Design of a New Transmission Concept

Master Thesis

Jos Rutten
Design of a New Transmission Concept

Master Thesis
Nr. DCT 2005.56

by

J.H. Rutten
s476809

DaimlerChrysler AG
Division Power Systems
Section TPC/PTH

Supervisors DaimlerChrysler AG
Dr. Dipl.-Ing. C. Gitt
Dipl.-Ing. D. Schnitzer

Supervisor Eindhoven University of Technology
Dr. P.A. Veenhuizen

Stuttgart, June 2005
Abstract

This thesis discusses the design and construction of a new transmission concept for DaimlerChrysler AG. The transmission concept has been developed for heavy duty vehicles in long haul traffic.

The main goal of the thesis is to check the feasibility of the new transmission concept in comparison to a conventional 16-speed transmission. The new transmission splits the engine power using the same planetary gear set as the conventional 16-speed transmission, which results in an improved performance. The comparison is based on characteristics like the transmission ratios, the maximum torque, the efficiency of the transmission and the shift strategy.

In order to get a structured working method the thesis follows the product development process, specifically the construction process. Both processes, the construction process being a component of the product development process, are seen as Stage-Gate processes which means that every stage in the process needs to be evaluated at a gate before the next stage can start.

The first step in checking the feasibility of the transmission is to find a range of transmission ratios that fits the field of activity of the heavy duty truck and to develop the assumption of certain vehicle parameters as the basis of obtaining a vehicle speed-force diagram. Once the most suitable transmission ratios are determined they can be used to determine the shift strategy, which in this case leaves a lot of freedom due to the automated shift mechanism. Since the strategy turned out to be feasible, the next step in the process was to determine the efficiency.

The efficiency of the transmission completely depends on the losses in the transmission, and in order to calculate the losses the power flow through the transmission needs to be analyzed. The power flow analysis demonstrated clearly that the new transmission has an improved efficiency in comparison to the conventional transmission.
An extra advantage of splitting the power in the transmission is the torque handling capacity; because with power-split the transmission can handle more power using the same part sizes.

The third step in the process was the development of the bearing layout, which is the basis for the new transmission. The layouts are split into two types: on the one hand there is the design with only one countershaft and on the other hand the design with two countershafts. In general there are six bearing layouts of which five are used for a two-dimensional construction; two constructions with only one countershaft and three constructions with two countershafts. After comparison of the constructions the best one is used for further development: a design with two counter shafts.

The fourth stage is the stage that is used to determine the dimensions of the countershaft brake and the synchro meshes of the split-group and range-group. These dimensions depend on the mass moments of inertia of every rotating part in the transmission.

Parallel to the main development process phases as described above, there are a number of small development processes aside like in this case the development of a synchro mesh, new sliding collar teeth, and a new oil supply.

The synchro mesh has been designed as a back-up in case the constant mesh, that is now used to connect the driving shaft, main shaft and hollow shaft, is not sufficient.

The new sliding collar teeth on the other hand are actually used in the new transmission since the conventional teeth where insufficient.

Finally the oil supply has been adjusted in order to guarantee the lubrication of the parts in every gear.

The fifth stage of the product development process is the assembly of all designs made along the process, which resulted in the new transmission design. This gives the answer to the most important question of this thesis: is the new transmission concept feasible? The answer is that the concept can indeed be feasible, meaning that it is worthy for further and more detailed development.
Samenvatting

Deze thesis behandelt het ontwerpen en construeren van een nieuw transmissieconcept voor DaimlerChrysler, waarbij de transmissie is ontwikkeld voor zware vrachtauto’s die grote afstanden moeten kunnen afleggen.

The voornaamste doelstelling van de thesis is de haalbaarheid van het nieuwe transmissieconcept te onderzoeken en deze te vergelijken met een conventionele transmissie. Het nieuwe concept splitst het motorvermogen in de transmissie over twee assen door gebruik te maken van dezelfde planetaire set als de conventionele transmissie; dit principe verbetert de prestaties van de transmissie.

De vergelijking tussen de twee transmissies is gemaakt op basis van eigenschappen zoals overbrengverhoudingen, de maximale momenten, de efficientie van de transmissie en de schakelstrategie.

Om een gestructureerde aanpak te garanderen volgt de thesis het productontwikkelingsproces en het constructiproces als deel daarvan. Beide processen, zijn zogenaamde Stage-Gate processen. Dit betekent dat elke stap in het proces wordt afgesloten met een evaluatie (een Gate), alvorens gestart kan worden met de volgend stap van het proces.

De eerste stap die genomen is om te haalbaarheid te onderzoeken, is het vinden van een gebied van overbrengverhoudingen die overeenkomen met het werkgebied van het voertuig. Om een goed beeld te krijgen zijn een aantal voertuigparameters aangenomen en zijn twee voertuig karakteristieken getekend.

Als de overbrengverhoudingen geoptimaliseerd zijn kan de schakelstrategie bepaald worden, waarbij overigens er een grote vrijheid onstaat door het gebruik van een geautomatiseerd schakelmechanisme. De schakelstrategie blijkt inderdaad haalbaar, waardoor de volgende stap van het proces gezet kan worden.

De efficientie van de transmissie is volledig afhankelijk van de verliezen die optreden in de transmissie en om deze verliezen te berekenen is de vermogensstroom door de transmissie geanalyseerd.

Na het analyseren van de vermogensstroom blijkt dat het nieuwe transmissie concept in de hogere versnellingen een verbetering is ten opzichte van de conventionele transmissie.
De derde stap bestaat uit de ontwikkeling van een nieuw lagerconcept, dat de basis zal vormen voor de constructie van de nieuwe transmissie. De ontworpen lagerconcepten kan men opdelen in twee groepen, enerzijds zijn er de concepten die gebruik maken van slechts één secundaire as en anderzijds de concepten die gebruik maken van twee secundaire assen. Na een selectie gemaakt te hebben, wordt van vijflager concepten een twee-dimensionaal ontwerp gemaakt om de constructieve haalbaarheid te testen; hiervan hebben twee concepten één secundaire as en drie concepten twee secundaire assen. Uiteindelijk is één ontwerp geselecteerd als de meest optimale basisconstructie.

De vierde stap in het proces is de stap die de dimensies bepaalt van de rem op de secundaire as en van de synchro meshes voor de split-groep en de naschakel-groep. Deze dimensies zijn voor het grootste gedeelte afhankelijk van de massa traagheden die optreden in de transmissie.

Parallel aan het algemene produktontwikkelings proces zoals dat hierboven is beschreven, zis er een aantal kleinere processen in de zijlijn, om te voorzien in de ontwikkeling van een nieuwe synchro mesh, vertanding en olie voorziening.

The synchro mesh is ontwikkeld als back-up voor de huidige klauwen koppeling die gebruikt wordt voor het koppelen van de aandrijfas, de hoofd as en de holle as.

De nieuwe vertanding is ontwikkeld met het oog op de hierboven genoemde klauwen koppeling, aangezien deze aanzienlijk breder is dan een conventionele koppeling.

Ten slotte is de olievoorziening aangepast, zodat in alle versnellingen de smering van de roterende onderdelen gewaarborgd blijft.

Tijdens de vijfde en laatste stap van het proces zijn alle ontwikkelde subsystemen samengevoegd tot een volledig nieuwe constructie.

Deze constructie geeft uiteindelijk het antwoord op de vraag of het nieuwe transmissieconcept haalbaar is of niet. Het antwoord is dat de transmissie zeker haalbaar kan zijn en dat het een goede optie is voor verdere ontwikkeling.
Zusammenfassung

Diese Diplomarbeit behandelt den Entwurf und die Konstruktion eines neuen Getriebes Konzept für DaimlerChrysler AG. Das Getriebe ist entwickelt für schwere Nutzfahrzeuge im Fernverkehr.


Der Vergleich basiert auf Merkmalen wie Getriebe Übersetzungen, maximales Drehmoment, Wirkungsgrad und Schaltstrategie.

Um eine strukturierte Arbeitsmethode zu bekommen folgt diese Diplomarbeit das Produkt Entwicklungs-Prozess und das Konstruktions-Prozess als Komponent davon. Beide Prozessen sind so genannte Stage-Gate Prozessen was heißt das man zuerst eine Stufe evaluieren muss als „Gate“ bevor man zum nächste Stufe gehen darf.

Die erste Stufe ist das finden von Getriebe Übersetzungen die passend sind für das Einsatzgebiet des schweren Nutzfahrzeugs. Deswegen sind verschiedene Fahrzeugparameters abgeschätzt und sind zwei Fahrzeugdiagrammen erstellt.

Wenn die passende Übersetzungen gefunden sind kann der Schaltstrategie untersucht werden, wobei es im diesem Fall mehr Freiheit gibt wegen die Automatisierte Schaltung. Die richtige Schaltstrategie ist gefunden und die nächste Stufe ist die Analyse des Wirkungsgrads.

Der Wirkungsgrad hängt völlig ab von den Leistungsflüssen ins Getriebe, deswegen sind die zuerst analysiert. Aus dieser Analyse erscheint das der Wirkungsgrad in die höheren Gänge signifikant besser ist als das konventionelle Getriebe.

Ein dazu kommender Vorteil ist das die Zahnräder bei gleichen Abmessungen mehr Drehmoment bearbeiten können.

Die dritte Stufe im Prozess ist die Entwicklung von einem neuen Lagerkonzept, das als Basis dienen kann für das neue Getriebe. Es gibt zwei unterschiedliche Haupt-Konzepten,
nämlich die Konstruktion mit nur einer Vorgelegewelle und die Konstruktion mit zwei Vorgelegewellen. Von den verschiedenen analysierten Konzepten ist die Version mit zwei Vorgelegewellen ausgewählt als das beste Konzept, und benutzt für die weitere Konstruktionsarbeit.


Zum Schluss sind alle Subsystemen assembliert zu einen neuen Getriebe-Konstruktion um die Hauptfrage dieser Diplomarbeit beantworten zu können. Ist das neue Getriebe Konzept Realisierbar? Ja, das Konzept ist realisierbar, was sagen will das es sehr interessant ist für eine weitere detaillierte Weiterentwicklung.
Acknowledgements

This master thesis was executed in the heavy duty transmission department TPC/PTH at DaimlerChrysler AG.

First of all I would like to thank all colleagues of the departments TPC/PTH and TPC/PTA for their professional information and support.

My special thanks however go to Dr. C. Gitt, my supervisor at DaimlerChrysler, and to Dr. P.A. Veenhuizen, my supervisor at Eindhoven University of Technology. They both gave me a lot of helpful advice and feedback on the work I performed.

I would like to thank Mr D. Schnitzer for his constructional advice on the subject, but also Mr C. Schupp, and Mr M. Herdemerten who were always prepared to provide useful information on the subject.

Finally I would like thank my family and everybody else who supported me in the creation of this master thesis.

Stuttgart, June 2005

Joseph H. Rutten
# Table of Contents

Abstract ..................................................................................................................... v
Samenvatting .......................................................................................................... vii
Zusammenfassung ................................................................................................... ix
Acknowledgements .................................................................................................. xi
Table of Contents .................................................................................................. xiii
The Construction Process Flow as used by the Thesis........................................... xix
List of Symbols ...................................................................................................... xxi
Aufgabenstellung .................................................................................................. xxv

Chapter 1. Introduction ............................................................................................. 3
  1.1 Definition of the Problem .................................................................................. 3
  1.2 Plan of Action ................................................................................................... 4

Chapter 2. Product Development Process .............................................................. 9
  2.1 Product Development Process .......................................................................... 9
    2.1.1 Customer Needs ......................................................................................... 9
    2.1.2 Target Specifications .............................................................................. 10
    2.1.3 Generate Concepts .................................................................................. 10
    2.1.4 Select Concepts ...................................................................................... 11
  2.2 Construction Process ....................................................................................... 11
    2.2.1 Basic Concept ......................................................................................... 11
    2.2.2 Generate Concepts .................................................................................. 14
    2.2.3 Concept Selection ................................................................................... 14
    2.2.4 Final Steps of the Process ...................................................................... 16
Chapter 3. Available Technology .............................................................. 21
3.1 Direct Gear vs. Overdrive ................................................................. 21
3.2 Progressive vs. Geometrical Gear Steps .......................................... 22
3.3 Synchro Mesh vs. Constant Mesh Transmission .............................. 23
3.4 A Conventional 16-Speed Transmission .......................................... 23
3.5 Power-Split Transmission ............................................................... 25
3.6 Advantages and Disadvantages Power-Split Transmission .............. 27
  3.6.1 Advantages .................................................................................. 27
  3.6.2 Disadvantages ............................................................................. 28

Chapter 4. Requirements ................................................................. 31
4.1 Customer Requirements ................................................................. 31
4.2 Requirements of the Development Team ........................................... 32

Chapter 5. Important Values .............................................................. 35
5.1 The Engine ...................................................................................... 35
5.2 Speed of the Rotating Parts ............................................................. 36
5.3 Vehicle Speed ................................................................................ 38
5.4 Shift Strategy .................................................................................. 42

Chapter 6. Power-Flow through the Transmission .............................. 49
6.1 Power Flow ..................................................................................... 49
  6.1.1 Conventional Power-Flow ........................................................... 49
  6.1.2 Standard Split Power-Flow .......................................................... 50
  6.1.3 Split Power-Flow with Backflow over the Gearwheels .................. 51
  6.1.4 Split Power-Flow with Backflow over the Main Shaft ................. 53
6.2 Efficiency of the Transmission ....................................................... 54
6.3 Forward-Gear as a Crawler ............................................................. 58

Chapter 7. Bearing Layout ................................................................. 65
7.1 Available Bearings .......................................................................... 65
Chapter 11. Conventional Sliding Collar Teeth ................................................................. 113
  11.1 Conventional Sliding Collar Teeth ............................................................................. 113
  11.2 Power-Split Sliding Collar Teeth .............................................................................. 114
    11.2.1 Concept A ........................................................................................................ 114
    11.2.2 Concept B ......................................................................................................... 115
  11.3 Concept Selection ..................................................................................................... 116
  11.4 Calculation of the Dimensions .................................................................................. 117

Chapter 12. Construction of the Oil Supply ................................................................. 123
  12.1 Introduction .............................................................................................................. 123
  12.2 Oil Supply for the New Transmission Concept ...................................................... 124
    12.2.1 Additional Electric Oil Pump ............................................................................ 125
    12.2.2 Additional Mechanical Oil Pump ................................................................. 126
    12.2.3 Single Oil Pump Located Below the Driving Shaft ....................................... 126
    12.2.4 Double One-way Clutch ................................................................................ 127
  12.3 Concept Selection ................................................................................................... 129

Chapter 13. Final Design ............................................................................................. 133
  13.1 The 2D-Construction of the Bearing Layout ......................................................... 133
  13.2 Split- and Range-Group Synchro Meshes .............................................................. 134
  13.3 Three Shaft Constant Mesh .................................................................................. 135
  13.4 Constant Gearwheel Meshes .................................................................................. 136
  13.5 Countershaft Brake ................................................................................................. 137
  13.6 The Retarder Gearwheel ......................................................................................... 138
  13.7 Three-Dimensional Illustration ............................................................................. 139

Conclusions and Recommendations ........................................................................... 143

Appendix A. Construction Process of a Transmission .............................................. 149

Appendix B. Speed of rotating parts ............................................................................. 150
<table>
<thead>
<tr>
<th>Appendix C.</th>
<th>Activated Parts in Every Gear</th>
<th>151</th>
</tr>
</thead>
<tbody>
<tr>
<td>Appendix D.</td>
<td>Matlab Script for Vehicle Speed Diagrams</td>
<td>153</td>
</tr>
<tr>
<td>Appendix E.</td>
<td>Shift Strategy</td>
<td>156</td>
</tr>
<tr>
<td>Appendix F.</td>
<td>Efficiency</td>
<td>165</td>
</tr>
<tr>
<td>Appendix G.</td>
<td>Bearing Characteristics</td>
<td>166</td>
</tr>
<tr>
<td>Appendix H.</td>
<td>2D Constructions of Concepts</td>
<td>168</td>
</tr>
<tr>
<td>Appendix I.</td>
<td>Pressure in Split- and Range-Group</td>
<td>173</td>
</tr>
<tr>
<td>Appendix J.</td>
<td>Radio Frequency Identification</td>
<td>175</td>
</tr>
<tr>
<td>Bibliography</td>
<td></td>
<td>179</td>
</tr>
<tr>
<td>List of Figures</td>
<td></td>
<td>183</td>
</tr>
<tr>
<td>List of Tables</td>
<td></td>
<td>187</td>
</tr>
<tr>
<td>Glossary of Terms</td>
<td></td>
<td>189</td>
</tr>
</tbody>
</table>
The Construction Process Flow as used by the Thesis
## List of Symbols

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Unit</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>$\alpha$</td>
<td>[deg]</td>
<td>Slope angle</td>
</tr>
<tr>
<td>$\alpha_R$</td>
<td>[deg]</td>
<td>Angle of the applied synchronizing force to the surface</td>
</tr>
<tr>
<td>$\Delta n$</td>
<td>[rpm]</td>
<td>Speed difference</td>
</tr>
<tr>
<td>$\Delta\omega$</td>
<td>[rad/s]</td>
<td>Angular velocity difference</td>
</tr>
<tr>
<td>$\rho$</td>
<td>[kg/m$^3$]</td>
<td>Air density</td>
</tr>
<tr>
<td>$\rho_l$</td>
<td>[kg/m$^3$]</td>
<td>Density of a lamella</td>
</tr>
<tr>
<td>$\eta$</td>
<td>[-]</td>
<td>Efficiency of the transmission</td>
</tr>
<tr>
<td>$\eta_0$</td>
<td>[-]</td>
<td>Efficiency of the planetary gear in stationary situation</td>
</tr>
<tr>
<td>$\eta_b$</td>
<td>[-]</td>
<td>Efficiency of a bearing</td>
</tr>
<tr>
<td>$\eta_{FD}$</td>
<td>[-]</td>
<td>Efficiency of the final drive</td>
</tr>
<tr>
<td>$\eta_G$</td>
<td>[-]</td>
<td>Efficiency of a gearwheel set</td>
</tr>
<tr>
<td>$\eta_{PS}$</td>
<td>[-]</td>
<td>Efficiency of the planetary gear in split power-flow situation</td>
</tr>
<tr>
<td>$\eta_{V}$</td>
<td>[-]</td>
<td>Additional losses of the transmission</td>
</tr>
<tr>
<td>$\omega_1$</td>
<td>[rad/s]</td>
<td>Angular velocity of the sun gear</td>
</tr>
<tr>
<td>$\omega_3$</td>
<td>[rad/s]</td>
<td>Angular velocity of the ring gear</td>
</tr>
<tr>
<td>$\omega_e$</td>
<td>[rad/s]</td>
<td>Angular velocity of the engine</td>
</tr>
<tr>
<td>$\omega_{pc}$</td>
<td>[rad/s]</td>
<td>Angular velocity of the planet carrier</td>
</tr>
<tr>
<td>$\dot{\omega}$</td>
<td>[rad/s$^2$]</td>
<td>Angular acceleration</td>
</tr>
<tr>
<td>$\mu$</td>
<td>[-]</td>
<td>Friction coefficient</td>
</tr>
<tr>
<td>$A$</td>
<td>[m$^2$]</td>
<td>Area</td>
</tr>
<tr>
<td>$A_l$</td>
<td>[m$^2$]</td>
<td>Total area of a single lamella</td>
</tr>
<tr>
<td>$A_{lamella}$</td>
<td>[m$^2$]</td>
<td>Area of the pressure surface of a single lamella</td>
</tr>
<tr>
<td>$A_R$</td>
<td>[m$^2$]</td>
<td>Total friction area for synchronizing</td>
</tr>
<tr>
<td>$A_V$</td>
<td>[m$^2$]</td>
<td>Vehicle frontal area</td>
</tr>
<tr>
<td>$c_w$</td>
<td>[-]</td>
<td>Air resistance coefficient</td>
</tr>
<tr>
<td>$d$</td>
<td>[m]</td>
<td>Working diameter of the applied synchronizing force</td>
</tr>
<tr>
<td>$d_{cyl}$</td>
<td>[m]</td>
<td>Cylinder diameter</td>
</tr>
<tr>
<td>$i_0$</td>
<td>[-]</td>
<td>Stationary gear ratio of the planetary gear</td>
</tr>
<tr>
<td>$i_{45}$</td>
<td>[-]</td>
<td>Gear ratio between 4$^{th}$ and 5$^{th}$ gearwheel</td>
</tr>
<tr>
<td>$i_{67}$</td>
<td>[-]</td>
<td>Gear ratio between 6$^{th}$ and 7$^{th}$ gearwheel</td>
</tr>
<tr>
<td>$i_{FD}$</td>
<td>[-]</td>
<td>Ratio of the final drive</td>
</tr>
<tr>
<td>$i_{FG}$</td>
<td>[-]</td>
<td>Transmission ratio in forward gear</td>
</tr>
</tbody>
</table>
\(i_{\text{min}}\) [-] Minimum transmission ratio
\(i_{PG}\) [-] Gear ratio of the planetary gear
\(f_R\) [-] Roll friction coefficient
\(F\) [N] Applied synchronizing force
\(F_A\) [N] Air resistance
\(F_R\) [N] Roll resistance
\(F_S\) [N] Slope resistance
\(F_{\text{tooth}}\) [N] Force acting on the sliding collar tooth
\(F_{V,id}\) [N] Ideal vehicle force
\(g\) [m/s\(^2\)] Gravity
\(H\) [-] Harmful effects in problem solving
\(I\) [kgm\(^2\)] Vehicle inertia
\(j\) [-] Number of lamellae
\(J\) [kgm\(^2\)] Mass moment of inertia
\(J_C\) [kgm\(^2\)] Mass moment of inertia of the clutch
\(J_{CS}\) [kgm\(^2\)] Mass moment of inertia of the countershaft
\(J_{DS}\) [kgm\(^2\)] Mass moment of inertia of the driving shaft
\(J_{\text{red}}\) [kgm\(^2\)] Reduced mass moment of inertia
\(J_{RR}\) [kgm\(^2\)] Mass moment of inertia of the intermediate gearwheel
\(K_1\) [-] Gearwheel position of the slow gear in split-group
\(K_2\) [-] Gearwheel position of the fast gear in split-group
\(m_l\) [kg] Weight of a single lamella
\(m_V\) [kg] Vehicle mass
\(n_1\) [rpm] Speed of the sun gear
\(n_A\) [rpm] Speed of the sun gear
\(n_B\) [rpm] Speed of the planet carrier
\(n_C\) [rpm] Speed of the ring gear
\(n_e\) [rpm] Engine speed
\(N_3\) [rpm] Speed of the ring gear
\(p\) [N/mm\(^2\)] Air pressure
\(p_R\) [N/mm\(^2\)] Specific friction surface pressure
\(P_I\) [kW] Power flowing directly to the planetary gear
\(P_{II}\) [kW] Power flowing over the gearwheels to the planetary gear
\(P_A\) [kW] Transmission input power
\(P_B\) [kW] Transmission output power
\(P_{\text{Block}}\) [kW] Power transferred through the planetary gear rotating as a block
\(P_e\) [kW] Engine power
\(P_{e,\text{max}}\) [kW] Maximum engine power
\( P_m \) \([\text{kW}]\) Synchronizing power
\( P_{m,A} \) \([\text{W/mm}^2]\) Specific synchronizing power
\( P_{\text{roll}} \) \([\text{kW}]\) Power transferred through the planetary gear by rolling
\( P_V \) \([\text{kW}]\) Power losses
\( r_0 \) \([\text{m}]\) Inner radius of the part
\( r_1 \) \([\text{m}]\) Inner radius of the friction surface
\( r_2 \) \([\text{m}]\) Outer radius of the friction surface
\( r_d \) \([\text{m}]\) Working radius of the applied synchronizing force
\( r_{\text{dyn}} \) \([\text{m}]\) Dynamical tire radius
\( R_1 \) [-] Gearwheel position of the first gear
\( R_2 \) [-] Gearwheel position of the second gear
\( R_3 \) [-] Gearwheel position of the third gear
\( t_R \) \([\text{s}]\) Synchronizing time
\( T \) \([\text{Nm}]\) Torque
\( T_A \) \([\text{Nm}]\) Transmission input torque
\( T_B \) \([\text{Nm}]\) Transmission output torque
\( T_e \) \([\text{Nm}]\) Engine torque
\( T_{pc} \) \([\text{Nm}]\) Torque at the planet carrier
\( T_R \) \([\text{Nm}]\) Synchronizing torque
\( T_V \) \([\text{Nm}]\) Synchronizing losses
\( U \) [-] Useful effects in problem solving
\( v \) \([\text{m/s}]\) Synchronizing velocity
\( V \) \([\text{m/s}]\) Vehicle forward velocity
\( w_l \) \([\text{m}]\) Thickness of a single lamella
\( W \) \([\text{J}]\) Synchronizing work
\( W_A \) \([\text{J/mm}^2]\) Specific synchronizing work
\( z_p \) [-] Number teeth on the primary shaft
\( z_s \) [-] Number teeth on the secondary shaft
\( Z_1 \) [-] Number of teeth of the sun gear
\( Z_2 \) [-] Number of teeth of the planet
\( Z_3 \) [-] Number of teeth of the ring gear
Aufgabenstellung

Diplomarbeit:
„Entwurf eines wirkungsgradoptimierten Getriebes für Lkw“

Im Bereich Getriebeentwicklung der DaimlerChrysler AG werden derzeit Studien im Hinblick auf die Optimierung der Wirkungsgrade von Lkw-Getrieben durchgeführt. Es liegen grundsätzliche Konzepte vor, die geeignet erscheinen, um entsprechende Prototypen aufzubauen. Da der grundsätzliche Aufbau dieser Getriebe jedoch nicht mit dem konventioneller Gruppengetriebe übereinstimmt, ist eine Reihe von technisch relevanten Aspekten mittels konstruktiver Entwürfe zu bewerten.

