s$$  \hspace{1cm} (4.8)
$$J_o\omega_o = T_d - T_v$$  \hspace{1cm} (4.9)

From the angular speed relation 4.4 a relation for the the gear neutral ratio $r_{GN}$ can be derived. From the definition of $r_{GN}$ follows that $\omega_s$ is zero at when $r_i = r_{GN}$. Equation 4.4 is rewritten in terms of $\omega_c$ and $\omega_s$ in 4.10.

$$\omega_s = (z + 1 - \frac{r_c}{r_{GN}})\omega_c = 0$$  \hspace{1cm} (4.10)
This can be rewritten in 4.11

\[ r_{GN} = \frac{r_c z}{z + 1} \quad (4.11) \]

The gear neutral ratio \( r_{GN} \) is a function of the 'kinematic ratio' \( z \) and the ratio \( r_c \). Another relation is needed to solve the values of \( r_c \) and \( z \). From figure 4.2 can be seen that the \( \omega_{e,max} \) will be reached in first gear at maximum engine speed \( \omega_{e,max} \). Equation 4.12 can be rewritten to 4.14 in terms of \( \omega_s \) and \( \omega_e \) using 4.11.

\[ \omega_s = (z + 1) r_c \omega_e - z r_{GN} \omega_e \quad (4.12) \]

\[ \omega_s = (z + 1)(r_i - r_{GN}) \omega_e \quad (4.13) \]

\[ z = \frac{\omega_{e,max}}{\omega_{e,max}} \left( \frac{1}{r_i - r_{GN}} \right) - 1 \quad (4.14) \]

From equation 4.11 follows

\[ r_c = r_{GN} \frac{z + 1}{z} \quad (4.15) \]

This implies that a certain value for \( r_{GN} \) defines the values for \( z \) and \( r_c \). Note that if the retarder is used for take off equation 4.14 doesn't hold. The speed relation in '0th' gear, when \( \omega_e \) is zero has to be used then.

### 4.3 Integration in the transmission

In the previous chapter the layout of the standard ZF-Astronic transmission is discussed. It is desirable to fit a part of the Astronic with an 'add-on' module to integrate the system without having to design a completely different transmission. This would save a lot of development effort in realizing a first prototype. The planetary gear has to be connected parallel to the transmission and to the retarder. The retarder is placed on the secondary side of the transmission. The favorable place for the planetary gear is the secondary side. The problem is how to make a connection between the primary (engine) side and the secondary side with the planetary gear?

A few options are evaluated to solve this problem:

1. Changing the shift order to one where the splitter \( R_1 \) doesn't change in the first 6 gears. The countershafts would be equal proportional to the engine torque during these first
6 gears. This was not an option because the 'range change' \( R_p \) has a ratio of 4.56 see figure 3.4. The first reduction ratio \( R_1 \) is limited because of its design to approximately 3. This means it’s not an option to keep the \( R_1 \) constant during the first 6 gears.

2. Using one of the two countershafts only for torque distribution to the BST planetary gear, and the other one for the normal transmission shifting doesn’t solve the problem. Because the splitter group changes between \( R_1 \) and \( R_2 \). An extra gear has to be fit in front of the splitter group, which makes it more complicated. Another disadvantage of using one of the countershafts is that the equal torque distribution is lost, leading to lower maximum torques for the transmission.

3. Changing the splitter group of the transmission, in order to keep \( R_1 \) attached to the engine during all times. This means that less gear ratios are possible. The 12 speed transmission would change into an 8 speed transmission illustrated in figure 4.5. Drawbacks are the higher gear ratio spread and the need for different gear ratios in the transmission.

4. A separate shaft from the engine (flywheel, primary side) to the secondary side of the transmission would solve the problem. An example is shown in figure 4.8. Notice that only the retarder unit is changed and an extra gear is added to the clutch housing. The clutch is left away. This option has the advantage that the 12 gear ratios can remain intact. The separate shaft would have to be added parallel to the original transmission in a new designed housing and would have to withstand the engine torque and inertia torques. This option can also be combined with a shiftable \( r_c \) or \( z \) value. The gear neutral ratio \( r_{GN} \) can then be shifted from the first \( r_{GN1} \) to the higher \( r_{GN2} \) to speed up the retarder.

The first two options do not appear to be feasible. The third option is further explained and calculated. The last option is mentioned as an alternative without the need to change the ratios and number of ratios of the gearbox.