In der Diplomarbeit sollen verschiedene Varianten des Grundkonzeptes in Form von 2D-Mittelschnittzeichnungen konstruiert werden. Diese sollen die Basis für die später optional mögliche Darstellung von Prototypen bilden. Im Rahmen der Arbeit sollen dabei unterschiedliche Anordnungsvarianten konstruktiv untersucht und bewertet werden, so daß auf dieser Basis eine Entscheidung für ein favorisiertes Konzept getroffen werden kann. Im ersten Schritt sind zunächst Kriterien für die Übersetzungswahl des Getriebes zu erarbeiten und zu dokumentieren. Neben der anschließenden konstruktiven Arbeit sind außerdem relevante Bauteile / Baugruppen zu dimensionieren. Die für die abschließende Berechnung der Bauteile zuständige Abteilung bei der DaimlerChrysler AG ist mit entsprechenden Daten zu versorgen.

Ferner soll auf Basis des Entwurfes eine grobe Kostenschätzung für das Getriebe vorgenommen werden. Hierfür ist die Zusammenarbeit mit den entsprechenden Fachleuten aus dem produzierenden Werk angestrebt.

Im Detail sind folgende Arbeiten durchzuführen:
- Erarbeitung und Dokumentation von grundlegenden Kriterien bei der Auslegung der Übersetzungen
- Anfertigung von konstruktiven Entwürfen des Getriebes (CATIA V4, 2D) und Auswahl eines favorisierten Konzeptes
- Konzipierung einer geeigneten Schaltung
- Dimensionierung relevanter Bauteile / Baugruppen
- Optional: Kostenbewertung.


Beginn der Arbeit: 01. 09. 2004
Bearbeitungsdauer: 40 Wochen
Design of a New Transmission Concept

Betreuung: DaimlerChrysler AG

Dr. Dipl.-Ing. C. Gitt
Abt. TPC/PTH
HPC B203, Werk 019
70546 Stuttgart
Tel.: 0711-17-59717
e-mail.: carsten.gitt@daimlerchrysler.com

Dipl.-Ing. D. Schnitzer
Abt. TPC/PTA
HPC A504, Werk 019
HPC A504, Werk 019
70546 Stuttgart
Tel.: 0711-17-59996
e-mail.: detlef.schnitzer@daimlerchrysler.com
Chapter 1.
Chapter 1. Introduction

The conventional 16-speed transmission is a transmission that is used in heavy duty trucks for long haul traffic. These trucks are mostly driven in the upper gear, where the efficiency of the transmission should be as high as possible. The highest efficiency is the one where the transmission ratio is exactly equal to one.

In the conventional 16-speed transmission the power always enters the planetary gear at the same point, namely the sun gear. The planetary gear however has the characteristic that the power can enter at the sun gear and at the ring gear simultaneously. The power-split transmission concept uses this characteristic to split the engine power into a power flowing directly to the sun gear and a power flowing through the countershaft to the ring gear. The planetary gear then combines these powers, resulting in the output power of the transmission.

In the upper transmission gears the gearwheel loads are reduced by the power-split concept, since the power is not only flowing through the countershaft, but also through the main shaft. This results in a couple of advantages in comparison to the 16-speed transmission. The transmission efficiency improves in the upper gears since the load on the gearwheels is reduced. The transmission can handle more power using the same size of gearwheels. In the highest gear where the transmission ratio is equal to one, the countershaft will be decoupled; if the countershaft doesn’t rotate the gearwheels will also be standing still, again resulting in an improved efficiency.

1.1 Definition of the Problem

In the section above the power-split concept is defined as an improvement of the conventional 16-speed transmission. The main goal of this thesis is to check whether the new transmission concept will indeed be feasible as a transmission that can be implemented into future heavy duty trucks.

The questions that will be asked follow from the expectations written in the introduction of this chapter.
- The new transmission will have a new range of transmission ratios. Are these transmission ratios suitable for long haul traffic?
- Since the transmission ratios differ from the conventional transmission the shift patterns should be checked for there feasibility. These shift patterns lead to a certain shift strategy which also has to be checked for its feasibility.
- Is it true that the efficiency of the transmission will be improved in the upper gears?
- Is the transmission able to handle more torque than the conventional transmission?
- If all the previous questions are positively answered the next task will be to check if the construction of the new transmission is feasible without exceeding the conventional dimensions, because it still has to fit in a conventional heavy duty truck.
- Finally the costs of the development and production of the new transmission should be kept as low as possible, is this feasible?

Next to the main goal of the thesis there are two additional assignments if the time limit permits to execute them: to design a gear shift mechanism and to perform a cost quantification.

### 1.2 Plan of Action

In order to get a structured method of working first of all the product development process, and specifically the construction process as part of it, is analyzed. A good understanding of the process shows where the critical points are and how they can be solved as smooth as possible.

Note that the product development process will be used as the structure of this thesis and the Process Flow on page number xix can be used as a guideline for reading the thesis.

Next the background information on the conventional transmission and the available technology should be considered.

In order to check the feasibility of the new transmission concept a number of calculations has to be made on the efficiency, power flow and torque handling. If these values come out to be worse than the values of the conventional transmission, it won’t be useful to develop the new transmission.
If the values seem to be alright the shift-strategy needs to be further developed. This shouldn’t be too problematic since the transmission will be an automated manual transmission, meaning that there is a lot of flexibility in the sequence of certain actions.

After the strategy is approved the first basic designs for the transmissions can be made. This process starts at the design of a bearing-layout, which will be the basis for the whole construction.

The different bearing layouts that seem to be suitable for the new transmission will be constructed in order to check the dimensions. During this stage small constructional problems will have to be solved.

Then the best concept will be selected for a more detailed construction including all subsystems. Of course this may be an iterative process. If any subsystem, like for example the retarder or auxiliary drive system cannot function in the current layout, another layout has to be chosen.

Finally the thesis and in specific the definition of the problem will be evaluated which will result in the conclusions and recommendations.
Chapter 2.
Chapter 2. Product Development Process

This chapter describes the development process in general, and the way it is applied to this project. This development process is used as a guideline throughout the project. At first the development process will be discussed in more detail, followed by the discussion of the construction process, which will eventually result in the structure of this master thesis (See page number xix).

2.1 Product Development Process

The product development process is a sequence of steps which covers the complete development of a product, starting at the idea and ending at recycling. The process can often be seen as a Stage-Gate Process as described by Dr. Robert Cooper [5], since every step stands for a stage and has to be evaluated at the gate in order to go to the next stage. The process is illustrated in Figure 2.1.

![Figure 2.1: Part of the Product Development Process.](image)

Only that part of the product development process, which is in scope for this master thesis, is shown in Figure 2.1; the part ranges from the requirements to the selection of concepts. This chapter will discuss the stages of the product development process. [4]

2.1.1 Customer Needs

Identifying the customer needs starts by gathering raw information from the customers which can be done by Interviews, Focus Groups, and observing the product in use. After the information is captured it has to be interpreted in order to remove the useless information. The useful information can then be used to set up a hierarchy of customer needs. The easiest way to document the information is by drawing up a table of three columns. The first column shows the products, the second shows the needs, and the third column shows the importance of that need to the customer.
The customer needs, captured during this master thesis, are provided by the DaimlerChrysler employees, who are in close contact with the customers. A great deal of the information provided, can directly be put in the stage “Target Specs”, the next stage in the product development process.

2.1.2 Target Specifications
The difference between customer needs and target specifications is in the way they are expressed. The customer needs are expressed in the language of the customer while the target specifications what the product has to do are spelled out in precise, measurable detail.

The easiest way to set up a target specifications list is by following the steps below.

1. Prepare a list of metrics or measurable requirements and associate all customer needs with these metrics. A customer need can be associated with multiple metrics.
2. Collect competitive benchmarking information. It is important to know what the specifications of the competition are in order to make sure that the new product will be better than theirs.
3. Set ideal and marginally acceptable target values. During this step there are actually put target values to the list of metrics. These are initial values and probably change along the development process.
4. Reflect on the result and the process which stands for the evaluation of the target values.

2.1.3 Generate Concepts
During this stage the design for the product is made, resulting in a variety of concepts. Along this process there are a few points of interest that should be kept in mind.

1. Clarify the problem by decomposing it into smaller problems and focusing on these sub-problems.
2. Search for information on the problem:
   - Patents
   - Experts
   - Literature
   - Benchmarking
3. Solve the problem.
2.1.4 Select Concepts

The final step of the product development process used in this master thesis is the concept selection. The previous stage may have delivered a variety of concepts for the new product and from these concepts the best one should be used for testing.

The easiest way to do this is by drawing up a selection table as illustrated in Table 2.1. This matrix consists of a list of selection criteria and a list of concepts set out against each other; also, all of the selection criteria are provided with a weight, showing the rate of importance. Next step is to rate every concept using the selection criteria. If all concepts are rated they can be ranked using the rate for every selection criteria in combination with the weight. The best concept will be the one that has the highest total score.

Table 2.1: The concept selection matrix

<table>
<thead>
<tr>
<th>Weight</th>
<th>Concept A</th>
<th>Concept A</th>
<th>Concept A</th>
</tr>
</thead>
<tbody>
<tr>
<td>Rating</td>
<td>Weighted Score</td>
<td>Rating</td>
<td>Weighted Score</td>
</tr>
<tr>
<td>Criteria 1</td>
<td>a</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Criteria 2</td>
<td>b</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Criteria 3</td>
<td>c</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Criteria 4</td>
<td>d</td>
<td></td>
<td></td>
</tr>
<tr>
<td><strong>Total Score</strong></td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

2.2 Construction Process

This section will take a closer look at the third step of the product development process, which is the generation of concepts. During this phase constructions for the transmission are made and the sequence of this process will be written out in this section.

2.2.1 Basic Concept

The construction process of a new product is part of the product development process as described in section 0; in specific, the construction process can be put in “stage n+2” and “stage n+3” of the product development process of Figure 2.1. The construction process starts at the design of a concept and ends in a product designed for mass production. During this master thesis only the major design steps will be performed, meaning the steps from Available Technology to Adjust 2D-construction. The single steps
of the process will be discussed in this section by dividing the complete process in three parts. The complete process is illustrated in Appendix A.

Figure 2.2: Basis

First step in the process will be a literature research in order to get to know the available technology, which will be the starting point for new developments.

Second step is the basic concept design where a new idea is worked out to a schematical representation of the basic design of the new product. This is a global design that only shows the functionalities of the new transmission. During this phase there is still a lot of room for constructional changes.

In the third step all requirements for the new transmission design will be formulated; requirements ranging from vehicle handling to construction boundaries and guidelines.

Fourth step is the determination of the transmission ratios and gear ratios, which will immediately lead to the fifth step, namely the calculation of all rotational speeds in the transmission.
At this point a loop occurs, because if the speeds in the transmission reach extraordinary values the gear ratios need to be adjusted.

In the sixth step the shift strategy will be determined and also here a loop occurs, because if the speed differences between the gear shifts are out of range for the engine, the gear ratios need to be adjusted.

Seventh step: Using the gear ratios of the second step, the amounts of power flowing through the different branches of the transmission can now be calculated. If the amounts of power are known, the efficiency in the transmission can be calculated, which will then be used to compare the new transmission to a conventional transmission. At this point an iterative process arises since the gear ratios have to be adjusted until the optimal transmission efficiency is reached.

Note that the sixth and seventh step can both lead to the rejection of the new concept, because if the speed difference between gear-shifts is too large, or if the new efficiency is worse than the conventional efficiency, the transmission won’t be adding any extra value and will thus be rejected.

The eighth step in the process is the first constructional part of the process and also the most important one, namely the design of the bearing-layout. The design of the bearing-layout will be the basis of the new transmission and influences almost all other parts. In this step a number of designs are made, and by reviewing the bearing-layouts on certain requirements the best solutions will be used for further development.
2.2.2 Generate Concepts

As mentioned before, the eighth step leads to a number of different bearing-layouts for the transmission. This is illustrated in Figure 2.3.

During the ninth step of the construction process the gearwheel dimensions and the gearwheel forces are calculated by Daimler Chrysler. These dimensions determine for a large part the length of the new transmission.

In the tenth step the chosen bearing-layouts of the ninth step will be constructed using the values of step ten. This will lead to a two-dimensional transmission design for every bearing-layout.

But also during this phase an iterative process may exist, because the construction of the two-dimensional transmission can ask for small changes in the bearing-layout and computed values.

2.2.3 Concept Selection

At this point the dimensions of the complete transmissions will become clear and the feasibility of these designs can be evaluated. The best design will, in the eleventh block Concept Selection, be selected on certain criteria and used for further development.
The twelfth step contains the calculation of the mass moments of inertia of the rotating parts in the transmission. The inertias, along with the speeds calculated in the third step, are important because they determine the dimensions of the countershaft-brake and the synchro meshes. Therefore this step will be used to determine the dimensions of the countershaft-brake and the synchro meshes to make sure that they meet up with their requirements.

During the thirteenth step the computed dimensions of the brake and the synchronizers will be applied in the construction of the transmission.

At this point it may be possible that the new dimensions don’t fit the preferred construction of step seven, which is not likely to occur. But if the dimensions cannot be changed, another bearing-layout, with of course another construction has to be chosen.

In addition to the general construction process it is possible to improve certain parts of the transmission that don’t necessarily have to be changed in order for the new concept to function. These improvements can be seen as a construction process on its own, which runs parallel to the main construction process. See Figure 2.4.

The process starts by noticing a part which isn’t constructed as optimally as possible, meaning that it can be improved.

Next, the requirements for the improvements will be analyzed, which will lead to a set of constructional specifications.
These specifications will be used to design a number of concepts from which the most suitable one will be chosen as the improved part for the transmission. This part will then be implemented into the construction at the thirteenth step of the construction process, which is the block Adjust 2D-Construction.

The block Adjust 2D-Construction contains, just like the block Concept Selection a feedback to the Requirements and Specifications in order to evaluate if all requirements are met.

### 2.2.4 Final Steps of the Process

The final steps in the construction process, as illustrated in Figure 2.5, will be a 3D-construction, single part design, building a prototype, and preparing the transmission for mass production.

![Diagram of final steps of the process](image)

**Figure 2.5: Design for mass production**

At first the 3D-construction will mostly be used to check if there are any colliding parts in the transmission. Then from this 3D-construction all single parts will be designed, which can lead to a loop in the process. If namely, a single part is designed, it may affect the 3D-construction because its dimensions are slightly changed.

Of course next, the prototype will be build to provide a lot of information on the transmission in use. This information may include maximum torques, efficiency, noises etc. And finally, preparing the transmission for mass production is a part of the process which takes a great amount of time. It means testing the construction by using one of the prototypes, finding the weak spots, reconstruction of the transmission and then again testing the construction.
Next to that, the production line needs to be prepared for new parts in the transmission, which again can be a construction process on its own. This final section denoted by the white blocks in Figure 2.5 isn’t part of this master thesis, because it simply doesn’t fit the time schedule of only one year.

The remaining of the report will be written according to the construction process of Appendix A. Every step in the process will be analyzed, performed, and of course evaluated in order to make the right decisions.
Chapter 3.
Chapter 3. Available Technology

This chapter provides a short overview on the current state of technologies for transmissions of heavy duty trucks. The first three sections of the chapter will give some general information on transmissions and the last section will describe the characteristics of a conventional 16-Speed Transmission.

3.1 Direct Gear vs. Overdrive

Direct gear transmission means that the final gear of the transmission has a ratio of \( i_{\text{min}} = 1 \) or in other words, in final gear the input torque of the transmission is equal to the output torque of the transmission.

Overdrive means that the ratio of the final gear is smaller than \( i_{\text{min}} = 1 \) and this means that the input torque of the transmission is larger than the output torque of the transmission.

The overdrive-transmission is a transmission that has the same transmission ratio range as the direct gear transmission except for the difference that the ratio range is situated in another area.

The final ratio which determines the vehicle speed is adjusted by the final drive at the axles and is for both applications the same. The main difference is therefore that an overdrive transmission can handle more torque than the direct gear transmission, while the direct gear transmission can reach a better efficiency by mostly driving in direct gear.

The final gear of a transmission is the gear ratio which is driven most of the time by heavy duty trucks.
The overview below shows the comparison of the two gear layouts:

Table 3.1: Comparison of direct-gear and overdrive.

<table>
<thead>
<tr>
<th>Direct Gear Layout</th>
<th>Overdrive Layout</th>
</tr>
</thead>
<tbody>
<tr>
<td>The direct gear needs a final drive with a smaller transmission ratio in order to provide the same vehicle speed as the overdrive transmission.</td>
<td>The overdrive already has a smaller transmission ratio; therefore the final drive can have a larger transmission ratio than in direct gear.</td>
</tr>
<tr>
<td>The torque leaving the transmission in direct gear is equal to the engine torque.</td>
<td>The torque leaving the transmission in overdrive is smaller than the engine torque, resulting in a smaller shaft to the final drive.</td>
</tr>
<tr>
<td>Due to the larger transmission ratios, the transmission can handle less engine torque than the overdrive transmission.</td>
<td>Due to the smaller transmission ratios, the transmission can handle a higher engine torque.</td>
</tr>
<tr>
<td>In direct gear instead of the gearwheels, only the driving- and main shaft are used to transfer the torque, which leads to an improved efficiency.</td>
<td>In overdrive the gearwheels are still used, which doesn’t improve the efficiency.</td>
</tr>
</tbody>
</table>

### 3.2 Progressive vs. Geometrical Gear Steps

If the gear steps take on a geometrical shape it means that the progressions between two adjacent gears are always of the same size. This layout is commonly used for group-transmissions in heavy-duty vehicles, because the split group doubles the gears. Doubling a geometrical layout will provide a better gear pattern than doubling a progressive layout. Another characteristic of the geometrical layout is that the gear steps always cover the same engine speed steps, which means that the vehicle is always able to shift at the same engine speed- and thus power-level.

It is also possible that the gear shifts have a so called progressive shape between the gear ratios, in other words the steps, as well as the engine speed steps, become smaller every time a higher gear is chosen. The result is that the next gear always starts at another engine power level than the previous gear.
The main advantage is the fact that the gear steps become smaller as the transmission ratios are laying closer to each other. A small transmission ratio will have a little overcapacity of power which means that a small gear step is easier to take than a large one. A large transmission ratio will have a lot of overcapacity meaning that it can handle large ratio steps without dropping below the vehicle force hyperbole like in Figure 5.4. Another advantage of progressive gear shifting over geometrical gear shifting is the fact that the shift pattern in higher gears has more equally divided velocity steps. This is interesting if these are the gears that are driven most of the time, like done by heavy duty trucks travelling long distances.

3.3 Synchro Mesh vs. Constant Mesh Transmission

A synchro mesh transmission is a transmission that uses a synchronizer to equalize the speed difference between the gearwheels and shafts that have to be connected. When a gear shift is made, the synchronizer will transform the shift force into a friction force and synchronize the two parts. When the speeds of the rotating parts are more or less the same the connection between the parts can be made.

A constant mesh transmission is a transmission that doesn’t have the synchronizer of the synchromeshes; as a matter of fact it only uses the coupling elements of the synchromesh. Since gearwheels and shafts cannot be connected without being synchronized another method is used. In this situation the synchronizing is performed by the engine, the clutch and a brake located at the countershaft of the transmission. If parts need to slow down the brake is activated and if parts need to accelerate the clutch is shortly closed, possibly in combination with engine acceleration.

3.4 A Conventional 16-Speed Transmission

This section will show an analysis of a conventional 16-Speed transmission, because it is important that the working principles of a transmission are well understood before a new design can be made. Consider Figure 3.1 for a schematical overview of the 16-Speed transmission.
The transmission is called a 3-groups transmission, because it can be subdivided into three groups denoted by the numbers 1, 2 and 3. The first group is the so called split-group, the second group the main-group which provides the gears, and the third group is a planetary gear denoted by the range-group.

The transmission is characterized by its 16 gears that are laid out as follows:
- There are three gears in the main-group that can firstly be doubled by the split-group, and then again doubled by the range-group resulting in twelve gears.
- Direct gear can be achieved by connecting the driving shaft and the main shaft over gear wheel K2. Also here the range-group doubles the gears.
- The last two gears can be achieved by directing the torque over gear wheel K1 and back over gear wheel K2, which can then again be doubled by the range-group.

If the meshes of the transmission are considered, it can be seen that the main-group on itself can be called a constant mesh transmission, because there are no synchronizers used. The synchronizing for this part of the transmission is performed by the engine and a brake. The brake is situated on the countershaft takes care of the decelerations while the engine takes care of the accelerations. The split-group and the range-group on the other hand do have synchronizers and can thus be called synchro mesh transmissions if they are considered on their own.
The layout of the gear ratios is geometrical which means that the ratio step between two adjacent gears is always more or less the same.
The transmission contains also two gear ratios smaller than one which makes it an overdrive transmission.

Finally, something can be said about the dimensions of the gear wheels and other parts. The size of the torque which can be handled by the gear wheels depends on the diameter and the width of the gear wheel. If the torque is increased, then the dimensions of the gear wheels should also be increased by either enlarging the width or the diameter. Although, there are some restrictions for dimension enlargement, namely a contact area which becomes too large has a bad influence on the wear and a diameter enlargement has its limits due to the shaft center distance.

The diameter of the synchronizers in the split-group and the range-group are in combination with the applied force and the friction responsible for the synchronizing time.

The teeth of the sliding collar used in all groups of the transmission need to have sufficient contact surface in order to be able to handle the torque in the transmission. The torque handling can either be improved by enlarging the contact area of every tooth or by enlarging the number of teeth by the same tooth size.

### 3.5 Power-Split Transmission

The terms used to describe the parts of the Power-Split transmission are clarified in Figure 3.2. They will be used throughout this master thesis to describe the construction and design of the Power-Split transmission.
Figure 3.2: Schematical representation of the power-split transmission.

The main layout of the Power-Split Transmission is similar to a conventional 16-Speed Transmission, a transmission system consisting of three groups: the split-group, the main-group and the range-group. In Figure 3.2 the three groups are schematically represented and denoted by the numbers 1, 2 and 3.

This transmission layout can theoretically achieve twenty-one forward gears and five reverse gears; in practice on the other hand only 13 forward gears and 4 reverse gears are used, because these gears provide a nice progressive layout.

How are these gears achieved:
The second group of the transmission contains three forward gears and the reverse gear. This group can then be split into high and low gear by the first group, also called the split group, which leads to six forward gears and two reverse gears.
If none of the gear wheels are used and the driving shaft, main shaft and hollow shaft are coupled, direct gear is achieved. Finally the third group or final drive can be used to introduce an extra reduction before the torque leaves the transmission. This reduction doubles the number of gears resulting in fourteen forward gears and four backward gears.
Figure 3.2 also shows synchro- and constant meshes that connect the gearwheels to the shafts, but there is a specific synchro- or constant mesh that needs some extra attention, which is the second one from the left, also called three-shaft synchro mesh or three-shaft constant mesh depending on the type that is used.

This synchro or constant mesh is able to connect the driving shaft, main shaft and hollow shaft at the same time, and therefore enables the transmission to split the power over the main shaft and the hollow shaft.

In combination with the split group, this again provides an additional number of seven forward gears and one backward gear, which makes the total number of forward gears twenty-one and reverse gears five.

As mentioned before not all of these gears are used for the transmission ratio range, the ones that are actually used can be determined if the single gear ratios are known, which will be investigated in Chapter 5.

3.6 Advantages and Disadvantages Power-Split Transmission

The advantages and disadvantages described in this section are assumed, because the concept is new and there is still no information that can prove them. During this thesis it will become clear whether the assumptions made in this section are correct or not.

3.6.1 Advantages

Why is the development of a power-split transmission so interesting if there already exist 16-speed transmissions that can handle today’s torque?

1. The power-split transmission will deliver a better performance and efficiency in the upper gears that are mainly used in heavy duty traffic.
2. The power-split transmission will have a greater torque and torsion capacity compared to the conventional transmission with similar shaft distance.
3. The power-split transmission will have a direct drive without a rotating countershaft. This improves the efficiency and will save on fuel expenses.
4. The power-split transmission will have a good drivability due to the progressive transmission ratio layout.
5. The power-split transmission will have a high capacity due to the large transmission ratio in first gear, which can be used as a crawler.
3.6.2 Disadvantages

Up till now only the advantages of the power-split concept have been shown, but of course these advantages take along constructional changes, that can lead to disadvantages:

1. If the transmission is shifted into direct gear the counter shaft is disconnected and thus standing still. Since the oil pump is connected to the countershaft it is also standing still and won’t provide the transmission of lubrication. A possible solution is to implement a second oil pump, which leads to extra costs and extra losses.

2. The extra synchromesh which connects the driving shaft, the main shaft and the hollow shaft, enlarges the transmission housing and shaft length. Also the extra dividing wall next to this synchromesh, which is the second bearing holder for the hollow shaft, needs extra space. Every extra length in comparison with the conventional transmission is undesirable since it makes the assembly of the truck more difficult.

3. Due to the additional hollow shaft, used to split the power over two shafts, the number of bearings increases. The bearing layout will change and the design needs to be altered. The question is if the extra sets of bearings can be implemented into the transmission construction.
Chapter 4.
Chapter 4. Requirements

The requirements determine the final design of the new transmission concept and can be split into two parts. The first part is set up by the customers and the other part is set up by the development team.

Finally, all requirements have to be translated into detailed and measurable specifications.

4.1 Customer Requirements

The customer requirements are the result of the customer needs that are translated into target specifications. Therefore the customer needs are the first information that has to be formulated:

- The transmission needs to be able to handle a heavy duty truck including its cargo without having problems driving up a steep hill.
- The truck will be traveling long distances and will mostly be driving in the upper gears, and therefore especially these gears need to be as efficient as possible.
- The transmission has to fit into a conventional heavy duty truck.
- The driver may not notice any difference when shifting gears.
- All functions like an Auxiliary Drive System and a Retarder have to remain in the system.

If these customer needs are converted to measurable and more detailed specifications the customer requirements will be formulated as follows:

- The transmission must be able to handle an engine torque of at least 3000 Nm.
- The efficiency in the upper gears of the new transmission must be higher than the efficiency of the conventional transmission.
- The dimensions and especially the length of the transmission should not exceed the dimensions of the conventional transmission since it would demand the adjustment of truck parts.
- All functionalities of the conventional transmission have to be preserved in one way or the other, preferably in the conventional way.
4.2 **Requirements of the Development Team**

The requirements of the development team are closer to the area of construction:

- Use as many conventional parts as possible. This will lower the costs since the machine park doesn’t have to be adjusted. Also a new part will have to be tested before it can be implemented in a transmission.
- If possible, construct parts that have the same function, identically; like for example the constant meshes for locking the gearwheels of the split-group and the range-group.
- Use the construction principle of light and rigid design.
- The construction may not be over-determined.
- Subsystems should be constructed in such a way that they have as less parts as possible, because this will improve the dynamical behaviour.
Chapter 5.
Chapter 5. Important Values

Before actually starting to design the construction of the transmission there are a number of values that should be known in advance, because these values can have an influence on the dimensions of the transmission.

5.1 The Engine

The Engine which is used as a reference-engine during this master thesis is called the Motor OM 502 LA. This is a standard heavy duty engine which is connected to transmissions comparable to the new power-split transmission.

The engine is an 8 cylinder, 16 liter engine producing a maximum power of 420 kW at 1800 rpm, and a maximum torque of 2700 Nm at 1080 rpm. The engine characteristic is shown in Figure 5.1, where the Torque is illustrated by the blue line and the Power by the red line.

![Engine Characteristic](image)

*Figure 5.1: Engine Characteristic of OM 502 LA*
The Torque and the Power are related to each other by the equation:

\[ P_e = T_e \cdot \omega_e = \frac{2 \cdot T_e \cdot n_e \cdot \pi}{60 \cdot 1000} \]  

In which \( n \) is the speed in rpm.

The maximum speed range of the engine of Figure 5.1 lies between 600 and 2000 rpm, although the range that is mostly driven lays between 800 and 1800 rpm.

Since the new transmission has to be able to handle a maximum torque of 3000 Nm, that value will be used for dimension calculations throughout this project. But in order to check the driving characteristics the engine of Figure 5.1 will be used, since that’s the engine that will be driven at the beginning.

### 5.2 Speed of the Rotating Parts

The speed of the rotating parts is important to know, because the dimensions of for example the countershaft-brake and the synchromeshes are determined by the speed changes they need to cover. Therefore this section will provide an overview of the speed of all relevant parts in the transmission in every gear.

At first the ratios between the gearwheels need to be calculated, and in order to do so the number of teeth of every gearwheel is needed; the gearwheels are located on the driving shaft, counter shaft, hollow shaft, and in the planetary gear. The numbers of teeth are shown in Table 5.1.

Dividing the number of teeth of the secondary wheel by the number of teeth of the primary wheel leads to the gear ratio of a gear wheel pair. The gearwheel ratios can be found in Table 5.1 as well.

The \( i_0 \)-ratio and the gear ratio of the planetary gear have been calculated using the following equations:

\[ i_0 = \frac{n_A - n_B}{n_C - n_B} \text{, or } i_0 = \frac{z_3}{z_2} \cdot \frac{z_2}{z_1} = \frac{z_3}{z_1} \]  

\[ (5.2) \]
In which \( n_A \) stands for the speed of the sun gear, \( n_B \) for the speed of the planet carrier and \( n_C \) for the speed of the ring gear; \( z_1, z_2, \) and \( z_3 \) are respectively the number of teeth of the sun gear, planet and ring gear.

If equation (5.1) is rewritten with \( n_C = 0 \), the gear ratio \( i \) will look as follows:

\[
i = 1 - i_0 \quad (5.3)
\]

**Table 5.1: Number of teeth and gear ratios of the gear wheels and the planetary gear**

<table>
<thead>
<tr>
<th></th>
<th>K1</th>
<th>K2</th>
<th>R3</th>
<th>R2</th>
<th>R1</th>
<th>RR</th>
</tr>
</thead>
<tbody>
<tr>
<td>Primary Shaft</td>
<td>25</td>
<td>35</td>
<td>37</td>
<td>36</td>
<td>44</td>
<td>40</td>
</tr>
<tr>
<td>Secondary Shaft</td>
<td>34</td>
<td>28</td>
<td>23</td>
<td>17</td>
<td>13</td>
<td>13</td>
</tr>
<tr>
<td>Gear Ratio</td>
<td>1.36</td>
<td>0.8</td>
<td>1.609</td>
<td>2.118</td>
<td>3.385</td>
<td>-3.077</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th></th>
<th>Sun Gear ( (z_1) )</th>
<th>Planet ( (z_2) )</th>
<th>Ring Gear ( (z_3) )</th>
</tr>
</thead>
<tbody>
<tr>
<td>Planetary Gear</td>
<td>31</td>
<td>25</td>
<td>84</td>
</tr>
<tr>
<td>Ratio ( (i_0) )</td>
<td>-2.71</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Gear Ratio ( (i) )</td>
<td>3.7</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

Using these ratios the speed of all rotating parts in the transmission can be calculated. The speeds will be normalized to the speed of the driving shaft and the computed values can be found in Appendix B.

The transmission ratios can be computed by dividing the input speed through the output speed, or respectively the speed of the driving shaft by the speed of the planet carrier. These ratios are calculated for fourteen forward gears and five reverse gears which lead to a nice progressive pattern of transmission ratios in Figure 5.

The schematical representation of the gears is shown in Appendix C, where the activated parts are pointed out by the color red. These illustrations should give a good idea on how the power is flowing through the transmission.
Forward Gears:

<table>
<thead>
<tr>
<th>Gear</th>
<th>Ratio</th>
<th>Gear</th>
<th>Ratio</th>
</tr>
</thead>
<tbody>
<tr>
<td>1st</td>
<td>17.03</td>
<td>8th</td>
<td>2.33</td>
</tr>
<tr>
<td>2nd</td>
<td>10.66</td>
<td>9th</td>
<td>1.91</td>
</tr>
<tr>
<td>3rd</td>
<td>8.09</td>
<td>10th</td>
<td>1.66</td>
</tr>
<tr>
<td>4th</td>
<td>6.27</td>
<td>11th</td>
<td>1.43</td>
</tr>
<tr>
<td>5th</td>
<td>4.76</td>
<td>12th</td>
<td>1.19</td>
</tr>
<tr>
<td>6th</td>
<td>3.70</td>
<td>13th</td>
<td>1.00</td>
</tr>
<tr>
<td>7th</td>
<td>2.88</td>
<td>14th</td>
<td>10.43</td>
</tr>
</tbody>
</table>

Reverse Gears:

<table>
<thead>
<tr>
<th>Gear</th>
<th>Ratio</th>
<th>Gear</th>
<th>Ratio</th>
</tr>
</thead>
<tbody>
<tr>
<td>1st</td>
<td>-38.19</td>
<td>4th</td>
<td>-4.18</td>
</tr>
<tr>
<td>2nd</td>
<td>-15.48</td>
<td>5th</td>
<td>-2.46</td>
</tr>
<tr>
<td>3rd</td>
<td>-9.11</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

Figure 5.2: Transmission ratios of the forward and the reverse gears.

Most of the gears are achieved by activating the synchro and constant meshes in a conventional way, except for the following gears:
- The eighth to twelfth gear use the power-split concept, which means that the torque is split over the main shaft and the hollow shaft using the forward gearwheels.
- The direct gear or thirteenth gear is achieved by connecting the driving shaft directly to the main shaft without using the counter shaft.
- The “forward gear” is an additional forward gear realized by the use of the power-split concept and the backward gearwheel. This gear will only be implemented if the application of the transmission asks for it.
- The “backward gear” is an additional reverse gear realized by the use of power-split in combination with the reverse gearwheel.

5.3 Vehicle Speed

The previous sections discussed the engine characteristics and the initially determined gear ratios. If that information is combined with additional vehicle parameters the vehicle speed diagrams of the heavy duty truck can be drawn up.

The assumed values of the vehicle parameters are shown in the table below and are based on the information found on the intranet of DaimlerChrysler.
Table 5.2: Vehicle Parameters

<table>
<thead>
<tr>
<th>Description</th>
<th>Symbol</th>
<th>Unit</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Vehicle Mass (Fully loaded)</td>
<td>$m_V$</td>
<td>[kg]</td>
<td>40000</td>
</tr>
<tr>
<td>Ratio of the Final Drive</td>
<td>$i_{FD}$</td>
<td>[-]</td>
<td>2.846</td>
</tr>
<tr>
<td>Efficiency of the Final Drive</td>
<td>$\eta_{FD}$</td>
<td>[%]</td>
<td>97.0</td>
</tr>
<tr>
<td>Frontal Vehicle Area</td>
<td>$A_V$</td>
<td>[m²]</td>
<td>8.75</td>
</tr>
<tr>
<td>Dynamical Tire Radius</td>
<td>$r_{dyn}$</td>
<td>[m]</td>
<td>0.492</td>
</tr>
<tr>
<td>Air Resistance coefficient</td>
<td>$c_w$</td>
<td>[-]</td>
<td>0.9</td>
</tr>
<tr>
<td>Roll Friction Coefficient</td>
<td>$f_R$</td>
<td>[-]</td>
<td>0.02</td>
</tr>
</tbody>
</table>

For the first diagram that has to be drawn up only the gear ratios of section 5.2 in combination with the final drive are used. They are multiplied by engine speed range of 800 to 1800 rpm, resulting in Figure 5.3.

![Figure 5.3: Shift diagram: Vehicle speed vs. engine speed resulting](image)

This figure shows the exact engine speed at which the transmission should be shifted, in order to speed up the vehicle as smoothly as possible. The green line shows the ideal shift speed for every gear en the red line shows the actual shift speed.
By changing the gear wheel ratios in the Matlab script of Appendix D there has been tried to get the red line as close as possible to the green line, but the gear ratios of section 5.2 are the best fit.

The maximum deviation is pointed out in the figure and has the value of 70 rpm at a speed of 62 km/h, by shifting between the 9th and the 10th gear. In a well automated transmission such a deviation is hardly noticed by the driver.

The next diagram, Figure 5.4 that is drawn up is the vehicle speed vs. vehicle force diagram. This diagram will provide a good indication of the driven gear and maximum vehicle speed under certain conditions.

In order to calculate the forces acting on the vehicle the equations from [1] are used in combination with the vehicle parameters.

The air resistance is calculated by:

\[ F_A = \frac{1}{2} \cdot \rho \cdot c_w \cdot A_v \cdot V^2 \]  
(5.4)

in which \( \rho \) stands for the density of air and \( V \) for the vehicle forward velocity.

The slope resistance:

\[ F_s = m_v \cdot g \cdot \sin(\alpha) \]  
(5.5)

in which \( \alpha \) stands for the slope of the road.

Finally the rolling resistance is calculated by:

\[ F_R = m_v \cdot g \cdot f_r \cdot \cos(\alpha) \]  
(5.6)

The total force acting on the vehicle is calculated by adding equations (5.4), (5.5), and (5.6) and if this resulting force is plotted in a diagram next to the vehicle force, then that will lead to Figure 5.4.
Chapter 5. Important Values

Figure 5.4: Vehicle Force vs. Vehicle Speed

The blue lines in the figure denote the vehicle force in every gear provided by the engine. The red line is fitted along the blue lines and is called the effective vehicle force hyperbole. The ideal vehicle force hyperbole is shown as the green line and is calculated using the following equation:

\[
F_{V, id} = \frac{P_{e, \text{max}}}{V}
\]  \hspace{1cm} (5.7)

In which \(P_{\text{max}}\) stands for the maximum power provided by the engine and \(V\) for the vehicle speed.

From this diagram there can be concluded that the maximum vehicle speed is about 120 km/h at a slope of 0%, which is where the 0 % line crosses the vehicle force in the 13th gear. A slope of 40 % can also be driven, but note that only the 1st gear is sufficient for the job.

The grey areas in Figure 5.4 become smaller as higher gears are driven, which means that the shift comfort is higher in this region.
5.4 Shift Strategy

The shift strategy contains the sequence of actions that has to be taken in order to shift from one gear to another. These actions consist of acceleration and deceleration of the shafts, and the sliding collar displacements.

Before a strategy can be determined there have to be made some assumptions:
- The engine can speed up the driving shaft including every part connected to the driving shaft. The maximum speed difference the engine handle by shift actions cover is set to 500 rpm.
- Since the engine has a mass moment of inertia larger than the inertia of the driving shaft and its connected parts, it can speed up the driving shaft only by closing the clutch while remaining at the same rotational speed.
- The countershaft brake can brake down the counter shaft including every part connected to the countershaft, and it can handle every speed difference of the transmission.
- The sequence of actions can be chosen completely free, because every mechanism has its own actuator.
- Initially the inertia of the rotating parts will be kept out of consideration, because the contribution is only of interest for the synchronizing time, and the time is not yet considered in determining the strategy.

The shift strategy is chosen in such a way that the speed differences between the actions are as small as possible. This leads to the strategy of Appendix E, where Table E.1 and Table E.3 show the single up- and down-shift strategy, and Table E.2 and Table E.4 show the double up- and down-shift strategy.

The tables must be interpreted as follows:
The first column shows between what gears is shifted, the second column shows the sequence, and the third column shows the actions that have to be performed. Next to these columns the speeds of the shafts in the transmission are shown. The displayed speeds are always the resulting speeds when the action of that same row is performed. Note that the time is not taken into account, since the synchronizing and brake times aren’t known. If they are known, the strategy should be reviewed and if necessary improved.

The numbers written next to the driving shaft represent the driven gear at that point of time.
Figure 5.5: Speed of the shafts during single up-shifts

Figure 5.6: Left: Speed of the shafts during “uneven” double up-shifts. Right: Speed of the shafts during “even” double up-shifts.

There are a couple of interesting values that can be pointed out in these figures:

- The maximum speed difference the engine needs to cover lays between gear 7 and 8 pointed out in Figure 5.5. The difference is about $\Delta n = 925$ rpm and is accelerated by the inertia of the engine while closing the clutch.
- The sudden speed difference of the counter shaft between gear 3 and 4, and gear 10 and 11 comes into existence, because shifting the synchromesh of the split group will speed up the shaft after which it has to be slowed down again in order to shift the constant mesh of the main group (Figure 5.5).
- The speed dip between gear 7 and 8 is the result of a brake action, because at that moment the constant mesh of the three shafts needs to be shifted (Figure 5.5).
- The maximum speed difference that has to be slowed down by the counter shaft brake lays between gear 4 and 6 in Figure 5.6 and is $\Delta n = 1024$ rpm. This value will later on be used to determine the dimensions of the counter shaft brake.

*Figure 5.7: Speed of the shafts during single down-shifts*
Chapter 5. Important Values

Also here a couple of interesting values can be extracted from the figures:
- The peak between gear 11 and 10 consists of a driving shaft acceleration since the synchro mesh of the split-group is shifted, and is followed by a deceleration by braking down the countershaft.
- The dip between gear 8 and 10 comes into existence, because the three shafts need to have the same speed in order to be able to shift the constant mesh of the three shafts.
- The largest speed difference of the counter shaft which needs to be slowed down lays between gear 8 and 7 and is about $\Delta n = 336$ rpm.
- The largest speed difference of the driving shaft that needs to be covered by engine acceleration lays in step shift between gear 3 and 1 and is about $\Delta n = 883$ rpm.

The last point isn’t consistent with the assumptions made earlier, where there was mentioned that the maximum speed difference delivered by the engine is $\Delta n = 500$ rpm. But since the first gear can be seen as the crawler, due to the large transmission ratio, this shift action won’t be performed in practice.
Chapter 6.
Chapter 6. Power-Flow through the Transmission

The principle of splitting the power that runs through the transmission is for the first time applied in a heavy duty transmission, which means the principle has to be analyzed. The main question that will be answered in this chapter is whether or not the transmission has an improved efficiency in comparison to the conventional transmission. Since the efficiency depends on the power-flow in the transmission, this has to be analyzed at first.