4.3.1 Changing the splitter group

The first gear \( R_1 \) is fixed to the primary shaft. The splitter no longer changes between \( R_1 \) and \( R_2 \), but can still function as a prius direct. The powerflow diagram is shown in figure 4.5. Unfortunately there is a loss of gear ratios because the secondary shafts are permanently attached to the engine. Two possible layouts of the final module in this concept are shown in figures 4.5 and 4.6. Notice that they involve the same functionality with the output shaft of the transmission attached to the carrier, the engine to the annulus and the retarder to the sun of the planetary gear.

The gear ratios are chosen in a way to have the same gear spread between 0.063 and 1.000. This means a step per gear of 15.858 \( ^{\frac{1}{12}} \) = 1.484 or 48.4\%. In a similar way as in figure 3.4 from the previous chapter the ratios \( R_1 \), \( R_2 \), \( R_3 \), \( R_4 \) and \( R_p \) are calculated shown in 4.2.

The gear neutral ratio \( r_{GN} \) is chosen at \( r_6 \). At higher ratios the torque would drop more during shift moments. Also the power consumed by the retarder would increase as can be seen in the next section. \( Z \) values and reductions \( r_c \) are also calculated for higher \( r_{GN} \). The \( z \) value would drop under 1.5, involving a different connection of the planetary gear leading to a more complex transmission design. For this value of \( r_{GN} \) the \( z \) value and \( r_c \) can be calculated using equation 4.14 and 4.15 leading to \( z = 3.98 \) and \( r_c = 0.566 \).

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Figure 4.5: Changing the splitter group; First layout with powerflow diagram of this 8 speed gearbox.

Figure 4.6: Changing the splitter group; Second layout
Table 4.1: Reductions on ratios

<table>
<thead>
<tr>
<th>Reduction</th>
<th>Value</th>
<th>Ratio</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>(i_1)</td>
<td>15.858</td>
<td>(r_1)</td>
<td>0.063</td>
</tr>
<tr>
<td>(i_2)</td>
<td>10.685</td>
<td>(r_2)</td>
<td>0.093</td>
</tr>
<tr>
<td>(i_3)</td>
<td>7.200</td>
<td>(r_3)</td>
<td>0.139</td>
</tr>
<tr>
<td>(i_4)</td>
<td>4.851</td>
<td>(r_4)</td>
<td>0.206</td>
</tr>
<tr>
<td>(i_5)</td>
<td>3.269</td>
<td>(r_5)</td>
<td>0.305</td>
</tr>
<tr>
<td>(i_6)</td>
<td>2.203</td>
<td>(r_6)</td>
<td>0.453</td>
</tr>
<tr>
<td>(i_7)</td>
<td>1.484</td>
<td>(r_7)</td>
<td>0.674</td>
</tr>
<tr>
<td>(i_8)</td>
<td>1.000</td>
<td>(r_8)</td>
<td>1.000</td>
</tr>
</tbody>
</table>

Table 4.2: Gears and reductions

<table>
<thead>
<tr>
<th>Gear</th>
<th>Reduction</th>
</tr>
</thead>
<tbody>
<tr>
<td>(R_1)</td>
<td>1.000</td>
</tr>
<tr>
<td>(R_2)</td>
<td>1.4841</td>
</tr>
<tr>
<td>(R_3)</td>
<td>2.2025</td>
</tr>
<tr>
<td>(R_4)</td>
<td>3.2688</td>
</tr>
<tr>
<td>(R_p)</td>
<td>4.8513</td>
</tr>
</tbody>
</table>

Table 4.3: Relation between the kinematic ratio \(z'\) and the kinematic ratio \(z\) when changing the connections.

<table>
<thead>
<tr>
<th>Engine</th>
<th>Retarder</th>
<th>Wheels</th>
<th>(z')</th>
</tr>
</thead>
<tbody>
<tr>
<td>a</td>
<td>s</td>
<td>c</td>
<td>(z=4)</td>
</tr>
<tr>
<td>a</td>
<td>c</td>
<td>s</td>
<td>(-\frac{z}{z+1} = -4/5)</td>
</tr>
<tr>
<td>s</td>
<td>a</td>
<td>c</td>
<td>(\frac{z}{z+1} = 1/4)</td>
</tr>
<tr>
<td>s</td>
<td>c</td>
<td>a</td>
<td>(-\frac{z}{z+1} = -5/4)</td>
</tr>
<tr>
<td>c</td>
<td>a</td>
<td>s</td>
<td>(-\frac{1}{z+1} = -1/5)</td>
</tr>
<tr>
<td>c</td>
<td>s</td>
<td>a</td>
<td>(-(z+1) = -5)</td>
</tr>
</tbody>
</table>
Figure 4.7: Take off with use of the retarder, and shifting from $r_{GN1}$ to $r_{GN2}$ in 6th gear while speeding up the retarder

If the engine could be connected to the sun instead of the annulus, the configuration would become simplified. This is not possible because the gear layout of the planetary gear could not be changed into a proper value as can be seen in 4.3. A value for $z$ between $1.5 < z < 5$ implies a conventional planetary gear.