6.1 Power Flow

The power output of the transmission is equal to the power input of the transmission if the efficiency of the transmission is hundred percent. This means that the value of the power running through the transmission depends only on the power produced by the engine, if no additional losses are taken into account. The power flow is subdivided into four different types of flows: the conventional, the standard power-split, the power-split with backflow over the gearwheels, and the power-split with backflow over the main shaft.

6.1.1 Conventional Power-Flow

In a conventional transmission the power enters at the driving shaft, then flows through the countershaft and the main shaft, and finally exits at the planet-carrier. The flow is illustrated in Figure 6.1, where $P_A$ stands for the input-power, $P$ for the power through the transmission, and $P_B$ for the output-power. The flows denoted by $P_V$ are the power losses at the gearwheels.

![Figure 6.1: Power Flow through a Conventional Transmission](image)
6.1.2 Standard Split Power-Flow

The power-split transmission as described in section 3.5 has the conventional power-flow in the first seven gears and in final gear, but has the split power-flow in gears eight to twelve.

The power flow through the power-split transmission is split at the driving shaft into a power flowing through the gearwheels and a power flowing directly to the planetary gear. This flow is illustrated in Figure 6.2.

![Figure 6.2: Power Flow through the Power-Split Transmission](image)

If the power losses $P_V$ are considered to be zero, then the power input has to be equal to the power output ($P_A = P_B$).

\[
\frac{P_I}{P_A} = \frac{i}{1-i_0} \quad (6.1a)
\]

\[
\frac{P_{II}}{P_A} = 1 - \frac{i}{1-i_0} \quad (6.1b)
\]

In which $i$ stands for the gear ratio and $i_0$ for the transmission ratio of the planetary gear.

The percentages of power flowing directly ($P_I$) and indirectly ($P_{II}$) to the planetary gear are shown in the table below.
Table 6.1: Power-flow through the main shaft ($P_I$), and though the gear wheels ($P_{II}$).

<table>
<thead>
<tr>
<th>Gear</th>
<th>$P_I/P_A$ [%]</th>
<th>$P_{II}/P_A$ [%]</th>
</tr>
</thead>
<tbody>
<tr>
<td>8th</td>
<td>62.8</td>
<td>37.2</td>
</tr>
<tr>
<td>9th</td>
<td>51.5</td>
<td>48.5</td>
</tr>
<tr>
<td>10th</td>
<td>44.7</td>
<td>55.3</td>
</tr>
<tr>
<td>11th</td>
<td>38.5</td>
<td>61.5</td>
</tr>
<tr>
<td>12th</td>
<td>32.1</td>
<td>67.9</td>
</tr>
</tbody>
</table>

In the section above only the forward gears are considered, but there’s of course also an intermediate gearwheel delivering five reverse gears and an additional forward gear. Of these six gears, four gears contain a conventional power-flow and two of them contain a split power-flow. Note that in this case, the split power flows differently through the transmission as with the standard forward gears.

There are two additional flows belonging to these gears, namely one “forward-gear” delivering a positively rotating transmission output and one “backward-gear” delivering a negatively rotating transmission output.

### 6.1.3 Split Power-Flow with Backflow over the Gearwheels

At first the forward-gear will be discussed, which is illustrated in Figure 6.3.

![Figure 6.3: Forward-gear with back-flow over the Gear Wheels](image)

In forward gear the sun gear is rotating relatively fast in forward direction and the ring gear is rotating relatively slow in backward direction, making the planet carrier rotate in forward direction at a relatively low speed.

Due to the fact that the ring gear is rotating slower than the sun gear, the power flows back over the gear wheels and counter shaft to the main shaft. The result is that the back flowing
power is added to the engine power and that there is more power is flowing through the main shaft than needed for the output.

The percentages of split-power relative to the input-power are calculated using equations (6.1a) and (6.1b).

\[
\frac{P_s}{P_d} = \frac{i}{1 - i_0} = \frac{10.43}{1 - (-84/31)} = 2.81 \text{ or } 281\%. \\
\frac{P_{II}}{P_d} = 1 - \frac{i}{1 - i_0} = 1 - \frac{10.43}{1 - (-84/31)} = -1.81 \text{ or } -181\%.
\]

The minus denotes a flow in backward direction and if the percentages are added 100 % will remain as output-power, which of course means \(P_d = P_B\).

Since the power in the transmission increases due to the backflow, there has to be checked if the moments in the parts don’t exceed their maximum values. The maximum values are assumed to be equal to the moments that occur in first gear, because the construction is designed to handle these moments.

The torque that occurs at first gear is shown in Table 6.2, where the rotational speeds are normalized to the engine speed. The input torque delivered by the engine is \(T = 3000 \text{ Nm}\) at a power of \(P = 346 \text{ kW}\) and a speed of \(n = 1100 \text{ rpm}\).

Table 6.2: Moments at the transmission parts in first gear

<table>
<thead>
<tr>
<th>Part in the Transmission</th>
<th>Torque [Nm]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Driving Shaft</td>
<td>3000</td>
</tr>
<tr>
<td>Counter Shaft</td>
<td>3000 * 1.360 = 4080</td>
</tr>
<tr>
<td>Hollow Shaft</td>
<td>3672 * 3.385 = 13811</td>
</tr>
<tr>
<td>Constant- or Synchromesh</td>
<td>13811</td>
</tr>
<tr>
<td>Main Shaft</td>
<td>13811</td>
</tr>
</tbody>
</table>

The power flowing directly to the planetary gear is 281 % of the input-power and thus equal to \(P_s = 2.81 \cdot 346 = 972 \text{ kW}\). Then the torque can be calculated using equation (5.1) at \(n = 1100 \text{ rpm}\) resulting in a torque of \(T = 8430 \text{ Nm}\).

The power flowing backwards through the gear wheels is 182 % of the input power and thus \(P_{II} = 1.81 \cdot 346 = 626 \text{ kW}\). Using equation (5.1) again, but this time at \(n = 1100 \cdot 0.735 = 809 \text{ rpm}\) results in a counter shaft torque of \(T = 7389 \text{ Nm}\)
In this situation the torque through the driving shaft and counter shaft is much higher than the torque in first gear. This means that if this gear will be used to drive the vehicle, the driving shaft and its synchronizer need to be able to handle the higher torque, and the construction has to be improved for better torsion stiffness. But this is not the only part affected by the increased power; also the gearwheels $K_1$ and $K_2$ as illustrated in Figure 3.2 and the countershaft have to be redesigned to handle the extra torque.

### 6.1.4 Split Power-Flow with Backflow over the Main Shaft

The final split power-flow belongs to the so-called backward-gear and will be illustrated in Figure 6.4.

![Figure 6.4: Backward-gear with back-flow over the main- and driving Shaft](image)

The amount of power flowing directly and indirectly through the transmission is respectively

$$\frac{P_I}{P_A} = \frac{i}{1 - i_0} = -\frac{38.19}{1 - \frac{84}{31}} = -10.29 \text{ or } -1029\%.$$  

$$\frac{P_{II}}{P_A} = 1 - \frac{i}{1 - i_0} = 1 - \frac{-38.19}{1 - \frac{84}{31}} = 11.29 \text{ or } 1129\%.$$  

The power through the driving- and main shaft is flowing in backward direction, because the ring gear is rotating faster than the sun gear; this increases the power in the transmission.

Also here the torque in the shafts needs to be calculated and the power flowing backwards through the driving- and main shaft is $P_I = 10.29 \times 346 = 3560$ kW. At a rotational speed of $n = 1100$ rpm this will lead to a torque of $T = 30905$ Nm.
The power flowing indirectly to the planetary gear is $P_{II} = 11.32 \cdot 346 = 3906$ kW. The counter shaft is rotating at $n = 1100 \cdot 1.25 = 1375$ rpm which results in a torque of $T = 27127$ Nm.

The calculations above clarify that if this gear is used to drive the vehicle the torque in the transmission becomes very high. Practically all parts in the transmission need to be redesigned to be able to handle the extra torque. In contrary to the forward gear with backflow this gear increases the torque so much, that the construction will be over-dimensioned for the other gears. Therefore this gear won’t be used for driving the vehicle. Though there is one option to drive the gear, namely if the engine is restricted it won’t be able to deliver a high torque meaning that the torque in the transmission will also remain at a lower level.

### 6.2 Efficiency of the Transmission

Since the power-flows are now known the efficiency of the new transmission concept can be checked. The efficiency of the new transmission concept should be improved in comparison to the efficiency of the conventional transmission, because if the efficiency turns out to be worse the concept won’t be interesting.

The power loss in the transmission is denoted by the efficiency of the power transfer between the parts.

The following assumptions have been made [1]:
- The efficiency of a gear wheel set is $\eta_G = 0.99$.
- The extra losses (e.g. oil) lead to an efficiency of $\eta_L = 0.985$.

The efficiency of the planetary gear depends, of course, on the amount of power that runs through it, but the amount of power that is responsible for the losses is not necessarily equal to the complete power running through the planetary gear. The power can namely be split up into two parts.

One part of the power is transferred by the planetary gear as it rotates as a block, and the other part is transferred by the planetary gear as all gear wheels rotate. The latter is of course responsible for the losses in the planetary gear.
The power can be split up as follows. The rotating part and the block part of the sun gear are respectively:

\[ P_{\text{Roll,1}} = T_1 \left( \omega_1 - \omega_{pc} \right) \]  
\[ P_{\text{Block,1}} = T_1 \cdot \omega_{pc} \]  
\[ \text{(6.2a)} \]
\[ \text{(6.2b)} \]

The rotating part and the block part of the ring gear is respectively:

\[ P_{\text{Roll,3}} = T_3 \left( \omega_3 - \omega_{pc} \right) \]  
\[ P_{\text{Block,3}} = T_3 \cdot \omega_{pc} \]  
\[ \text{(6.3a)} \]
\[ \text{(6.3b)} \]

Since the efficiency is only affected by the power loss in the rotating parts, the transmission is considered when the planet carrier is standing still. In that case all power running through the planetary gear is transferred by rotation, leading to an efficiency that is built up by the gear wheel efficiency and the bearing-efficiency \[ \eta_0 = \eta_G \cdot \eta_B \cdot \eta_G = 0.976 \], where \( \eta_B \) stands for bearing-efficiency.

In order to calculate the efficiency of the planetary gear in a power-split situation, the following equation can be used:

\[ \eta_{PS} = \frac{\left( n_1 / n_3 - i_0 \right) \left( 1 - i_0 \eta_0^w \right)}{\left( n_1 / n_3 - i_0 \eta_0^w \right) \left( 1 - i_0 \right)} \]  
\[ \text{(6.4)} \]

In which \( n_1 \) and \( n_3 \) are the speeds for respectively the sun gear and the internal gear. If \( n_1 > n_3 \) then \( w = +1 \) and if \( n_1 < n_3 \) then \( w = -1 \).

Computing the efficiency of the planetary gear in power-split situation (gears 8 to 12) will give the following results.
Table 6.3: Efficiency in Power-Split Situation

<table>
<thead>
<tr>
<th>Gear</th>
<th>( n_1 ) [rpm]</th>
<th>( n_3 ) [rpm]</th>
<th>( \eta_{\text{Plan.Gear}} ) [-]</th>
<th>( \eta ) [-]</th>
</tr>
</thead>
<tbody>
<tr>
<td>8th</td>
<td>1.00</td>
<td>0.22</td>
<td>0.991</td>
<td>0.969</td>
</tr>
<tr>
<td>9th</td>
<td>1.00</td>
<td>0.35</td>
<td>0.994</td>
<td>0.970</td>
</tr>
<tr>
<td>10th</td>
<td>1.00</td>
<td>0.46</td>
<td>0.996</td>
<td>0.970</td>
</tr>
<tr>
<td>11th</td>
<td>1.00</td>
<td>0.59</td>
<td>0.997</td>
<td>0.970</td>
</tr>
<tr>
<td>12th</td>
<td>1.00</td>
<td>0.78</td>
<td>0.999</td>
<td>0.970</td>
</tr>
</tbody>
</table>

The efficiency of the complete transmission can now be computed by:

\[
\eta = \left( \frac{P_n}{P_A} \cdot \eta_G^n + \frac{P_I}{P_A} \right) \cdot \eta_{\text{PS}} \cdot \eta_L \tag{6.5}
\]

Where \( n \) stands for the number of gear-sets in the transmission which is in this case \( n = 2 \).

The results are shown in the final column of Table 6.3.

It is interesting to calculate if there are efficiency improvements regarding the conventional 16-speed transmission.

The efficiency of the conventional transmission can be calculated by multiplying the gear-set efficiency and the efficiency decreased by the losses. The efficiency of the planetary gear is equal to one, because during these gears it is rotating as a block. The comparison of the power-split and the conventional transmission is shown in Table 6.4 and shows that the new transmission has indeed an improved efficiency.

Table 6.4: Efficiency Comparison

<table>
<thead>
<tr>
<th>Gear</th>
<th>( \eta_{\text{PS}} ) [%]</th>
<th>( \eta_{\text{conv.}} ) [%]</th>
<th>( \Delta \eta ) [%]</th>
</tr>
</thead>
<tbody>
<tr>
<td>8th</td>
<td>96.9</td>
<td>96.5</td>
<td>0.4</td>
</tr>
<tr>
<td>9th</td>
<td>97.0</td>
<td>96.5</td>
<td>0.4</td>
</tr>
<tr>
<td>10th</td>
<td>97.0</td>
<td>96.5</td>
<td>0.5</td>
</tr>
<tr>
<td>11th</td>
<td>97.0</td>
<td>96.5</td>
<td>0.5</td>
</tr>
<tr>
<td>12th</td>
<td>97.0</td>
<td>96.5</td>
<td>0.5</td>
</tr>
</tbody>
</table>

The following standard equations [1] can now be used to calculate the torque proportions in the transmission, when it is shifted into a power-split gear.

If the efficiency is taken along into the calculations as in the equations (6.6a), the transmission efficiencies of Table 6.4 can be checked.
Chapter 6. Power-Flow Through the Transmission

\[
T_A = +1 \\
T_B = -i \cdot \eta \cdot T_A \\
T_S = -T_B \\
T_I = -\frac{T_S}{1 - i_0 \cdot \eta_0} \\
T_3 = -i_0 \cdot \eta_0 \cdot T_I \\
T_4 = -T_3 \\
T_5 = -\frac{1}{i_{45}} \cdot \frac{1}{\eta_G} \cdot T_4 \\
T_6 = -T_5 \\
T_7 = -\frac{1}{i_{67}} \cdot \frac{1}{\eta_G} \cdot T_6 = -(T_A + T_1) \\
\]

(6.6a)

The subscripts of Equations (6.6a), (6.6b) and (6.6c) are illustrated in Figure 6.5

![Figure 6.5: Notations for torque calculations](image)

Finally the power-split situation in combination with the reverse gear activated has to be criticized. These situations are described in the section above by forward-gear and backward-gear, with respectively back-flow over the gear wheels and back-flow over the main- and driving shaft.

In order to do so, the equations (6.6a) have to be rewritten before they can be used to calculate the efficiency and the moments occurring during the forward and the backward gear.

Note that the direction, in which the power is flowing, is of great influence. Namely, if the power is flowing backwards over the gear wheels, or in other words when the forward gear is shifted, the equations have to be rewritten as follows:
If the power is flowing back over the main shaft and driving shaft, or in other words if the backward gear is shifted, the equations are again rewritten as follows:

\[
\begin{align*}
T_A &= +1 \\
T_B &= -i \cdot \eta \cdot T_A \\
T_S &= -T_B \\
T_I &= \frac{-T_S}{1 - i_0 \cdot \eta_0} \\
T_3 &= -i_0 \cdot \eta_0 \cdot T_I
\end{align*}
\]

Since all variables in the equations are known, except for the efficiency \( \eta \), the torque proportions can be calculated. The torque on the seventh gear wheel is calculated in two manners, and can therefore be used to compute the efficiency by solving the two equations.

Both the efficiencies of the transmission, with and without additional losses, are shown in Table 6.5 and the calculations can be found in Appendix F. The additional losses are described by an efficiency of \( \eta_L \).

<table>
<thead>
<tr>
<th>Table 6.5: Efficiency of the transmission with back-flow of the power</th>
</tr>
</thead>
<tbody>
<tr>
<td>Efficiency excl. additional losses.</td>
</tr>
<tr>
<td>Forward-gear</td>
</tr>
<tr>
<td>Backward-gear</td>
</tr>
</tbody>
</table>

### 6.3 Forward-Gear as a Crawler

The previous sections made clear that the forward-gear is achieved by activating the reverse gear in the power-split situation.
The power-flow through the transmission contains a back-flow over the gear wheels raising the power running through the driving shaft and main shaft.

The originally computed gear ratio of the reverse gear wheels, which has a value of $i_{RR} = -40/13 = -3.077$, leads to a forward-gear transmission ratio of $i_{FG} = 10.43$. This transmission ratio is already pretty large, and the idea arose to change the gear ratio in such a way that the forward-gear could be used as a crawler gear; Of course making sure that the reverse transmission ratios would still remain in a useful range.

The concrete goal was to create a new reverse gear ratio in order to achieve a forward gear with a transmission ratio of between $i_{FG} = 20$ and $i_{FG} = 25$. The possibilities are shown in Table 6.6.

<table>
<thead>
<tr>
<th>Table 6.6: Gear ratios and transmission ratios for a crawler</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Primary Gear Wheel</strong></td>
</tr>
<tr>
<td>z_{RR}</td>
</tr>
<tr>
<td><strong>Secondary Gear Wheel</strong></td>
</tr>
<tr>
<td>z_{R}</td>
</tr>
<tr>
<td><strong>Gear Ratio</strong></td>
</tr>
<tr>
<td>i_{RR}</td>
</tr>
<tr>
<td><strong>Transmission Ratio</strong></td>
</tr>
<tr>
<td>i_{FG}</td>
</tr>
</tbody>
</table>

The power-flows through the two branches can again be calculated for every possible transmission ratio of Table 6.6, using equations (6.1a) and (6.1b):
The efficiencies can now be calculated using equations (6.4b):

**Table 6.8: Efficiencies of the forward-gear at different gear ratios**

<table>
<thead>
<tr>
<th></th>
<th>Efficiency excl. additional losses</th>
<th>Efficiency incl. additional losses</th>
</tr>
</thead>
<tbody>
<tr>
<td>Orig.</td>
<td>$\eta = 89.5%$</td>
<td>$\eta = 88.2%$</td>
</tr>
<tr>
<td>1st</td>
<td>$\eta = 79.4%$</td>
<td>$\eta = 78.2%$</td>
</tr>
<tr>
<td>2nd</td>
<td>$\eta = 77.0%$</td>
<td>$\eta = 75.8%$</td>
</tr>
<tr>
<td>3rd</td>
<td>$\eta = 73.9%$</td>
<td>$\eta = 72.8%$</td>
</tr>
<tr>
<td>4th</td>
<td>$\eta = 78.0%$</td>
<td>$\eta = 76.7%$</td>
</tr>
<tr>
<td>5th</td>
<td>$\eta = 75.0%$</td>
<td>$\eta = 73.9%$</td>
</tr>
<tr>
<td>6th</td>
<td>$\eta = 79.1%$</td>
<td>$\eta = 77.9%$</td>
</tr>
</tbody>
</table>

Finally the reverse gears for the new gear ratios are evaluated:

**Table 6.9: Reverse transmission ratios at the different gear ratios of Table 6.6.**

<table>
<thead>
<tr>
<th></th>
<th>Orig.</th>
<th>1st</th>
<th>2nd</th>
<th>3rd</th>
<th>4th</th>
<th>5th</th>
<th>6th</th>
</tr>
</thead>
<tbody>
<tr>
<td>1st</td>
<td>-15.48</td>
<td>-12.27</td>
<td>-11.95</td>
<td>-11.64</td>
<td>-12.08</td>
<td>-11.74</td>
<td>-12.22</td>
</tr>
<tr>
<td>2nd</td>
<td>-9.11</td>
<td>-7.22</td>
<td>-7.03</td>
<td>-6.85</td>
<td>-7.10</td>
<td>-6.91</td>
<td>-7.19</td>
</tr>
<tr>
<td>4th</td>
<td>-2.46</td>
<td>-1.95</td>
<td>-1.9</td>
<td>-1.85</td>
<td>-1.92</td>
<td>-1.87</td>
<td>-1.94</td>
</tr>
</tbody>
</table>

The most suitable gear ratios for a crawler seem to be the second and the fifth option; for these options the minimum vehicle velocity of the reverse gears will be compared to the velocity of the original gear ratio, because the velocity should be about 2.5 km/h.

The equation that will be used to calculate the vehicle speed is:

$$V = \frac{n \cdot 2 \cdot \pi \cdot r_{\text{dyn}}}{i \cdot i_{\text{FD}}} \cdot \frac{60}{1000}$$ (6.7)

This results in the following values.
Table 6.10: Reverse Vehicle Speed

<table>
<thead>
<tr>
<th>Options</th>
<th>Engine Speed [rpm]</th>
<th>Gear Ratio (1st Reverse)</th>
<th>Vehicle Speed (km/h)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Orig. option</td>
<td>600</td>
<td>-15.48</td>
<td>2.53</td>
</tr>
<tr>
<td>2nd option</td>
<td>600</td>
<td>-11.95</td>
<td>3.27</td>
</tr>
<tr>
<td>5th option</td>
<td>600</td>
<td>-11.74</td>
<td>3.33</td>
</tr>
</tbody>
</table>

Conclusion:

The new gear ratios deliver a power-flow between about five and seven times the input power. This leads to enormous torques in the transmission, which would ask for a redesign of the transmission parts in order to be capable of handling the torques.

Another way to reduce the torques in the transmission can be realized by adding a software restriction to the engine, assuring that it doesn’t deliver too much torque. Since the maximum torque of the engine is never needed in this gear it would be a good solution to the problem.

But, the new gear ratios also have a negative influence on the transmission ratios of the reverse gears. By changing the gear ratios the first reverse gear gets a smaller ratio, resulting in a larger reverse velocity of the vehicle.

The critical value which has to be achieved in first reverse gear is 2.5 km/h, which is more or less reached by the original gear ratio. If the gear ratio becomes smaller the vehicle velocity becomes about 3.3 km/h, which is too large for the first reverse gear.

Therefore the overall conclusion will be that an extra crawler using the forward gear won’t be interesting for this transmission.
Chapter 7.
Chapter 7. Bearing Layout

The part of the construction process that has the most influence on the final construction is the bearing layout. Therefore this part will be at the beginning of the actual construction process as can be seen in Appendix A. This chapter analyzes and compares layout concepts that were developed during this project. The best concepts will then be further developed by creating a 2D-construction, but first of all there this chapter will provide some information on bearings in general, because every bearing type has a different purpose.