### 4.3.2 A separate shaft with shiftable $r_{GN}$

If a separate shaft apart from the unchanged transmission is used, the gears and layout of the transmission can be left intact. Also a shiftable $r_c$ and $z$ can be installed to allow $r_{GN}$ to change. The system can be used for all twelve gears, because the first $r_{GN1}$ is chosen at $r_7$ (equals the 7th gear in the original gearbox) and the second $r_{GN2}$ at $r_{13}$, which is enough above $r_7$ to 'power shift' into 12th gear. The working can be understood by looking at figure 4.7. Driving off is done in 0th gear with no gear selected, by braking the retarder. The available wheel torque will be equal as in $r_{GN1} = r_7$ which of course is lower then in first gear.

Driving in 6th gear the retarder has to be speeded up while shifting from $r_{GN1}$ to $r_{GN2}$. Because of the inertia of the retarder some torque will be involved to speed up the retarder. A friction clutch to synchronize the gears may be needed. If for example the retarder is accelerated in 1 second from $\omega_{r,min} = 70 \text{ rad/s}$ in $r_{GN1}$ to $\omega_{r,max} = 360 \text{ rad/s}$, $\dot{\omega}_r = 290 \text{ rad/s}^2$ and a retarder inertia of $0.125 \text{ kgm}^2$ (equation 2.7) the power to speed up the retarder can be calculated as in the following equation:

$$P = T_r \omega_{r,max} \dot{\omega}_{r,max} = 13 \text{kW}$$

This power has to be dissipated in the synchronization device.
In figure 4.8 the layout of the transmission can be seen. The separate shaft can be connected directly to the engine. Torques through the shaft will be in the order of magnitude of engine torques. If the retarder is used for take off, the clutch is no longer needed. Also the friction brake in the conventional transmission is no longer needed because the BST system is used to synchronize the gears. This leaves a lot of space on the primary side of the transmission to fit the shiftable gear \( r_c \) to the shaft outside the transmission housing. The extra planetary gear with shiftable \( z \) value is integrated with the retarder unit. Also the ratio between the sun of the planetary gear and the retarder can be chosen freely between certain boundary's.

In order to change \( r_G, r_c \) and \( z \) can be shifted. In equation 4.11 the relation between \( r_c, z \) and \( r_G, r_c \) was derived. This equation still holds. The following boundary conditions have to be satisfied for the system to work:

1. **1st** boundary condition: The retarders maximum speed (366 rad/s) in first \( r_G1 = r_7 \) is reached by take off in '0th' gear. The vehicle speed \( \omega_v \) is zero and the engine speed will be around 1200 rpm = 125 rad/s. When the vehicle speed is zero \( \omega_v = 0 \). A speed relation between \( \omega_v \) and \( \omega_e \) can be derived according to equation 4.13 with \( r_l = 0 \).

\[
\omega_s = -(z + 1)r_G\omega_v
\]  
(4.16)

This can be rewritten into an expression for \( z \):

\[
z = -\frac{\omega_s - \omega_v}{r_G}\omega_v \]  
(4.17)

For \( r_G1 = r_7 \) a value of \( z = 5.78 \) is calculated. The value for \( r_c \) follows from equation 4.11 and equals 0.337

2. **2nd** boundary condition: The retarders speed at \( r_6 \) in \( r_G1 \) has to be above the retarders minimum speed (around 80 rad/s). Equation 4.13 shows the speed relation. This can be written in the form of equation 4.18.

\[
\omega_s = (z + 1)(r_6 - r_G1)\omega_v
\]  
(4.18)

With the already calculated values for \( z \) and \( r_c \) the retarders minimum speed will drop to approximately 70 rad/s in 6th gear.

3. **3rd** boundary condition: The retarders maximum speed in second \( r_G2 = r_{13} = 1.30 \) has to be reached in 6th gear at maximum engine speed. In order to change \( r_G \) both \( z \) or \( r_c \) can be shifted, or both.