7.1 Available Bearings

There is a large number of bearings on the market, but only a few of them are useful within a transmission. These bearings are shown in Figure 7.1; a comparison of their characteristics can be found in the overview of Appendix G. This overview provides information which can be useful to choose bearings that have the right functionality for a certain situation in the new transmission. All bearings in Figure 7.1 are so called rolling contact bearings; this means that the friction is reduced by rolling bodies that are kept in place by a ring. Some general advantages of a rolling contact bearing are high usage durability, high reliability, and high economic efficiency.
Besides the choice of the bearing-type it is important to choose the proper bearing-dimensions, and in order to make the right decisions some additional information is required:

- The locations of the bearings.
- The load, the number of revolutions, the temperature, the shaft arrangement and the stiffness of modification parts.
- Requirements like life cycle, accuracy, noise, friction and operation temperature, lubrication and maintenance, and assembly and disassembly.
- Commercial information

For the initial construction of bearing layouts later on in this chapter the bearings will chosen from other transmissions, since the life-cycle calculations are performed afterwards by the DaimlerChrysler calculation division.

The locations of the bearings in the new transmission will be compared to the locations in a conventional transmission, after which the most suitable bearing is chosen according to the list mentioned above.
7.2 Requirements

Every new bearing layout concept has to be compliant with the requirements of this section. These requirements are critical for the functioning of the transmission and therefore need to be followed at all times:

1. All shafts have to be fixed in axial direction, either by bearings or by the gearwheel geometry.
2. All shafts need their fixation in radial direction by no more than two bearings. If more than two bearings are simultaneously in contact, it means that the construction is overdetermined and the shaft cannot move freely which can lead to clamping forces and wear.
3. The length of the shafts has to be as small as possible, because the longer the shafts the larger the bending deformation by the gearwheel forces.
4. The planetary gear must be able to adjust the position of the ring gear, the planet and the sun gear in order to reduce the wear; in other words the gearwheels need to be able to “swim”.
5. The assembly of the transmission must be possible at all times.
6. The functionality of other parts, like for example the countershaft brake, may not be affected.
7. The costs have to be as small as possible, which again means using as many conventional parts as possible.

7.3 Bearing Layout for One Countershaft

Note that in the following concepts all gearwheels connected to the driving shaft and hollow shaft are pivoted by needle bearings. These bearings are needed for the fixation of the gearwheels in loaded and unloaded situation. The needle bearings are not drawn in the figures of this section.

7.3.1 Concept A

The general idea behind this concept is fixing all shafts in longitudinal direction as close to the input-side of the transmission as possible. This way the clamping forces will be applied over a short distance, which reduces the forces in the main and hollow shaft. The sun gear, connected to the end of the main shaft, will then be able to adjust its position relative to the planets, resulting in an equal wear of the gear wheel teeth. Consider Figure 7.2.
Figure 7.2: Concept A - One Countershaft; Small Distance Fixation

- The driving shaft is fixed in longitudinal and radial direction by two taper roller bearings, one of them (2) is fixed to the housing and the other one (3) to the main shaft.
- The main shaft is fixed in longitudinal and radial direction by two taper roller bearings whereby the first bearing is connected to the driving shaft (3) and the second one (4) connected to the hollow shaft.
- Since the main shaft is fixed over such a short distance on the left side of the transmission, the right side is able to give the sun gear some play.
- The hollow shaft is then fixed in longitudinal direction over the bearing connected to the main shaft (4) and a second taper roller bearing (5) fixed to a dividing wall.
- An extra cylinder roller bearing (6) between the hollow shaft and a dividing wall will fix all shafts in radial direction.
- This would mean that the driving shaft is over-determined by the grooved ball bearing (1) at the left side of the figure. Since this bearing is only used as a positioning of the driving shaft it will be given some play, getting rid of the over-determination.
- The planet carrier has to be fixed in radial direction which is done by a grooved ball bearing (8) and a smaller needle bearing (7) attached to the hollow shaft.
- Finally the counter shaft is fixed in longitudinal and radial direction by two taper roller bearings (9, 11). If the countershaft needs an extra bearing to reduce bending by the gearwheel forces, the dividing wall (10) should be a suitable location for an additional cylinder roller bearing. Of course this bearing will also need some play in order to realize a determined construction.
If the location of the taper roller bearing (5) should lead to constructional problems, it is always possible to change its position with the cylinder roller bearing (6) in the other dividing wall. However the disadvantage will be that the shafts will be fixed in longitudinal direction over a larger distance as can be seen in Figure 7.3.

**Figure 7.3: Concept A - One Countershaft; Large Distance Fixation**

### 7.3.2 Concept B

This concept is in many ways the same as the first concept, except for the fact that the taper roller bearing of the main shaft (4) in Figure 7.3 and Figure 7.2 is moved to the position of the next dividing wall (5) as shown in Figure 7.4. This relocation should create more room for the synchro mesh between the driving shaft, main shaft and hollow shaft.
Design of a New Transmission Concept

Figure 7.4: Concept B - One Countershaft

- The driving shaft is fixed in longitudinal direction over bearings (2), (3), (5), and (6).
- In radial direction the driving shaft is fixed by a grooved ball bearing (1) and a taper roller bearing (2). The driving shaft is not over-determined as by the first concept, because this time the main shaft is fixed on the right side of the transmission.
- The planet carrier is just like the first layout fixed by the grooved ball bearing (8) and the needle bearing (7).
- The countershaft is fixed by two taper roller bearings (9, 11) and has the possibility for an extra bearing at the dividing wall (10), just like the first concept.
- Note that the main shaft will still be fixed to the driving shaft and hollow shaft, and not to the housing.

7.3.3 Concept C

The third concept contains a main shaft that isn’t fixed to the hollow shaft, but to the planet carrier. This means that the forces on the hollow shaft have no influence on the main shaft, except of course for the forces applied through the synchro mesh located at taper roller bearing (3) in Figure 7.5.
Chapter 7. Bearing Layout

Figure 7.5: Concept C - One Countershaft

- The driving shaft and main shaft are both fixed in longitudinal and radial direction by the taper roller bearings (2), (3), (7) and (8). But for the radial fixation they also need the grooved ball bearing (1) in order to make the construction determined; otherwise the construction may bend at the location of bearing (3).
- The hollow shaft is completely fixed on its own by two taper roller bearings (4, 5) in the dividing walls.
- The planet carrier is now pivoted by the taper bearing of the driving shaft (7) and a needle bearing at the hollow shaft (6). Since the main shaft is connected to the planet carrier by a taper roller bearing, these two parts are now fixed to each other, which is in conflict with the design requirements. Therefore an additional constructional adjustment has to be made, which will be shown later on in this chapter.
- Finally, the counter shaft is pivoted by two taper roller bearings (9, 11), and the planet carrier is fixed to the housing by taper roller bearing (8).

7.4 Bearing Layout for Two Countershafts

Since the bearing layout concepts for one countershaft aren’t that satisfying, another option has been examined; namely the implementation of a second countershaft, lying exactly opposite to the first countershaft.

An extra countershaft gives the following advantage: If a gearwheel is lying between two oppositely positioned gearwheels, it will be pivoted by these gearwheels, or in other words the gearwheel will be fixed in radial direction.
Two countershafts therefore pivot the intermediate lying gearwheels that are connected to the driving shaft and main shaft. This section will provide a couple of concepts based on this principle.

### 7.4.1 Concept D

![Figure 7.6: Concept D - Two Countershafts](image)

- The driving shaft and the hollow shaft are fixed in radial direction by the gearwheels of the countershafts, and will adjust their position relative to these shafts.
- The grooved ball bearing (1) would have a negative influence on the radial fixation if it was directly connected to the driving shaft; therefore a little play is added between the bearing and the driving shaft.
- The driving shaft, hollow shaft, and main shaft are all fixed in longitudinal direction by the two taper roller bearings (3) and (4), and by the gearwheels. When torque is applied the gearwheels (K1, K2) will push the driving shaft to the right and the gearwheels (3, 2, 1, R) will push the hollow shaft to the left, clamping the main shaft in between.
- Since it is difficult to equalize these gearwheel forces two additional cylinder roller bearings (2) and (5) will secure the position of the shafts.
- Both countershafts are fixed by two taper roller bearings (7) and (8), securing them in radial and longitudinal direction.
- The sun gear will be able to adjust itself relative to the planets since the main shaft is only fixed at the left side.
- The planet carrier is fixed to the housing by two opposing taper roller bearings (6) realizing a large fixation basis by a small mounting place. This is possible since the centrelines of the taper rollers are pointing outwards.

7.4.2 Concept E

The second concept has only small changes compared to the first concept, therefore only those changes will be pointed out.

The two taper roller bearings (3) and (4) of Figure 7.6 are replaced by two cylinder roller bearings (3) and (4), and a needle bearing (5).

- The two cylinder roller bearings will now take over the longitudinal fixation of the taper roller bearings as described in the third point of section 7.4.1.
- The needle bearing (5) will take care of the radial fixation of the main shaft. This has the advantage that the driving shaft and the countershaft are decoupled in radial direction.
- Since there is some play present in the new bearings, the over-determination of the grooved ball bearing (1) has gone. This bearing can now be installed without play.
7.4.3 Concept F

The third concept functions on a different principle as the other two concepts. In the first two concepts the gearwheels are fixed to the driving- and hollow shaft without play, because the shafts are able to swim. This means that if the gearwheels have to adjust themselves relative to the countershafts, the driving shaft and the hollow shaft are adjusted. In the third concept the gearwheels are attached to the driving shaft and hollow shaft with play, which means that they can adjust themselves without influencing the shafts and the shafts can be fixed to the housing. Consider Figure 7.8.

Figure 7.8: Concept F - Two Countershafts

- The gearwheels will push the driving shaft and hollow shaft together if torque is applied. This way the main shaft will be clamped between the bearings (3) and (4), pivoting the three shafts in longitudinal direction.
- The driving shaft and hollow shaft will be fixed in radial position by three grooved ball bearings (1), (2), and (3).
- The main shaft will be fixed in radial direction by a grooved ball bearing (5) that is attached on the left side of the shaft. This means that the right side of the main shaft still has enough play for the sun gear to adjust itself.
Chapter 8.
Chapter 8. 2D-Construction of Bearing Concepts

In the previous chapter six bearing layouts were discussed as an option for the new transmission concept. In this chapter a two dimensional construction of every concept is created in order to check if it is applicable in the new transmission. Every section of this chapter will discuss one two-dimensional construction of the concept which will be done in the same sequence as the previous chapter; larger drawings of the constructions can be found in Appendix H. The final section will select the most suitable concept for further development.

Note that the positions of the bearings in the illustrations are described by the red numbers in the figures. The blue arrows point out interesting areas of the construction.

8.1 One Countershaft

The drawings of this section will clearly show the three shafts: The driving shaft on the left, connected to the main shaft over the taper roller bearing, and the hollow shaft which rotates around the main shaft. There is only one countershaft in the construction which means that all torque will have to be handled by that shaft. The gearwheel dimensions and shaft diameters for all constructions are similar which makes a good comparison possible.
8.1.1 Concept A

- The extra dividing wall denoted by arrow 1 can clearly be seen in the figure, and it also shows that the dividing wall can easily be attached to the housing.
- The transmission is longer than the conventional transmission for heavy duty trucks due to the extra synchromesh or constant mesh denoted by arrow 2.
- Arrow 3 shows the location of the needle bearing that secures the planet carrier relative to the hollow shaft.
- The dividing wall can cause some problems for the assembly of the transmission, since the diameter of the cylinder roller bearing is smaller than the diameter of the gearwheels on the countershaft (Arrow 4).
- Bearings (5) and (6) can easily switch position, changing the construction in a “Small Distance Fixation”.
- All other parts in the transmission remained the same as in a conventional transmission.

8.1.2 Concept B

The second concept with only one countershaft couldn’t be constructed due to the following problem:
If the taper roller bearings (5) as illustrated in Figure 7.4 would be constructed in the hollow shaft it would result in a very wide hollow shaft at that location. The hollow shaft
cannot be made any wider at that location, because the gearwheels and the synchro mesh of the range group are located in the same area.

8.1.3 Concept C

The construction of this concept is quite similar to concept A, except of course for the taper roller bearing (7) that used to be located at the dividing wall denoted by arrow 1 in Figure 8.2.

- The extra dividing wall denoted by arrow 1 can clearly be seen in Figure 8.2. The figure also shows that it can easily be attached to the housing.
- The transmission has become longer than the conventional transmission due to the extra synchro or constant mesh denoted by arrow 2.
- The dividing wall can cause some problems for the assembly of the transmission, since the diameter of the cylinder roller bearing is smaller than the diameter of the gearwheels on the countershaft (Arrow 4).
- Arrow 3 shows the location of the needle bearing that secures the planet carrier relative to the hollow shaft. The radial displacement will normally be handled by the taper roller bearing (8), but since the forces acting on the planet carrier can become pretty large an extra radial bearing can provide some help.
Due to the taper roller bearing (7) the main shaft and the sun gear are completely fixed to the planet carrier, meaning that the sun gear cannot adjust its position relative to the planet carrier. In order to solve this problem there has to be created a solution that delivers additional flexibility in radial direction for the sun gear.

In order to solve this problem there will be taken a closer look at the construction part denoted by arrow 5. As mentioned above the taper roller bearing (7) was initially attached directly to the main shaft, which is illustrated in Figure 8.3.

The first option shows a taper roller bearing that is mounted to the main shaft with play in radial direction; this gives the main shaft the option to move relatively to the taper roller bearing.
Some characteristics are:
- The bearing will be taken along in the rotations of the main shaft by a rubber ring that is mounted between the bearing and the shaft.
- In order to make sure that the bearing cannot fall from the main shaft during assembly, it is secured by a ring.
- A disadvantage occurs at the vertical contact surface between the bearing and the shaft. At that point the material will wear out by intensive adjusting, leading to unwanted play in longitudinal direction.

The second option adds a double cardan joint to the shaft by removing material from the shaft, which can be done by Spark Erosion, because that technique is able to remove very small amounts of material. The material that has to be removed is enlarged in Figure 8.5 and is illustrated out by the white lines between the blue areas.

![Figure 8.5: Double Cardan Joint (90 degrees rotated)](image)

Due to the double cardan the end of the main shaft will be completely flexible in radial direction and still be able to deliver a longitudinal fixation.

The third option shows a cylinder rolling bearing or needle bearing, depending on the value of the longitudinal forces that is fixed at the end of the main shaft. This bearing assures a longitudinal fixation while it still has some play in radial direction; this play is sufficient for the main shaft and sun gear to adjust themselves to each other.

The best solution to the problem seems to be the cardan joint, since this option uses the elastic characteristic of the material and thus rejects all friction from the system. It is also the most expensive solution, but the extra production costs will probably repay themselves since this solution rules out transmission damage.
8.2 Two Countershafts

In case of two countershafts the engine torque is split over these two shafts, resulting in a lower gearwheel load. The dimensions of the gearwheels can therefore be reduced to about seventy percent of the original gearwheels, resulting in a shorter transmission.

8.2.1 Concept D

- The first impression is indeed the fact that this transmission concept is shorter than the previous two transmission concepts, but also that it is wider due to the two countershafts.
- Since the shafts lengths are shorter than the lengths of the conventional transmission additional dividing walls won’t be necessary.
- Arrow 1 denotes the constant mesh. If a synchro mesh would be needed, there should be enough room left for the construction of that synchro mesh.
- The sliding collars denoted by arrow 2 have got an outer gearing. That way they can slide into the gearwheels instead of over them, which results in a simpler and smaller construction with fewer parts.
- All shafts are lying in the same horizontal plane which makes the construction wide and flat.

Figure 8.6: 2D Construction of Concept D
8.2.2 Concept E

This construction is almost completely the same as the previous construction, except for the needle bearing (4). The bearing is placed in the sliding collar, securing the main shaft in radial direction.

The critical point in this construction is the longitudinal translation of the sliding collar. If the main shaft has to keep the same radial position, then the sliding collar has to be fixed to the housing while keeping the longitudinal degree of freedom.
8.2.3 Concept F

Figure 8.8: 2D Construction of Concept F

This construction again shows a couple of differences:
- The difference that will probably be noticed first is denoted by arrow 3. The grooved ball bearing and the needle bearing are replaced by two taper roller bearings. The advantage is that the hollow shaft is not influenced by the forces on the planet carrier, but the disadvantage is the extra length that is needed for the construction.
- The bearings (3), (4) and (5) realize the fixation of the driving shaft, the main shaft and the hollow shaft.
- Another interesting difference is the replacement of the two cylinder roller bearings by two grooved ball bearings (2) and (6).

8.3 Concept Selection

The best concept for the construction of the new transmission will be chosen via a selection matrix similar to the one in section 2.1.4.

All weights (W) are denoted by a number between 1 and 5, where a weight of 1 means unimportant and a weight of 5 very important. A rating (R) of 1 means: very bad, and a value of 5: very good. The weighted rating is then calculated and placed in the column WR.
Table 8.1: Selection matrix for bearing concepts

<table>
<thead>
<tr>
<th>Criteria</th>
<th>Concept A</th>
<th>Concept C</th>
<th>Concept D</th>
<th>Concept E</th>
<th>Concept F</th>
</tr>
</thead>
<tbody>
<tr>
<td>Transmission length</td>
<td>W 5</td>
<td>R 1</td>
<td>WR 5</td>
<td>R 5</td>
<td>WR 25</td>
</tr>
<tr>
<td>Transmission width</td>
<td>R 4</td>
<td>20</td>
<td>R 5</td>
<td>20</td>
<td>4</td>
</tr>
<tr>
<td>Transmission height</td>
<td>R 4</td>
<td>16</td>
<td>WR 4</td>
<td>20</td>
<td>5</td>
</tr>
<tr>
<td>Gearwheel width</td>
<td>WR 5</td>
<td>15</td>
<td>R 3</td>
<td>15</td>
<td>5</td>
</tr>
<tr>
<td>Mass of all gearwheels</td>
<td>WR 4</td>
<td>2</td>
<td>WR 8</td>
<td>2</td>
<td>5</td>
</tr>
<tr>
<td>Gearwheel calculations</td>
<td>WR 2</td>
<td>5</td>
<td>WR 10</td>
<td>5</td>
<td>10</td>
</tr>
<tr>
<td>Mass of all shafts</td>
<td>WR 4</td>
<td>16</td>
<td>WR 4</td>
<td>16</td>
<td>3</td>
</tr>
<tr>
<td>Number of bearings</td>
<td>WR 2</td>
<td>2</td>
<td>WR 1</td>
<td>2</td>
<td>5</td>
</tr>
<tr>
<td>Effects on tolerances</td>
<td>WR 3</td>
<td>12</td>
<td>WR 4</td>
<td>12</td>
<td>3</td>
</tr>
<tr>
<td>Shaft length</td>
<td>WR 5</td>
<td>2</td>
<td>10</td>
<td>2</td>
<td>5</td>
</tr>
<tr>
<td>Effects on countershaft</td>
<td>WR 5</td>
<td>4</td>
<td>WR 20</td>
<td>4</td>
<td>2</td>
</tr>
<tr>
<td>Clamping length of the shafts</td>
<td>WR 3</td>
<td>4</td>
<td>WR 12</td>
<td>3</td>
<td>9</td>
</tr>
<tr>
<td>Extra dividing wall</td>
<td>WR 3</td>
<td>3</td>
<td>WR 9</td>
<td>3</td>
<td>5</td>
</tr>
<tr>
<td>Assembly</td>
<td>WR 5</td>
<td>4</td>
<td>20</td>
<td>4</td>
<td>20</td>
</tr>
<tr>
<td>Flexibility of the sun</td>
<td>WR 5</td>
<td>5</td>
<td>25</td>
<td>3</td>
<td>15</td>
</tr>
<tr>
<td>Total weighted rating</td>
<td>WR 51</td>
<td>209</td>
<td>49</td>
<td>197</td>
<td>65</td>
</tr>
</tbody>
</table>

If the total weighted ratings of Table 8.1 are considered it becomes immediately clear that the concept of two countershafts is a better basis for the new transmission than the concept of only one countershaft.

Now a choice has to be made between the three concepts containing two countershafts. Out of these three concepts, concept F has a slightly higher total weighted rating, than the other two concepts, and that’s why this concept will be used as the basis for further development of the power-split transmission.
Chapter 9.
Chapter 9. Critical Dimensions

A transmission has to be constructed as light and rigid as possible which means that the dimensions of the parts should fit their tasks as precise as possible. This chapter will provide the dimension optimization for the countershaft brake and the synchronizers of the split-group and the range-group.

9.1 Mass Moments of Inertia

The mass moments of inertia in a transmission have a great influence on the synchronizing time, and it also determines which rotating part of the transmission will undergo the largest revolution change during a synchronizing action.

The mass moments of inertia in the gearbox are computed using the CAD program CATIA V4R2.4. At first a simple part has been created in order to validate the results of CATIA. Next, the parts of the transmission have been modelled and the mass moments of inertia have been calculated. The results are shown in Table 9.1 in which the parts of Figure 3.2 are used as description.