If only \( z \) is changed, the new \( z \) can be calculated using 4.17. The new \( z \) will become -1.35. This will implies a different layout of the planetary gear (see table 4.3). \( r_c \) will remain 0.337.

Also \( r_c \) can be shifted. From equation 4.11 follows that the new \( r_c2 \) will be 1.53. The \( z \) value will then remain 5.78. With this new value for \( r_c2 \) the maximum retarder speed in 6th gear is -1383 rad/s, which is too high. To satisfy the boundary condition an extra reduction between the planetary gear and the retarder will be needed.

If both \( r_c2 \) and \( z \) can be chosen freely, the boundary condition can easily be satisfied by using equation 4.13. The value for \( z = 0 \), 80 and \( r_c = 2 \), 92.
4.4 Required torques and powers

4.4.1 Torques

Torques and powers are calculated for the option with changed splitter group, although the last option with a separate shaft probably will be favorable. Calculations for the other system can be performed in a similar way. Of course the wheel torque will drop less, because of the better matching $r_{GN}$'s.

A certain acceleration of the truck is presumed. A value of $2\text{m/s}^2$ is achievable during the first few gears of the truck. In the lower gears this acceleration is not limited by the maximum engine torque, but by the friction of the tyres (equation 4.19)

$$F_{x,\text{max}} = \mu F_z$$  \hspace{1cm} (4.19)

This seems a very acceptable value. After 10 seconds the truck would travel $20\text{m/s} = 72\text{km/h}$, which is even a fast acceleration for a normal car. The force needed to obtain this acceleration is $40,000\text{N}$. This means $T_d = F r_{\text{wheel}} = 20,000\text{Nm}$. With a friction coefficient $\mu = 0.8$ and a vertical load of $50,000 \text{ N}$ on the driven wheels the maximum driving force is $40,000\text{N}$. The down force on the driven wheels 25% of the total weight load. The acceleration can be expressed in $\dot{\omega}_v$. Because $v = \omega r$, $a = \dot{\omega} r$, $\omega_v = 4\text{rad/s}^2$. To maintain this acceleration during shift moments, the $\dot{\omega}_v$ should remain constant, and thus $T_d$ has to remain constant. Notice that for the first gears there is excessive torque from the engine available and that air resistance is neglected.

Torques during shifting can easily be derived when neglecting the inertias $J_r$ and $J_c$. The result is shown in figure 4.9. The wheel torque remains constant at the maximum value of $T_d = 20,000\text{Nm}$ for as long as possible. At fifth gear it drops because there is less torque
available. The retarder torque $T_r$ during shifting is 841 Nm. This torque can even be delivered in 5th gear, when the retarder speed $\omega_r$ has dropped to approximately 133 rad/s. This can be seen in figure 2.7. During shift moments the engine is throttled up to produce maximum torque of 1900 Nm. This results in a maximum shift torque $T_d$. The value of $T_d$ could become even higher if the torque as a result of slowing down the engine ($J_\omega \dot{\omega}_e$) is also added to the engine torque. This inertia part is only available during deceleration of the engine. If the engine speed should remain constant for a period of time to allow the transmission to engage the next gear, the inertia torque is zero.

From the torque relations 4.5 and 4.6 exact expressions for $T_r$, $T_e$ and $T_o$ can be derived if $T_d$ is known. These also include inertias.

\[
T_e = T_d r_d
\]
\[
T_s = \frac{T_d r_d}{(z + 1)}
\]
\[
T_r = J_\omega \dot{\omega}_r + T_s = J_\omega \dot{\omega}_r + T_d \frac{r_d}{(z + 1)}
\]
\[
T_o = J_\omega \dot{\omega}_e + T_2 = J_\omega \dot{\omega}_e + \frac{2 r_d^2 c_r T_d}{z + 1} = J_\omega \dot{\omega}_e + r_G N r_d T_d
\]
\[
T_v = J_\omega \dot{\omega}_e + T_d
\]

These three coupled differential equations are not solved, but a solution based on assumptions is proposed. If a shift time of 1 second is assumed, $\dot{\omega}_e$ can be calculated. The maximum engine speed during normal economic driving will not exceed 1600 rpm, which equals $\omega_e = 167\text{rad/s}$. At each shift the engine speed will drop from 1600 rpm to approximately 1100 rpm. The engine acceleration will be approximately:

\[
\dot{\omega}_e = -55\text{rad/s}^2
\]
The engine inertia is estimated from the size of the engine flywheel, similar as in equation 2.7. \( J_e \) is approximately 1.5kgm\(^2\), so the additional torque from slowing down the engine is:

\[
T_{e,\text{inertia}} = J_e \dot{\omega}_e = 82.5Nm
\]

With a wheel torque \( T_{d,\text{shift}} \) during shifting of 18.000 [Nm] the acceleration of the vehicle can be calculated.

\[
T_{d,\text{shift}} = J_v \dot{\omega}_v
\]

With \( J_v \) calculated as:

\[
J_v = m_v r_v^2 \text{wheel} = 5000kgm^2
\]

The vehicle acceleration is:

\[
\dot{\omega}_v = 3.6rad/s^2
\]

From the time derivative of equation ??, a known vehicle acceleration \( \dot{\omega}_v \) and engine acceleration \( \dot{\omega}_e \), the retarder acceleration \( \dot{\omega}_r \) can be calculated.

\[
\dot{\omega}_r = \frac{z+1}{r_d} \dot{\omega}_v - zr_c \dot{\omega}_e = 196rad/s^2
\]

Note that the retarder's maximum speed is negative \( \dot{\omega}_r = -366rad/s^2 \). The deceleration of the retarder requires an additional retarder torque. This additional torque equals \( J_r \dot{\omega}_r = 24.5Nm \).

### 4.4.2 Powers

When the engine speed and retarder speed are kept constant the powers delivered by the engine and dissipated by the retarder are:

\[
P_e = T_e \omega_e
\]

\[
P_r = T_r \omega_r
\]

Equation 4.18 describes a relation between \( \omega_r \) and \( \omega_e \) while driving in certain gear:

\[
\omega_r = ((z + 1)r_1 - zr_c) \omega_e
\]

Using the torque relation between \( T_r \) and \( T_e \) and the expression for \( r_{GN} \) (4.11) the equation can be rewritten in terms of powers \( P_e \) and \( P_r \) and \( r_{GN} \) in 4.21:

\[
T_r = \frac{T_e}{zr_c}
\]

\[
P_r = \left( \frac{r_1}{r_{GN}} - 1 \right) P_e
\]

This shows that in lower gears a large part of the engine power is dissipated in the retarder. For the example with \( r_{GN} = 0.453 \) driving in \( r_1 = 0.063 \) the power dissipated by the retarder is 86% of the engine power. This is only during shifting when the retarder supports the engine torque.
4.5 Performance expectations

The wheel torque during shifting drops to zero in drivetrains with manual transmissions due to the limited energy dissipation in the synchromeshes. Using the BST system the wheel torque will never drop to zero during shifting. As can be seen in figure 4.9 the wheel torque barely decreases during shifting. This will increase the drivability of the truck. Especially during lower gears the drive torques are relatively high, which results in high acceleration interruptions.

The shifting behavior can be improved during up- and downshifting. Especially during upshifting the oncoming gear can more easily be reached, because the vehicle no longer stops accelerating during shifting. In conventional drive trains the engine is rapidly decelerated during upshift interruptions to synchronize the next gear. This kinetic energy is lost in the engine (exhaust) brake. With the BST system a lot of the power is lost in the retarder, but part of it is still used to drive the truck and part of it comes from decelerating the engine. This will improve economic driving. Also more frequent shifts to economic gears can be performed without penalty.

The option with changed splitter group has the disadvantage of only 8 gears and the system can only be used until the 5th gear. Also the gear ratios of the Astronic have to be modified. If the option with a separate shaft with shiftable TGN is chosen the wheel torque during shifting will be higher because of the better matching rGN’s. The ratios of the Astronic can be left unchanged, only the retarder unit and clutch housing have to be modified. The clutch is also no longer needed because the retarder can be used to drive off.
Chapter 5

Conclusions

The possibility of the use of a retarder for supporting the drive torque during shifting through a BST system is investigated. A hydraulic retarder seems the best option, although controllability has to meet certain requirements. This needs to be investigated further. The ZF Astronic with Intarder is used to design alternative solutions. This is done in two different ways. The first option changes the ratios and shifting of the Astronic. During discussion at DAF trucks experts mentioned that the loss of ratios from 12 to 8 is not desirable (especially for long distance travel). The gears are needed for economic driving. Also the change of ratios in the Astronic would be very difficult and expensive. This led to the design of the other option with separate shaft and shiftable $\tau_{GN}$. Although the total system may seem more complex, less changes to the Astronic are needed. Better performance is expected with this system because of the better matching shift torques. During shifting the maximum torque at the wheels can almost be constant. The intarder can easily handle the torques and powers involved. The use of the retarder for BST application is feasible.
Bibliography