Table 9.1: Mass Moments of Inertia

<table>
<thead>
<tr>
<th>Gear Wheels</th>
<th>Symbol</th>
<th>Unit [kgm²]</th>
</tr>
</thead>
<tbody>
<tr>
<td>K1</td>
<td>$J_{K1}$</td>
<td>0.014</td>
</tr>
<tr>
<td>K2</td>
<td>$J_{K2}$</td>
<td>0.021</td>
</tr>
<tr>
<td>R1</td>
<td>$J_{R1}$</td>
<td>0.070</td>
</tr>
<tr>
<td>R2</td>
<td>$J_{R2}$</td>
<td>0.051</td>
</tr>
<tr>
<td>R3</td>
<td>$J_{R3}$</td>
<td>0.021</td>
</tr>
<tr>
<td>RR</td>
<td>$J_{RR}$</td>
<td>0.039</td>
</tr>
<tr>
<td>ZR</td>
<td>$J_{ZR}$</td>
<td>0.006</td>
</tr>
</tbody>
</table>
### Planetary Gear Symbol Unit [kgm²]

<table>
<thead>
<tr>
<th>Planetary Gear</th>
<th>Symbol</th>
<th>Unit [kgm²]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Sun Gear</td>
<td>Jₜₜ</td>
<td>0.0018</td>
</tr>
<tr>
<td>Planet</td>
<td>Jₚ</td>
<td>0.0017</td>
</tr>
<tr>
<td>Ring Gear</td>
<td>Jᵦᵦ</td>
<td>0.19</td>
</tr>
<tr>
<td>Planet Carrier</td>
<td>Jₚₚ</td>
<td>0.082</td>
</tr>
<tr>
<td>Compleet Set</td>
<td>Jₚₜ</td>
<td>0.344</td>
</tr>
</tbody>
</table>

### Other Parts Symbol Iₜₜₜₜₜ [kg m²]

<table>
<thead>
<tr>
<th>Other Parts</th>
<th>Symbol</th>
<th>Iₜₜₜₜₜ [kg m²]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Clutch</td>
<td>Jₜₜ</td>
<td>0.123</td>
</tr>
<tr>
<td>Driving Shaft</td>
<td>Jₜₜₜₜₜₜ</td>
<td>0.035</td>
</tr>
<tr>
<td>Main Shaft</td>
<td>Jₜₜₜₜₜₜ</td>
<td>0.012</td>
</tr>
<tr>
<td>Sliding Collar 1</td>
<td>Jₜₜₜₜₜₜₜ</td>
<td>0.0054</td>
</tr>
<tr>
<td>Sliding Collar 4</td>
<td>Jₜₜₜₜₜₜₜ</td>
<td>0.0094</td>
</tr>
<tr>
<td>Hollow Shaft</td>
<td>Jₜₜₜₜₜₜₜ</td>
<td>0.079</td>
</tr>
<tr>
<td>Counter Shaft</td>
<td>Jₜₜₜₜₜₜₜ</td>
<td>0.044</td>
</tr>
</tbody>
</table>

Sliding collar 3 connects gear wheels 2 and 3 to the main shaft, and sliding collar 4 connects the gear wheels 1 and R to the main shaft.

The constant mesh or synchronmesh bodies are modelled together with the shaft they’re on and the gear wheels on the counter shaft are modelled as part of the countershaft.

### 9.2 Countershaft Brake

The countershaft brake is a lamellae brake fitted on the countershaft of the transmission. The brake is able to brake down that part of the transmission which is connected to the countershaft. The number of parts that will be braked down depends on the synchro meshes or constant meshes that are closed.

The brake reduces the number of synchronmeshes needed in the transmission, because instead of using the synchronizers to equalize the speed of different parts, the countershaft brake will be used.

In order to know exactly which synchronizers can be removed from the transmission and which ones are still required, there has to be calculated what the capacity of the brake is. If the capacity of the brake is sufficient, then the synchronizer can be removed from the transmission.
The following equations [1] can be used to calculate the maximum speed difference that can be handled by the brake in a certain time period.

\[ T_R = \frac{F \cdot j \cdot \mu \cdot d}{2 \cdot \sin(\alpha_R)} \]  \hspace{1cm} (9.1)

\[ \Delta\omega = \frac{t_R \cdot (T_R + T_Y)}{-J_{red}} \]  \hspace{1cm} (9.2)

In which \( F \) stands for the applied force and \( d \) for the working diameter of the brake at the friction coefficient \( \mu \), the number of lamellas is given by \( j \), and \( \alpha_R \) is the angle which the lamellas make with the center line.

In the second equation \( t_R \) is the synchronizing time, \( T_Y \) is the loss, and \( J_{red} \) is the mass moment of inertia reduced to the countershaft. The loss will be assumed equal to zero, so \( T_Y \) will be kept out of consideration.

The working diameter \( d \) of a lamella, which is illustrated in Figure 9.1, can be calculated by dividing the surface of a single lamella into two surfaces of equal size, split by the diameter \( d \).

\[ r_1 = 44.0 \text{ mm}; \ r_2 = 32.5 \text{ mm}; \ r_0 = 22.5 \text{ mm} \]

Using the fact that \( A_1 = A_2 \) leads to:

\[ \pi(r_1^2 - r_d^2) = \pi(r_d^2 - r_2^2) \]  \hspace{1cm} (9.3)
Design of a New Transmission Concept

\[ r_d = \sqrt{\frac{r_1^2 + r_2^2}{2}} \quad (9.4) \]

\[ r_d = \frac{44.0^2 + 32.5^2}{2} \approx 38.5 \]

\[ d = 2 \cdot r_d = 77.0 \text{ mm} \]

9.2.1 Conventional Countershaft Brake

The size of the reduced mass moment of inertia depends on the number of parts that are directly connected to the countershaft. The equation for this inertia reduction will be as follows and is computed using the data of section 9.1.

\[ J_{\text{red}} = J_{\text{CS}} + (J_{K1} + J_{DS} + J_{C}) \left( \frac{Z_4^2}{Z_{K1}^2} \right) + (J_{K2}) \left( \frac{Z_5^2}{Z_{K2}^2} \right) + ... \]

\[ ... (J_{R3}) \left( \frac{Z_6^2}{Z_3^2} \right) + (J_{R3}) \left( \frac{Z_7^2}{Z_2^2} \right) + (J_{R4}) \left( \frac{Z_8^2}{Z_1^2} \right) + (J_{RR}) \left( \frac{Z_9^2}{Z_R^2} \right) \quad (9.5) \]

\[ J_{\text{red}} = 0.4493 \text{ kgm}^2 \]

In order to perform the calculations, the following information by DaimlerChrysler is used:

<table>
<thead>
<tr>
<th>Description</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Air-pressure</td>
<td>( p = 0.85 \text{ N/mm}^2 = 8.5 \text{ bar} )</td>
</tr>
<tr>
<td>Cylinder diameter</td>
<td>( d_{\text{cyl}} = 84.4 \text{ mm} )</td>
</tr>
<tr>
<td>Pressure surface of a single lamella</td>
<td>( A_{\text{lamella}} = 2453 \text{ mm}^2 )</td>
</tr>
<tr>
<td>Number of lamellae</td>
<td>( j = 10 )</td>
</tr>
<tr>
<td>Synchronizing-time</td>
<td>( t_R = 0.3 \text{ s} )</td>
</tr>
<tr>
<td>Cylinder force</td>
<td>( F_{\text{cyl}} = 4476 \text{ N} )</td>
</tr>
<tr>
<td>Angle between the lamella and the centerline</td>
<td>( \alpha = 90^\circ )</td>
</tr>
<tr>
<td>Dynamical friction coefficient</td>
<td>( \mu = 0.13 )</td>
</tr>
</tbody>
</table>

Implementing these values in equations (9.1) and (9.2) leads to a friction moment of \( T_R = 224 \text{ Nm} \) and a speed difference of \( \Delta \omega = -150 \text{s} \), or \( \Delta n = \frac{60 \cdot \Delta \omega}{2 \cdot \pi} \approx -1428 \text{ rpm} \). The negative sign shows that the revolution difference is deceleration and thus a brake action.
Using the information of the section above, the following values can be calculated in order to criticize the brake parameters. At first the total friction surface needs to be calculated, which allows the computation of specific values.

\[ A_R = \sum_{i=1}^{j} A_{R,i} \quad (9.6) \]

Then the specific work of friction can be calculated:

\[ W = \frac{1}{2} \left( -J_{\text{red}} \cdot \Delta \omega^2 - T_v \cdot \Delta \omega \cdot t_R \right) \quad (9.7a) \]

\[ W_A = \frac{|W|}{A_R} \quad (9.7b) \]

There is assumed that the losses are equal to zero \( (T_v = 0) \). The specific work of friction will be \( W_A = 0.205 \) J/mm².

The specific power of friction can now be calculated using:

\[ P_m = \frac{W}{t_R} \quad (9.8a) \]

\[ P_{m,A} = \frac{|P_m|}{A_R} \quad (9.8b) \]

The value will be about \( P_A = 0.683 \) W/mm².

Finally the velocity of friction and the specific friction-surface pressure can be calculated by respectively using the following equations:

\[ v = \Delta \omega \cdot \frac{d}{2} \quad (9.9) \]

\[ p_R = \frac{F}{A_R \cdot \sin(\alpha_R)} \quad (9.10) \]

The velocity of friction will be around \( v = -5.76 \) m/s, and the specific friction-surface pressure will be around \( p_R = 0.183 \) N/mm². As a checkup, all values computed above are compared to literature about heavy duty trucks [1], and the values are of the same range.
The maximum speed difference that needs to be reduced during gear shifts can be found in section 5.4 and Appendix E, and is $\Delta n = 1024$ rpm. If this value is compared to the value of the standard counter shaft brake, it is immediately clear that the standard brake has an overcapacity of about 400 rpm. In the next section will be calculated in what ways the dimensions of the brake can be adjusted in order to get rid of the overcapacity.

### 9.2.2 Capacity Reduction

All calculations made in this section are performed by rearranging equations (7.1) and (7.2) in order to retrieve the desired variables. Next to these calculations, the remaining values are calculated using equations (7.3) to (7.7). This enables a good comparison of the different capacity reductions.

Another point required to make a good comparison is the use of the same parameters in all calculations. These are the parameters of Table 9.2 in the previous section, with additionally a speed difference of $\Delta n = 1024$ rpm.

### Fewer Lamellae

One part that can be reduced is the number of lamellae. Fewer lamellae mean a smaller friction surface and thus less capacity. The minimum number of lamellae is $j = 7.17$, resulting in $j = 8$ because the construction requires an even number of lamellae.

If the remaining values are calculated using $j = 8$, the results are as follows:

- $A_r = 19624$ mm$^2$
- $W_A = 0.164$ J/mm$^2$
- $P_A = 0.546$ W/mm$^2$
- $v = -4.61$ m/s
- $p_r = 0.228$ N/mm$^2$
- $\Delta n = -1143$ rpm

Note that also the new speed difference $\Delta n$ is calculated, because since $j = 8$ leads to a larger speed reduction than $j = 7.17$.

Since the number of lamellae is now reduced by two also the weight of the construction is reduced. The weight reduction is equal to $m_i = \sum_{i=1}^j A_i w_i \rho_l$ in which $A_i$ is the area of the lamella, $w_i$ the thickness of the lamella, and $\rho_l$ the density of the lamella material. The total weight reduction will be $m_i = 128.2$ g.
**Smaller Working Diameter**

A reduction of the working diameter would lead to a reduction of the friction torque and thus of the brake capacity. The reason is that the force is applied on a smaller diameter. Performing the calculations with the working diameter as variable, results in a working diameter of $d = 55.2$ mm, which is slightly bigger than the diameter of the countershaft ($d_0 = 45$ mm). Now, the inner diameter of the lamellae needs to be calculated.

The smallest inner diameter of a lamella can be calculated using the pressure surface $A_R$. Dividing this area into two equal areas, and rearranging the equation for calculating the area of a circular surface gives

$$r_i = \sqrt{\left(\frac{d}{2}\right)^2 - \frac{A_R}{2}} = \sqrt{\frac{(55.2)^2}{2} - \left(\frac{2453}{2\pi}\right)}.$$

The number under the square root is a negative number, meaning that the inner diameter of the pressure surface is smaller than the counter shaft diameter. This way of capacity reduction can therefore not be applied.

**Smaller Applied Force**

The third option to reduce the brake capacity is to reduce the applied force. Again performing the calculations, this time using the force as output, results in a force of $F \approx 3201$ N.

Next the remaining values are calculated:

- $A_R = 24530$ mm$^2$
- $W_A = 0.105$ J/mm$^2$
- $P_A = 0.351$ W/mm$^2$
- $v = -4.13$ m/s
- $p_R = 0.131$ N/mm$^2$

The force can be reduced in two ways:

1. **Decrease the cylinder pressure:** At this moment the pressure is $p = 8$ bar or $p = 0.8$ N/mm$^2$, delivering a force of $F = 4476$ N on an area of $A \approx 5595$ mm$^2$. If the force is reduced to $F = 3201$ N and the cylinder-piston area kept the same the pressure can be reduced to $p = 5.72$ bar.

2. **Use a smaller cylinder-piston area:** If the force is reduced to $F = 3201$ N by a pressure remaining at $p = 0.8$ N/mm$^2$ the cylinder-piston area can be reduced to $A = 4001$ mm$^2$ with a diameter of $d_p = 71.4$ mm. The new diameter is about 15 mm smaller than the conventional diameter.
Conclusion

First of all, if the capacity will be reduced it can not be done using the smaller working diameter; it has to be reduced by a smaller number of lamellae or a smaller applied force. In this case the smaller number of lamellae is more interesting due to the following advantages:

- Since there are fewer lamellae, there are less moving parts resulting in an improved dynamical behavior.
- Reconstruction of the brake is much easier in case of lamellae reduction than in case of applied force reduction. The applied force reduction needs to be realized by a smaller cylinder-piston surface, since the pressure must remain at 8 bar. A smaller cylinder-piston surface leads to the reconstruction of the gearbox housing and will be a more expensive solution compared to removal of three lamellae.

In the end it is the conventional countershaft brake that will be used in the transmission, simply because of the fact that the brake has sufficient capacity and it is less expensive to implement than a new brake which has to be tested again.

The figure below shows how the synchronizing time is related to the speed difference; with on the left side the conventional brake and on the right side the one with lamellae reduction.

![Figure 9.2: Left: standard brake. Right: Reduced lamellae brake.](image)

### 9.3 Dimensions of Synchronizers

The previous sections made clear that synchronizers are only needed at the split-group and the range-group. The dimensions of these synchromeshes depend completely on the synchronizing forces they need to handle due to the inertia of the involved elements.
The synchronizer uses more or less the same equations as the ones used for the countershaft brake, except for the fact that in this case there are two mass moments of inertia involved instead of only one. This means that both elements adjust themselves to each other, instead of just one to the other, like by the countershaft brake. The synchromesh is illustrated in Figure 9.3.

![Figure 9.3: Synchronizer of two masses.](image)

The two masses are rotating at their own speed and when the synchromesh is closed they start adjusting themselves to each other. The mass with the highest rotational velocity will be slowed down by the mass with the lower rotational velocity, and vice versa. The result will be that both masses run at the same rotational velocity, whose value will depend on the mass moments of inertia.

There are a number of parameters that influence the synchronizing time and the revolution difference. These parameters are the same as used for the countershaft brake, only the values are different.

The strategy that leads to the ideal dimensions of a synchronizer is formulated as follows:

1. At first the capacity of synchronizer, which can be expressed as the maximum torque, has to be calculated using equation (9.1).
2. Then the acceleration and deceleration of the masses, when the synchronizer is closed, can be calculated using the following equations.

$$\dot{\omega}_1 = \frac{T_1 - T_R}{J_1} \quad (9.11a)$$

$$\dot{\omega}_2 = \frac{T_R - T_2}{J_2} \quad (9.11b)$$

In which $T_1$ and $T_2$ stand for torque inputs at the masses, and $J_1$ and $J_2$ for the mass moments of inertia. Synchronizing, depending on the type of gear shift, occurs
mostly when the engine and vehicle torque are disconnected which means that $T_1$ and $T_2$ are equal to zero.

3. Now if the initial angular velocity of the masses is known, the angular velocity after a certain time period can be calculated too using:

$$\omega_1 = \omega_{1,0} + \dot{\omega}_1 \cdot t_R$$  \hspace{1cm} (9.12a)$$

$$\omega_2 = \omega_{2,0} + \dot{\omega}_2 \cdot t_R$$  \hspace{1cm} (9.12b)$$

4. The synchronizing time is the time after which there is a certain difference in angular velocity between the two masses. This difference should be about $\Delta n = 50$ rpm or $\Delta \omega = 5.24$ rad/s and makes sure that the teeth fall smoothly into each other. After the sliding collar is shifted to its final position the new number of revolutions will be at $\Delta \omega = 0$ rad/s.

5. Finally the boundary values of the parameters can be adjusted by determining the synchronizing time and rearranging the equations.

### 9.3.1 Synchronizing of the Split-Group

The synchronizer for the split-group is already designed and it is said that the capacity should be sufficient for the power-split transmission. This section will check if this assumption is true and, if necessary, improve the dimensions.

The dimensions and other parameters for the worst case synchronizing scenario are:

<table>
<thead>
<tr>
<th>Description</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Friction coefficient</td>
<td>$\mu = 0.13$</td>
</tr>
<tr>
<td>Angle of the cone with centerline</td>
<td>$\alpha = 6.5^\circ$</td>
</tr>
<tr>
<td>Working diameter</td>
<td>$d = 200$ mm</td>
</tr>
<tr>
<td>Number of cones</td>
<td>$j = 1$</td>
</tr>
<tr>
<td>Input torque</td>
<td>$T_1 = T_2 = 0$ Nm</td>
</tr>
<tr>
<td>Inertia reduced to the driving shaft</td>
<td>$J_1 = 0.2490$ kgm$^2$</td>
</tr>
<tr>
<td>Inertia reduced to the gearwheel $K_2$</td>
<td>$J_2 = 0.1842$ kgm$^2$</td>
</tr>
</tbody>
</table>

$$J_1 = J_c + J_{DS} + J_{HS} + J_{MS}$$  \hspace{1cm} (9.13a)$$

$$J_2 = J_{K2} + \left( J_{CS} + J_{K1} \left( \frac{z_3}{z_1} \right)^2 + J_1 \left( \frac{z_6}{z_3} \right)^2 + J_2 \left( \frac{z_1}{z_2} \right)^2 + J_1 \left( \frac{z_6}{z_1} \right)^2 + J_2 \left( \frac{z_2}{z_3} \right)^2 \right) \left( \frac{z_{K2}}{z_2} \right)^2$$  \hspace{1cm} (9.13b)$$
The applied force depends on the value of the cylinder area and the working pressure. The cylinder area is different on both sides, because on one side the shift-bar is attached which reduces the cylinder area.

<table>
<thead>
<tr>
<th>Shifting</th>
<th>Surface</th>
<th>Pressure</th>
<th>Applied Force</th>
</tr>
</thead>
<tbody>
<tr>
<td>K₁</td>
<td>A₁ = 1809 mm²</td>
<td>p = 0,30 N/mm²</td>
<td>F₁ = 542,7 N</td>
</tr>
<tr>
<td>K₂</td>
<td>A₂ = 1429 mm²</td>
<td>P = 0,55 N/mm²</td>
<td>F₂ = 786,0 N</td>
</tr>
</tbody>
</table>

The most critical shift action for the split-group is the gear shift between the 13th and 12th gear or 13th and 11th gear, where gear wheel K₂ is activated. This means that a force of \( F_2 = 786.0 \) N is applied to the synchronizer. Implementing the values into equation (9.1) leads to a friction moment of about \( T_R = 90.3 \) Nm.

Then combining equations (9.11) and (9.12) leads to the following resulting equation:

At first let’s say:

- \( \Delta \omega_0 = \omega_{1,0} - \omega_{2,0} \), which is the speed difference in rad/s before synchronizing starts and
- \( \Delta \omega = \omega_1 - \omega_2 \), which is the speed difference at the moment of shifting the sliding collar to its final position.

\[
\begin{align*}
\Delta \omega &= \omega_{1,0} + \dot{\omega}_1 \cdot t_R - \omega_{2,0} - \dot{\omega}_2 \cdot t_R \\
&= \frac{\Delta \omega_0 - \Delta \omega}{\dot{\omega}_2 - \dot{\omega}_1}
\end{align*}
\]

(9.14a)

(9.14b)

Computing equation (9.10b) for a speed difference of \( \Delta \omega_0 = 800 \) rpm, leads to a synchronizing time of \( t_R = 0.0884 \) s. After this time both masses are rotating at the same speed, which can be computed using equation (9.12). The value of the resulting speed is about \( n = 494 \) rpm.

It is clear that the synchronizer has sufficient capacity, even over-capacity. So if the maximum allowable synchronizing time is 0.1 s, then the dimensions can be adjusted. The most common solution will be a diameter adjustment, resulting in a material reduction.

The relation between synchronizing time and speed difference is illustrated in Figure 9.4.
9.3.2 Synchromesh of the Range-Group

The synchromesh of the range group brings along an extra difficulty since it synchronizes the internal-gear and the sun-gear of the planetary gear when shifted from the 6th to the 7th gear.

The output of the planetary gear is connected to the final drive, which is connected to the wheels and thus vehicle mass. Therefore it can be assumed that the planet carrier is rotating with a constant speed.

The mass moments of inertia need to be reduced to the internal gear and to the hollow shaft, since these are the parts that will be coupled; this is illustrated in Figure 9.5.

---

Figure 9.4: Split-group synchromesh characteristics

Figure 9.5: Synchronizing of the ring gear and the hollow shaft
The equations for the mass moments of inertia are now formulated as follows:

\[
J_1 = J_{\text{HS}} + J_{\text{MS}} + J_5 + \left( J_{\text{HS}} \left( \frac{z_2}{z_3} \right)^2 + 5 \cdot J_P \left( \frac{z_1}{z_2} \right)^2 \right) \quad (9.15a)
\]

\[
J_2 = J_{\text{RG}} + \left( J_{\text{HS}} + J_{\text{MS}} + J_5 \left( \frac{z_2}{z_1} \right)^2 + 5 \cdot J_P \left( \frac{z_3}{z_2} \right)^2 \right) \quad (9.15b)
\]

The rest of the specifications of the synchromesh are shown in Table 9.4.

**Table 9.4: Specifications of the standard range-group synchromesh**

<table>
<thead>
<tr>
<th>Description</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Friction coefficient</td>
<td>( \mu = 0.13 )</td>
</tr>
<tr>
<td>Angle of the cone with centerline</td>
<td>( \alpha = 6.5^\circ )</td>
</tr>
<tr>
<td>Working diameter</td>
<td>( d = 220 \text{ mm} )</td>
</tr>
<tr>
<td>Number of cones</td>
<td>( j = 1 )</td>
</tr>
<tr>
<td>Input torque</td>
<td>( T_1 = T_2 = 0 \text{ Nm} )</td>
</tr>
<tr>
<td>Inertia reduced to the hollow shaft</td>
<td>( J_1 = 0.1085 \text{ kgm}^2 )</td>
</tr>
<tr>
<td>Inertia reduced to the ring gear</td>
<td>( J_2 = 0.7963 \text{ kgm}^2 )</td>
</tr>
</tbody>
</table>

The applied force depends on the value of the cylinder surface and the working pressure. The cylinder surface is different on both sides, because on one side the shift-bar is attached.

<table>
<thead>
<tr>
<th>Surface</th>
<th>Pressure</th>
<th>Applied Force</th>
</tr>
</thead>
<tbody>
<tr>
<td>Connecting Ring gear to housing</td>
<td>( A_1 = 6362 \text{ mm}^2 )</td>
<td>( p = 0.30 \text{ N/mm}^2 )</td>
</tr>
<tr>
<td>Connecting Ring gear and Hollow shaft</td>
<td>( A_2 = 5982 \text{ mm}^2 )</td>
<td>( P = 0.55 \text{ N/mm}^2 )</td>
</tr>
</tbody>
</table>

Now, the strategy of the previous chapter is followed for the shift action between gear 6 and 7 or gear 5 and 7. During that action the ring gear will be connected to the hollow shaft using an applied force of \( F_1 = 1908.6 \text{ N} \).

The maximum speed difference between the ring gear and the hollow shaft is \( \Delta n = 2000 \text{ rpm} \) and will result in a synchronizing time of \( t_R = 0.0922 \text{ s} \). With, in locked situation, a resulting speed of about \( n = 282 \text{ rpm} \).
Figure 9.6: Range-group synchronizer characteristic
Chapter 10.
Chapter 10. Construction of a Synchro Mesh

The synchro mesh, as it is commonly known, locks a gearwheel to the shaft by firstly equalizing the rotational speed followed by the gearwheel lock. The synchronizing itself is performed by two conical surfaces that are pressed against each other developing a large friction force. The teeth that should lock the gearwheel to the shaft are now in contact, but can only be locked if the speed of both parts is more or less the same; a small difference is always needed in order to let the teeth run smoothly into each other.

10.1 A Synchro Mesh Connecting Three Shafts

The new gearbox concept characterizes itself by the hollow shaft which rotates around the massive main shaft as illustrated in Figure 10.1. In comparison to the conventional transmission, this means that the new transmission has one additional shaft, because the conventional transmission only contains a driving shaft and a main shaft. The gear patterns as illustrated in Appendix C show that the driving shaft, the main shaft, and hollow shaft need to have the ability to be connected to each other and since there may exist a difference in rotation speed that cannot be equalized by Active Synchronizing (AS), a synchro mesh may be desirable.

Figure 10.1: Schematic drawing of the Shaft Synchronizer
The functional capabilities of the synchro mesh require the ability to make the following connections possible:

- The hollow shaft connected to the main shaft
- The driving shaft connected to the main shaft
- The driving shaft and the hollow shaft connected to the main shaft

Another important point which has to be kept in mind while constructing a new synchro mesh is the desirability to use as many conventional parts as possible for the simple reason that conventional parts are already tested, and that the production area is already adjusted for these parts.

With these design requirements in mind there are three concepts developed for the new synchro mesh. The first construction is a conventional Mercedes-Benz Synchronizer used in several truck transmissions of DaimlerChrysler. The second concept is a constructional improvement of the first concept, but isn’t necessarily the better solution. The third concept is a totally different concept and is based on the Borg-Warner Synchronizer.

10.1.1 First Concept

As mentioned above the first concept is based on the conventional Mercedes-Benz Synchronizer with a number of adjustments to make it fit into the new transmission. Consider Figure 10.2.

![First synchronizer concept](image)

*Figure 10.2: First synchronizer concept*
The mechanical characteristics of the concept are as follows:

- The sliding collar is wider than the conventional sliding collar, because in neutral position it has to connect all three shafts.
- The dimensions and design of the synchronizing cone remained unchanged, except for one part. The “fingers” that hold the spring are stretched out to give the spring more freedom.
- The conical designed holder of the spring attached to the synchronizing cone is enlarged in longitudinal direction, because of the fact that in neutral position the spring is already pushed into the holder and it must be possible to push it even further in order to disconnect one of the shafts.
- The synchronizer body is redesigned in order to fit on the main shaft.

10.1.2 Second Concept

While designing the second concept the requirement of using as many conventional parts as possible has been rejected. Instead, the basic principles of construction and design are considered, like light and rigid constructing and reducing the number of parts for better dynamical behavior. Using these principles lead to the following mechanical characteristics which can be seen in Figure 10.3.

Figure 10.3: Second synchronizer concept
- The spring holder and the teeth are built into the shaft in order to reduce the number of parts; thereby ruling out the connection between the holder and the shaft and thus making the construction more rigid. An extra advantage is the play that is reduced to zero and the improved dynamical behavior.
- Due to the smaller diameter of the parts, the mass of the complete synchro mesh is reduced.
- The smaller construction generates more room for the gearwheels fitted next to the synchronizer.
- Since the spring holder is now built into the shaft, there is less room available for the roller taper bearing between the hollow shaft and the main shaft. Therefore there is chosen for a smaller bearing which is possible, because the main shaft isn’t subjected to large bending forces like in a conventional transmission.

The second concept isn’t necessarily better that the first concept, because there are several other points that should be taken in consideration.
- The most important one is the fact that the synchronizer should be able to handle the forces that exist during the synchronizing of the shafts. If the forces are relatively large then of course the bigger synchronizing cone is needed, if not a smaller one should satisfy.
- Another point which lies more in the financial area is the extra costs that have to be made in order to make the parts ready for series production.

10.1.3 Third Concept

The third concept is based on the Borg-Warner Synchronizer and is chosen for its inner synchronizing cone. Since all parts of the synchronizer are constructed beneath the sliding collar it is easier to modify than the Mercedes-Benz Synchronizer. The modifications made to the conventional Borg-Warner Synchronizer can be seen in Figure 10.4.
The sliding collar is made wider to make sure that it connects all three shafts in the neutral position of the synchronizer.

The driving shaft and the hollow shaft have been constructed in such a way that they contain the connection teeth and the conical surface for synchronization. It is of course also possible to construct the teeth and the conical surface as an extra part just like the conventional synchronization uses.

The synchronization is constructed with a diameter as small as possible since that way it leaves more room available for the gear wheels installed next to the synchronization.

10.2 Conclusion

From these three concepts there has to be chosen one concept which will be used in the new transmission if the active synchronizing turns out to be insufficient. In order to make the right decision it’s important to calculate a number of things:

The synchronization forces needed to synchronize the shafts in a certain time period. Depending on the size of these forces a synchronization cone can be constructed. The required diameter of the synchronization cone can eliminate the second concept as a possible solution for the new gearbox.
Initially it is probably desirable to use the present production area, but if the transmission will be taken into production it may be even more desirable to develop a new production area. In mass production the extra advantages like saving material and a more rigid construction can match the extra costs for a new production area, while the better construction enlarges the transmissions lifecycle.

But for now the concepts of this chapter are only to check the feasibility of a synchro mesh between the three shafts. The active synchronizing performed by the engine and the countershaft break should suffice for the connection of the three shafts. May there ever be needed a synchro mesh, then one of the concepts above can be used
Chapter 11.
Chapter 11. Construction of Sliding Collar Teeth

Whether a synchro mesh or a constant mesh is used, the part of the system that makes sure that the involved elements are coupled is the teeth of the sliding collar. By shifting the teeth of the sliding collar into the teeth of the element to be coupled, the connection is secured. But how does this connection stay secured when the clutch is closed again, and the engine torque reapplied.

11.1 Conventional Sliding Collar Teeth

The teeth, as can be seen clearly in Figure 11.1, aren’t cut in a straight way, but under a certain angle alpha of about 6-7 degrees. The reason for the use of such a shape is quite simple if the forces on the teeth are considered.

The torque provided by the engine leads to an applied force by the teeth of the gear wheel and a reaction force of the teeth of the sliding collar. This reaction force can be dissolved in a force perpendicular to the tooth surface and a force parallel to the surface. The force which is parallel to the surface tries to pull the sliding collar into the direction of the gear wheel and thus makes sure that the sliding collar stays in place.
11.2 Power-Split Sliding Collar Teeth

Now the Power-Split concept contains a driving shaft, a hollow shaft and a massive shaft which have to be coupled by the use of synchro mesh or a constant mesh. In the neutral position of the sliding collar all three shafts have to be coupled and therefore the sliding collar needs to be wider than the conventional sliding collar. By the conventional teeth the sliding collar isn’t secured in neutral position, but in the power-split transmission it has to be.

The next section discusses two concepts for the teeth of the new synchromesh or constant mesh. The first concept is based on the conventional geometry and the second concept can be described as the opposite geometry.
Some general information, as can be seen in both Figure 11.2 and Figure 11.3, is the wider sliding collar, which connects all three shafts in the neutral position. The neutral position is always shown in the left side of the figure.

11.2.1 Concept A

The first concept is based on the conventional geometry, but still has a couple of changes.
1. It is easy to see that the teeth of the driving shaft and the hollow shaft are quite similar to the teeth of the gear wheel in Figure 11.1; the only difference is that they are enlarged.
2. Then the sliding collar lost its rectangular middle part and now consists only of two conical shapes under an angle $\alpha$.
3. Finally the geometry of the teeth of the mesh-body in Figure 11.1 had to be adjusted to the shape of the sliding collar; this lead to the geometry of the teeth of the massive shaft in Figure 11.2. The double conical shape in the middle of the teeth is created in such a way that there doesn’t originate material damage in case only two of the three shafts are connected.

With these geometry changes the concept can perform every type of connection needed and of course this connection will be secure. Therefore consider the forces in the figure and it can be seen that the forces between the sliding collar and the outer teeth are always pointed outward. In other words the sliding collar is always pulled into place.
11.2.2 Concept B

Another possible solution is shown in Figure 11.3. This concept is actually the opposite of the first concept shown. In the first concept it is clear that the sliding collar uses its teeth to pull itself into place, just like in the conventional geometry. The idea behind the second concept is to use a force that pushes the sliding collar into place.

The second concept also uses a number of conical faces with an angle alpha, but this time not only pointing inwards.

1. Every tooth contains a conical face pointing inwards and a face pointing outwards.
2. The result is that in neutral position, the left side of the figure, the sliding collar is pushed to the middle by two parallel forces pointing inwards and securing the connection.
3. When the sliding collar decouples one of the shafts, in this case the hollow shaft, then the sliding collar remains secure by a pulling force. If the sliding collar tries to slip back into the neutral position, a pushing force due to the conical surface of the teeth of the massive shaft will push him back into position.
11.3 Concept Selection

If both concepts are considered it seems that concept A looks a bit complicated for its task and Concept B maybe too simple. Therefore there has to be checked which one is most suited for the job.

1. If both sliding collars are in neutral position there is actually no difference in performance. Accept of course for the fact that in the first concept the sliding collar is pulled into place while in the second concept it is pushed into place.

2. If the sliding collar is then transferred to the left there are still no big differences, because they both have to overcome a small hump. But since the hump of concept one is a bit steeper on one side it won’t go as smoothly as by concept two.

3. If the sliding collar has reached its position, there can be seen that the contact surface between the sliding collar and the other teeth is bigger by concept two. Thus there is less chance of material damage. Between the sliding collar and the driving shaft concept one has a larger contact surface.

4. Finally if the sliding collar has to be moved back to the neutral position, in the first concept it needs to overcome a steep hump which is not recommended for smooth shifting.

These criteria are implemented in a selection table which will look as follows. All values are denoted by a number between 1 and 5, where a weight of 1 means unimportant and a weight of 5 very important. A rating of 1 means: very bad, and a value of 5: very good.
Table 11.1: Selection table for coupling teeth.

<table>
<thead>
<tr>
<th>Criteria</th>
<th>Concept A</th>
<th>Concept B</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Weight Factor</td>
<td>Rating</td>
</tr>
<tr>
<td>Force equilibrium in neutral position</td>
<td>5</td>
<td>5</td>
</tr>
<tr>
<td>Contact area in neutral position</td>
<td>5</td>
<td>4</td>
</tr>
<tr>
<td>Force equilibrium in shifted position</td>
<td>5</td>
<td>5</td>
</tr>
<tr>
<td>Contact area in shifted position</td>
<td>5</td>
<td>5</td>
</tr>
<tr>
<td>Sliding to neutral position</td>
<td>4</td>
<td>1</td>
</tr>
<tr>
<td>Sliding to shifted position</td>
<td>4</td>
<td>5</td>
</tr>
<tr>
<td>Material wearing out</td>
<td>5</td>
<td>2</td>
</tr>
<tr>
<td>Manufacturing</td>
<td>3</td>
<td>3</td>
</tr>
<tr>
<td>Manufacturing Costs</td>
<td>3</td>
<td>3</td>
</tr>
</tbody>
</table>

Total weighted rating | 33 | 145 | 41 | 179

The total weighted ratings of the table above show that concept B should be the most suitable construction for this application. Therefore this concept will be described in more detail.

### 11.4 Calculation of the Dimensions

The dimensions of the coupling teeth depend completely on the torque that has to be transferred. The torque delivers a certain surface pressure that may not be larger than the material specific value. In this case a maximum surface pressure of $p_{sp} = 450 \text{ N/mm}^2$ is used and the dimensions are calculated according to DIN 5480.

$$p_{\text{surface}} = \frac{M_{\text{teeth}}}{0.75 \cdot z \cdot h \cdot I \cdot s \cdot r_m}$$  \hspace{1cm} (11.1)
In which 0.75 means that only 75% of the teeth are in contact and thus transfer the torque, \( z \) is the number of teeth, \( h \) the height, \( l \) the length, \( r_m \) the working radius, and \( s \) the part of the tooth surface that makes contact.

The number of teeth is calculated according to the teeth spread of the split-group sliding collar. This sliding collar contains 66 teeth at a working radius of 85 mm, which leads to \( z = 41 \) teeth at a radius of \( r_m = 54 \) mm.

The width of the teeth will be taken equally to the conventional tooth width as used for the sliding collar of the split-group.

The contact surface of the teeth will be 70% or in other words \( s = 0.7 \). The length and the height of the teeth are chosen to be \( h = 5 \) mm, \( l_1 = 6.73 \) mm, \( l_2 = 4 \) mm, \( l_3 = 4.8 \) mm and \( l_4 = 4.5 \) mm. The contact length in neutral position is denoted by \( l_2 \) and in shifted position by \( l_1 \). The different lengths are needed because the torque in shifted position will be much higher.

The maximum torque on the teeth occurs in first gear, where the torque has the value of

\[
M_{\text{teeth}} = T_{\text{engine}} \cdot i_{k1} \cdot i_{R1} = 3000 \cdot 1.36 \cdot 3.385 = 13811 \text{ Nm.}
\]

In this case the sliding collar is shifted and the critical contact length will be \( l_1 \).

If all information is implemented in equation (10.1) the result will be a surface pressure of

\[
p_{\text{surface}} = 353 \text{ N/mm}^2,
\]

which doesn’t exceed the maximum pressure of 450 N/mm².

If the sliding collar remains in the neutral position the critical contact length will be \( l_2 \) at a maximum torque equal to the engine torque \( T_{\text{engine}} = 3000 \text{ Nm.} \)
If this information is implemented in equation (10.1) the surface pressure will be \( p_{\text{surface}} = 129 \text{ N/mm}^2 \), also smaller than the maximum pressure of 450 N/mm\(^2\).
Chapter 12.
Chapter 12. Construction of the Oil Supply

The oil supply of the transmission protects the rotating parts against rapid wear of the material surface. The main parts that are subjected to wear are the gearwheels and the bearings.

12.1 Introduction

In a conventional transmission the oil pump is driven by the countershaft, which is illustrated in Figure 12.1.

Figure 12.1: Conventional oil pump
The oil pump is fixed in the housing denoted by arrow 5. The oil pump consists of an outer ring (arrow 4) and an inner ring (arrow 3) that rotate relative to each other. The oil pump is driven by the auxiliary drive shaft (arrow 2) which is connected to the countershaft denoted by arrow 1.

The total oil supply in the transmission can be subdivided in three parts:
1. The oil pump pumps the oil through the driving shaft, main shaft and hollow shaft in order to lubricate the gearwheels and bearings from the inside out.
2. The oil pump pumps the oil through a rod with small holes mounted above the gearwheels. As the oil runs through the holes of the rod it lubricates the gearwheels from the outside.
3. The third part of the oil supply is provided by the gearwheels themselves. Since the gearwheels are partly lying in the oil, they splash the oil around creating a mist of oil in the transmission.

Note that the oil supply only functions if the countershaft rotates, because both the oil pump and the splashing gearwheels are connected to the countershaft.

12.2 Oil Supply for the New Transmission Concept

The new transmission concept is based on a final direct gear. The advantage of this gear is, that the countershaft including the gearwheels is not rotating, which results in an improved efficiency. However if the countershaft is standing still the oil pump won’t be driven and the oil supply is cut off.

One general adjustment of the oil supply can be made in contrary to the conventional oil supply. Since the auxiliary drive system as constructed in the conventional transmission does not per se have to be constructed in the same way in the new transmission; the oil pump can thus be relocated. The oil pump can be moved to the front end of the second countershaft as shown in Figure 12.2; a couple of concepts will use this option as a basis for their design.
Figure 12.2: Oil pump relocated on the second countershaft

But, the transmission will still need lubrication of the bearings in direct gear which means that the construction needs to be adjusted even more and the way this can be done is discussed in the following sections.

12.2.1 Additional Electric Oil Pump

An additional electric oil pump would be turned on if the direct gear is shifted, and could be very small since it only needs to lubricate the bearings.

The advantage of an electric oil pump is that it can easily be driven by a battery and can easily be mounted outside the transmission.

The disadvantage of such a system is that there is an extra part included to the transmission for a situation in which only one part should satisfy. It would of course also mean that an extra circular flow from and to the electric oil pump is needed.
12.2.2 Additional Mechanical Oil Pump

Another option is an additional mechanical oil pump mounted on the driving shaft. This small pump will always be running if the driving shaft rotates, which means that during the first twelve gears it will run next to the countershaft oil pump and in final gear it will run on its own. The additional oil pump will have to pump the oil directly into the driving shaft and main shaft in order to lubricate the bearings.

The construction of the additional oil pump is illustrated in Figure 12.3. De main oil pump on the countershaft is denoted by arrow 1.

![Figure 12.3: Additional mechanical oil pump on the driving shaft](image)

The next logical step would be to try if the large oil pump on the countershaft fits on the driving shaft. Unfortunately this doesn’t fit, because if the clutch has to be opened it needs a certain offset (arrow 2) that has to stay free at all time.

12.2.3 Single Oil Pump Located Below the Driving Shaft

The oil pump is relocated in the housing below the driving shaft and will be driven by a gearwheel connected to the driving shaft.

The fact that the oil pump is driven by the driving shaft means that the oil pump works in every gear, whether the countershaft is rotating or not. The construction of this concept is shown in Figure 12.4.

![Figure 12.4: Single oil pump located below the driving shaft](image)
An extra advantage of this construction is that the ratio of the drive can easily be adjusted by changing the gearwheel of either the oil pump or the driving shaft. If the oil pump is located at the countershaft the rotation speed depends purely on the gear ratios of the split group, if located below the driving shaft the oil pump will always rotate at the same ratio.

The gearwheel that provides the oil pump drive can be a very small gearwheel, since the torque needed for the oil pump is very small. Therefore the extra room needed on the driving shaft doesn’t interfere with the offset (arrow 1) needed for the clutch.

A disadvantage of the concept could be the following. All shafts in the transmission are built in a horizontal plane, which can make the transmission relatively thin. By relocating the oil pump below the driving shaft, this might cost extra room.

### 12.2.4 Double One-way Clutch

The main idea behind this concept is to drive the oil pump by the countershaft and by the planet carrier depending on their rotational speeds. If the countershaft is rotating faster than the planet carrier the countershaft has to drive the oil pump and vice versa. This means that the system will need two one-way clutches where the possible locations are illustrated in Figure 12.5.
Both one-way clutches will have there free wheel running in the same direction; the countershaft will be driving the oil pump directly and the planet carrier will be driving the oil pump via an additional gear ratio.

Analysis of the rotational directions and speed of the countershaft and the planet carrier:
- If the countershaft is rotating faster than the planet carrier the countershaft will drive the oil pump, because the second one-way clutch will relatively rotate in the free rotating direction.
- If the planet carrier is rotating faster than the countershaft the opposite situation occurs and the planet carrier will drive the oil pump.
- In reverse gear the planet carrier, as well as the second one-way clutch will rotate in the opposite direction, and the countershaft will drive the oil pump.

**Figure 12.5: Locations of the one-way clutch**

Locations:
- The first one-way clutch is located between the countershaft and the oil pump driving shaft (arrow 1). This one-way clutch needs to have sufficient torque capacity, because it is also used in the auxiliary drive system.
- A second option for this one-way clutch is the location denoted by arrow 3, that way the countershaft can still drive the oil pump, but in this case with a smaller one-way clutch since it isn’t part of the auxiliary drive system.
- The second one-way clutch will be located on the oil pump driving shaft and is denoted by arrow 2. This one-way clutch will be driven by a gearwheel mounted on the planet carrier (arrow 4).
The main disadvantage of this concept is the fact that the location of the gearwheel on the planet carrier is exactly the same as the location for the retarder gearwheel. Would this system be constructed, then it would have a great impact on the length of the transmission.

12.3 Concept Selection

The most suitable concept for the situation will in this case be the third concept: A single oil pump located below the driving shaft.

The main advantage is the fact that there is no additional oil pump needed, which is better for the efficiency of the transmission. An additional oil pump would namely decrease the advantage of the efficiency improvement in the direct gear.

The oil pump is connected to the driving shaft by a fixed gearwheel ratio that can be chosen for an optimal functioning of the oil pump.

As the driving shaft speeds up the oil pump will provide a larger flow which is also needed due to the higher loading on the rotating parts.
Chapter 13.
Chapter 13. Final Design

This chapter will combine the results of the previous chapter in a single new transmission construction, where the two-dimensional construction of the selected bearing-layout from section 8.3 will be taken as a basis for the new transmission. Then the dimensions of numerous parts will be adjusted for an optimal performance, and the parts that are redesigned will be added to the transmission.

Every section in this chapter will discuss a part of the new transmission design that needs some extra attention. The final section will provide a simplified three dimensional construction of the transmission to get a better idea on the dimensions.

13.1 The 2D-Construction of the Bearing Layout

The basis of the new transmission construction is the two-dimensional construction of the selected bearing layout from section 8.3. This layout is again illustrated in Figure 13.1.

Figure 13.1: 2D-Transmission construction of the selected bearing layout
- Note that the housing of the transmission is not optimized during this project, and is only adjusted in order to get the most compact design.
- The overall image of the transmission clearly shows that the two countershafts are lying in the same horizontal plane as the driving shaft, main shaft and hollow shaft. All shafts are fixed to the housing without being over-determined and the gearwheels on the driving shaft and hollow shaft can swim relatively to fixed gearwheels of the countershaft.
- The distance between the shaft centers of the main shaft and countershaft is set at 152 mm, which is equal to the conventional 16-speed transmission.

13.2 Split- and Range-Group Synchro Meshes

The synchro mesh of the split group is copied from the conventional transmission and remained unchanged. But if there is take a closer look at the left side of Figure 13.2 it seems that the diameter of the synchro mesh can be reduced. The constructional requirements say that there should be as many parts as possible that are copied one to one from the conventional transmission to the new transmission. Therefore the synchro mesh of the split-group is not adjusted.

Figure 13.2: Left: Synchromesh of the split-group. Right: Reversed synchro mesh of the range group
The synchro mesh of the range-group on the other hand is adjusted, but the parts remained unchanged in comparison to the conventional transmission. The adjusted in this case is the relocation of the synchromesh.

The conventional transmission had a synchro mesh located at the right side (arrow 1) of the planet carrier, but the new transmission needs a synchro mesh that can be coupled to the hollow shaft. Therefore the synchro mesh had to be relocated at the left side (arrow 2) of the transmission.

### 13.3 Three Shaft Constant Mesh

After evaluating the shift strategy of section 5.4 there is chosen for a constant mesh that connects the driving shaft, main shaft and hollow shaft. The automated shifting which uses the engine and the countershaft brake to speed up and slow down the parts should be sufficient for smoothly connecting the parts. The constant mesh is illustrated in Figure 13.3.

![Figure 13.3: Left: The construction of the constant mesh. Right: The top view of the teeth used in the construction.](image)

The construction of the constant mesh is in fact quite simple; The mesh body is connected to the main shaft and the sliding collar connects the main shaft to one or both of the other shafts. The sliding collar is attached to the mesh body and can slide to both sides until it hits the stop unit (arrow 1).

Chapter 11 discussed the need to improve the geometry of the sliding collar teeth that are connecting the three shafts. The teeth that are selected at the end of the chapter are implemented into the construction of the constant mesh.
13.4 Constant Gearwheel Meshes

The design of the constant meshes is different for the new transmission in comparison to the conventional transmission. The conventional transmission uses constant meshes that lock the gearwheels with the same teeth as the ones that are used to connect them to the shaft. The constant meshes of the new transmission on the other hand, contain extra teeth on top of the sliding collar which are used to lock the gearwheels (Arrow 1 in Figure 13.4).

The reason for using this construction is the fact that the sliding collars can now run into the gearwheels, which spares some room in longitudinal direction. Another advantage is the fact that the gearwheels can be constructed using less material since they are hollow now.

In order to illustrate the difference between the design with two constant meshes of different diameters, and the design of two constant meshes with equal diameters, both designs are constructed in Figure 13.4. Above the center-line of the shafts the design of different diameters is shown, and below the center-line the design of equal diameters.

![Figure 13.4: Two designs for the constant gearwheel meshes](image)
Design with different diameters:

- Arrow 2 denotes the different diameters of the constant meshes, which has the advantage that the gearwheels can be placed against the massive material of the hollow shaft creating a good longitudinal fixation.
- The disadvantage is the two sliding collars having different sizes and characteristics. This means that both parts need to be tested and produced using different tools, which again results in additional costs.

Design with equal diameters:

- Arrow 3 denotes the long gearing that fixes both constant meshes to the hollow shaft. Since there is only one type of sliding collar used, the test and production costs are significantly reduced.
- The gearwheels are floating on the long gearing and are fixed by retaining rings, just like EATON and ZF Friedrichshafen use in their transmissions. The rings are actually small ring gears that are slided over the long gearing until they reach a groove. At that point they are rotated by one tooth and fixed by a pen that is slided through the gearing in longitudinal direction. The gearwheels will have a relatively large mount of play in longitudinal and radial direction, but this shouldn’t affect the wear.
- The long gearing can be manufactured in one action, this in contrast to the design with different diameters which needs at least two actions.

### 13.5 Countershaft Brake

The construction of the new transmission in Figure 13.1 is provided with the conventional countershaft brake for the simple reason that this brake is already applicable in series production, and that it has more than sufficient capacity.

The countershaft with improved dimensions of Chapter 9 is also constructed in the final transmission design and the differences between the two brakes are illustrated in Figure 13.5.
- The conventional number of lamellae is reduced by two, which is illustrated by the blue colour.
- The lamellae are fitted to the housing and to the countershaft and since there are fewer lamellae the dimensions of both parts can be adjusted. The countershaft in the right part of the figure is shorter than the one in the left part, and also the housing is smaller in the right part of the figure.

13.6 The Retarder Gearwheel

The gearwheel that drives the retarder is fixed to the planet carrier as shown in Figure 13.6. The gearwheel puts some extra length in the transmission which makes it of approximately the same length as the conventional 16-speed transmission.
13.7 Three-Dimensional Illustration

This section will provide an illustration of the transmission in three-dimensional space in order to give a better idea of the dimensions.

The different colours in the illustrations denote the following parts:

- Light grey: Driving shaft, Hollow shaft
- Dark grey: Main shaft, Planet carrier
- Light blue: Gearwheels of the main-group, Retarder
- Dark blue: Gearwheels of the split-group
- Light green: Countershaft including its gearwheels
- Dark green: Ring gear, Synchro mesh of the range-group
Figure 13.7: Side, Top and ISO view of the Rotating parts in the transmission
Chapter 15.
Conclusions and Recommendations

Conclusions

The main question asked in the introduction is whether the new transmission concept is feasible or not from a construction perspective. This question was subdivided in a number of questions that are evaluated here:

- The selected transmission ratios are perfect for long haul traffic. The vehicle-speed/force diagram shows that the vehicle can handle various slopes and that it has a top speed above the maximum allowed speed. The vehicle-speed/engine-speed diagram shows that the transmission has a progressive shift-pattern which means that the engine can pick up the speed differences in upper gears more easily.
- The shift strategy including the shift patterns has been determined in Chapter 5 and is enabled by the automated shift mechanism.
- The efficiency of the new transmission will be the same as the conventional efficiency from the first gear to the seventh gear, because the working principle of the transmission remains the same. From the eighth gear to the twelfth gear the efficiency is improved, because the power is split over the sun gear and the ring gear of the planetary gear. Finally the efficiency of the thirteenth or direct gear is well improved, because the countershaft, including the gearwheels, is not rotating.
- The transmission is constructed to be able to handle an engine torque of 3000 Nm, which is more than the conventional transmission.
- The length of the new transmission only exceeds the length of the conventional transmission by a couple of millimeters which won’t be a problem. The width of the transmission on the other hand has become wider, but this is compensated by the height of the transmission which is reduced. The truck has sufficient room for the new transmission which means that the dimensions of the new transmission can be maintained.
- The final question is related to the costs of the new transmission. The costs of the transmission will be higher than expected, but since the transmission is planned to be the new standard transmission in future, these costs should be repaid.
It can be concluded that it is definitely interesting to use this transmission for further development. Some small constructional problems may occur, but it should be no problem to solve them; the concept itself is feasible and possibly for the future transmission for heavy duty trucks.

The additional assignments, namely the design of a shift mechanism and the cost quantification are not executed, because the there wasn’t enough time left. Therefore these assignments have been included to the recommendations paragraph.

**Recommendations**

The key recommendation from the analysis is that Concept 6 as identified in Chapter 8 should be the starting point for further construction and prototyping of the Power-Split transmission system.

As already mentioned in the conclusion there are still two assignments that should be executed: to design a gear shift mechanism and to perform a cost quantification; this indeed should be done in the further development process.

In Chapter 9 the synchronizing time for the countershaft brake and the synchro meshes is calculated, including the new speeds after the synchronization. It is recommended to use these time values for the shift strategy determined in section 5.4, because it can change the strategy. Then compare the old and new strategy and determine if these changes affect the construction.

It may be possible that a synchro mesh for connecting the driving shaft, main shaft and hollow shaft is needed. In that case the constructions of Chapter 10 can be used as the basis for a new synchro mash.

This thesis checked the feasibility of a concept where the distance between the shaft centres is set to 152 mm; it may be interesting to check the feasibility of a smaller shaft centre distance of 142 mm as well.

Some parts in the transmission can still be improved. The thesis already demonstrated that the countershaft brake and the synchromeshes are over-dimensioned, which makes the transmission heavier than perhaps necessary.
Several subsystems contain too many parts which increases the play and have therefore a negative influence on the dynamical behaviour in the transmission, for example the brake system and the synchro mesh. The recommendation is an optimization action on these systems.

It would be interesting to investigate the possibilities of RFID technology (Radio Frequency Identification), which is a relatively new technology. It may not be interesting at this moment, but in future it definitely will. Think about the assembly of trucks where they say that no truck is the same; if every part is identified the possibility of failures will be reduced enormously because of improved logistical management. Also coupling RFID to sensors in the transmission could provide the driver with additional information on the truck that could help him to identify a problem before it causes major damage to the truck.
Appendices.
Appendix A. Construction Process of a Transmission
# Appendix B. Speed of Rotating Parts

<table>
<thead>
<tr>
<th>Name</th>
<th>RPM</th>
<th>Diameter</th>
<th>Length</th>
<th>Weight</th>
<th>Material</th>
<th>Speed</th>
</tr>
</thead>
<tbody>
<tr>
<td>Item 1</td>
<td>1500</td>
<td>0.5</td>
<td>2</td>
<td>100</td>
<td>Steel</td>
<td>50</td>
</tr>
<tr>
<td>Item 2</td>
<td>2000</td>
<td>1</td>
<td>3</td>
<td>200</td>
<td>Aluminum</td>
<td>60</td>
</tr>
<tr>
<td>Item 3</td>
<td>2500</td>
<td>1.5</td>
<td>4</td>
<td>300</td>
<td>Copper</td>
<td>70</td>
</tr>
<tr>
<td>Item 4</td>
<td>3000</td>
<td>2</td>
<td>5</td>
<td>400</td>
<td>Stainless Steel</td>
<td>80</td>
</tr>
</tbody>
</table>

Note: RPM stands for revolutions per minute.
Appendix C. Activated Parts in Every Gear

First Gear

Second Gear

Third Gear

Fourth Gear

Fifth Gear

Sixth Gear

Seventh Gear

Eighth Gear

Ninth Gear

Tenth Gear
Appendix D. Matlab Script for Vehicle Speed Diagrams

clear all; close all;

% parameters
iFD   = 2.846;                  % Gear ratio final drive
iFD1  = iFD*0.97;               % Final drive with efficiency (estimated value)
iPG   = 3.7;                    % Gear ratio planetary gear (n_ring gear = 0)
i0    = -84/31;                 % Gear ratio planetary gear (n_planet carrier = 0)
rdyn  = 0.492;                 % Dynamical tyre radius
Zp    = [25 35 37 36 44 40];   % Gear wheels on primary shaft
Zs    = [34 28 23 17 13 13];    % Gear wheels on secondary shaft

% calculation of the gear ratios
i = [Zs(1)*Zp(5)*iPG/(Zp(1)*Zs(5))
     Zs(1)*Zp(4)*iPG/(Zp(1)*Zs(4))
     Zs(1)*Zp(3)*iPG/(Zp(1)*Zs(3))
     Zs(2)*Zp(4)*iPG/(Zp(2)*Zs(4))
     Zs(2)*Zp(3)*iPG/(Zp(2)*Zs(3))
     iPG
     Zs(1)*Zp(4)/(Zp(1)*Zs(4))
     1/((i0*(1*(Zp(1)/Zs(1))*(Zs(5)/Zp(5)))-1)/(i0-1))
     1/((i0*(1*(Zp(1)/Zs(1))*(Zs(4)/Zp(4)))-1)/(i0-1))
     1/((i0*(1*(Zp(2)/Zs(2))*(Zs(4)/Zp(4)))-1)/(i0-1))
     1/((i0*(1*(Zp(2)/Zs(2))*(Zs(3)/Zp(3)))-1)/(i0-1))
     1]

% Implementation of the efficiency
i1 = [i(1)*(0.99^2)*0.9825                      % Plan.Gear eta = (1-i0*eta0)/(1-i0)
i(2)*(0.99^2)*0.9825                           % with: iO = -2.71 and eta0 = 0.976
     i(3)*(0.99^2)*0.9825
     i(4)*(0.99^2)*0.9825
     i(5)*(0.99^2)*0.9825
     i(6)*0.9825
     i(7)*(0.99^2)
     i(8)*0.969                                    % from here on efficiencies of chapter 4
     i(9)*0.97
     i(10)*0.97
     i(11)*0.97
     i(12)*0.97
     i(13)*1];

% calculation of the speed in every gear
n = 800:1:1800;                % Engine speed
Vmax = [];% maximum speed
for k=1:13
    v = n^2*pi*rdyn*60/(i(k)*iFD*1000); % Maximum speed of the gear
    vmax = v(length(n));
end
Vmax = [Vmax vmax]; hold on; grid;
plot(v,n)
xlabel('Vehicle Speed [km/h]'); ylabel('Engine Speed [rpm]');
title('Gears');
end

pause

% Vertical lines
n1=[]; nfit=[]; Vfit=[];
for m=1:12
  for n=800:1800
    if Vmax(m) < n*2*pi*rdyn*60/(i(m+1)*iFD*1000)
      n1 = [n1 n];
      % Return the engine speed, where Vmax of gear m is equal to the
      % speed of gear m+1.
      % This engine speed will be used to draw the vertical line
    end
  end
  % Fit a line through the section points of the vertical lines,
  % and the speed lines
  nfit = [nfit n1(1)];
  vfit = nfit(m)*2*pi*rdyn*60/(i(m+1)*iFD*1000);
  Vfit = [Vfit vfit];
  plot(Vmax(m),n1,'b',Vfit,nfit,'r')
  n1=[];
end
pause

% Vehicle Force, Vehicle Speed diagram
n = [600 650 700 750 800 850 900 950 1000 1050 1100 1150 1200 1250 1300 1350 1400 1450 1500
  1550 1600 1650 1700 1750 1800 1850 1900 1950 2000];
T = [1470 1660 1840 2000 2150 2300 2450 2580 2660 2699 2700 2690 2680 2670 2660 2640 2610
  2570 2540 2490 2450 2400 2350 2285 2230 2150 2070 1980 1900];
alpha = (pi/180)*90*[0 0.1 0.2 0.3 0.4];  % Slope angles [rad]

% Vehicle parameters
g = 9.81;                                    % Gravity [m/s^2]
m_v = 40000;                                % Vehicle mass [kg]
rho_l = 1.2;                                % Air density [kg/m^2]
cw = 0.9;                                    % Vehicle Cw coefficient
A_v = 3.5*2.5;                               % Frontal vehicle area [m^2]
fr = 0.02;                                   % Roll Friction value [-]
Fv_Ae = []; V_Ae = [];
for k = 1:13
  V = n*2*pi*rdyn*60/(i(k)*iFD*1000);    % Vehicle speed [km/h]
  F = T*i1(k)*iFD1 /(rdyn*1000);           % Vehicle force [kN]
  figure(2)
  hold on;
  plot(V,F); grid;
xlabel('Vehicle Speed [km/h]'); ylabel('Vehicle Force [kN]')
  Fv_Ae = [Fv_Ae F(24 )];                   % Forces of force hyperbole [kN]
  V_Ae = [V_Ae V(24) ];                   % Speeds of force hyperbole [km/h]
end
for s = 1:5                               % 5 slope-angles
    v = 0:0.1:130/3.6;                    % Vehicle speed range
    F_A = .5*rho_l*cw*A_v*(v).^2;        % Air resistance [N]
    F_S = m_v*g*sin(alpha(s));           % Slope resistance [N]
    F_R = fr*m_v*g*cos(alpha(s));        % Roll resistance [N]
    Floss = (F_A+F_S+F_R)/1000;          % Total resistance [kN]
    plot(v*3.6,Floss,'k-');              % Total resistance [kN]
end
end
pause

speed = 6:120;                                % Vehicle speed [km/h]
Fv_Aid = 420./(speed/3.6);                   % Ideal Vehicle force hyperbole [kN]
plot(speed,Fv_Aid,'g',V_Ae,Fv_Ae,'r');
## Appendix E.  Shift Strategy

### Table E.1: Single Up-Shifts

<table>
<thead>
<tr>
<th>1-&gt;2</th>
<th>DS</th>
<th>CS</th>
<th>HS</th>
<th>MS</th>
<th>PC</th>
</tr>
</thead>
<tbody>
<tr>
<td>1. Initial cond.</td>
<td>2000,000</td>
<td>1470,588</td>
<td>434,492</td>
<td>434,492</td>
<td>117,430</td>
</tr>
<tr>
<td>1-&gt;2</td>
<td>DS</td>
<td>CS</td>
<td>HS</td>
<td>MS</td>
<td>PC</td>
</tr>
<tr>
<td>1. Initial cond.</td>
<td>2000,000</td>
<td>1470,588</td>
<td>434,492</td>
<td>434,492</td>
<td>117,430</td>
</tr>
<tr>
<td>2. SM4 L-&gt;N</td>
<td>2000,000</td>
<td>1470,588</td>
<td>434,492</td>
<td>434,492</td>
<td>117,430</td>
</tr>
<tr>
<td>3. CS Braking</td>
<td>1252,000</td>
<td>920,588</td>
<td>434,492</td>
<td>434,492</td>
<td>117,430</td>
</tr>
<tr>
<td>4. SM3 N-&gt;R</td>
<td>1252,000</td>
<td>920,588</td>
<td>434,722</td>
<td>434,722</td>
<td>117,492</td>
</tr>
<tr>
<td>2-&gt;3</td>
<td>DS</td>
<td>CS</td>
<td>HS</td>
<td>MS</td>
<td>PC</td>
</tr>
<tr>
<td>1. Initial cond.</td>
<td>2000,000</td>
<td>1470,588</td>
<td>694,444</td>
<td>694,444</td>
<td>187,688</td>
</tr>
<tr>
<td>2. SM3 R-&gt;N</td>
<td>2000,000</td>
<td>1470,588</td>
<td>694,444</td>
<td>694,444</td>
<td>187,688</td>
</tr>
<tr>
<td>3. CS Braking</td>
<td>1518,560</td>
<td>1116,588</td>
<td>694,444</td>
<td>694,444</td>
<td>187,688</td>
</tr>
<tr>
<td>4. SM3 N-&gt;L</td>
<td>1518,560</td>
<td>1116,588</td>
<td>694,095</td>
<td>694,095</td>
<td>187,593</td>
</tr>
<tr>
<td>3-&gt;4</td>
<td>DS</td>
<td>CS</td>
<td>HS</td>
<td>MS</td>
<td>PC</td>
</tr>
<tr>
<td>1. Initial cond.</td>
<td>2000,000</td>
<td>1470,588</td>
<td>914,149</td>
<td>914,149</td>
<td>247,067</td>
</tr>
<tr>
<td>2. SM3 L-&gt;N</td>
<td>2000,000</td>
<td>1470,588</td>
<td>914,149</td>
<td>914,149</td>
<td>247,067</td>
</tr>
<tr>
<td>3. SM1 L-&gt;R</td>
<td>2000,000</td>
<td>2500,000</td>
<td>914,149</td>
<td>914,149</td>
<td>247,067</td>
</tr>
<tr>
<td>4. CS Braking</td>
<td>1548,800</td>
<td>1936,000</td>
<td>914,149</td>
<td>914,149</td>
<td>247,067</td>
</tr>
<tr>
<td>5. SM3 N-&gt;R</td>
<td>1548,677</td>
<td>1935,846</td>
<td>914,149</td>
<td>914,149</td>
<td>247,067</td>
</tr>
<tr>
<td>4-&gt;5</td>
<td>DS</td>
<td>CS</td>
<td>HS</td>
<td>MS</td>
<td>PC</td>
</tr>
<tr>
<td>1. Initial cond.</td>
<td>2000,000</td>
<td>2500,000</td>
<td>1180,556</td>
<td>1180,556</td>
<td>319,096</td>
</tr>
<tr>
<td>2. SM3 R-&gt;N</td>
<td>2000,000</td>
<td>2500,000</td>
<td>1180,556</td>
<td>1180,556</td>
<td>319,096</td>
</tr>
<tr>
<td>3. CS Braking</td>
<td>1520,000</td>
<td>1900,000</td>
<td>1180,556</td>
<td>1180,556</td>
<td>319,096</td>
</tr>
<tr>
<td>4. SM3 N-&gt;L</td>
<td>1519,324</td>
<td>1899,155</td>
<td>1180,556</td>
<td>1180,556</td>
<td>319,096</td>
</tr>
<tr>
<td>5-&gt;6</td>
<td>DS</td>
<td>CS</td>
<td>HS</td>
<td>MS</td>
<td>PC</td>
</tr>
<tr>
<td>1. Initial cond.</td>
<td>2000,000</td>
<td>2500,000</td>
<td>1554,054</td>
<td>1554,054</td>
<td>420,015</td>
</tr>
<tr>
<td>2. SM3 L-&gt;N</td>
<td>2000,000</td>
<td>2500,000</td>
<td>1554,054</td>
<td>1554,054</td>
<td>420,015</td>
</tr>
<tr>
<td>3. CS Braking</td>
<td>1553,600</td>
<td>1942,000</td>
<td>1554,054</td>
<td>1554,054</td>
<td>420,015</td>
</tr>
<tr>
<td>4. SM2 R-&gt;N</td>
<td>1554,054</td>
<td>1942,000</td>
<td>1554,054</td>
<td>1554,054</td>
<td>420,015</td>
</tr>
<tr>
<td>5. SM1 R-&gt;L</td>
<td>1554,054</td>
<td>1142,687</td>
<td>1554,054</td>
<td>1554,054</td>
<td>420,015</td>
</tr>
</tbody>
</table>
### 6->7

<table>
<thead>
<tr>
<th></th>
<th>DS</th>
<th>CS</th>
<th>HS</th>
<th>MS</th>
<th>PC</th>
</tr>
</thead>
<tbody>
<tr>
<td>1.</td>
<td>Initial cond.</td>
<td>2000,000</td>
<td>1470,588</td>
<td>2000,000</td>
<td>2000,000</td>
</tr>
<tr>
<td>2.</td>
<td>SM2 N-&gt;R</td>
<td>2000,000</td>
<td>1470,588</td>
<td>2000,000</td>
<td>2000,000</td>
</tr>
<tr>
<td>3.</td>
<td>SM5 R-&gt;L</td>
<td>2000,000</td>
<td>1470,588</td>
<td>540,541</td>
<td>540,541</td>
</tr>
<tr>
<td>4.</td>
<td>CS Braking</td>
<td>1556,640</td>
<td>1144,588</td>
<td>540,541</td>
<td>540,541</td>
</tr>
<tr>
<td>5.</td>
<td>SM3 N-&gt;R</td>
<td>1556,757</td>
<td>1144,674</td>
<td>540,541</td>
<td>540,541</td>
</tr>
</tbody>
</table>

### 7->8

<table>
<thead>
<tr>
<th></th>
<th>DS</th>
<th>CS</th>
<th>HS</th>
<th>MS</th>
<th>PC</th>
</tr>
</thead>
<tbody>
<tr>
<td>1.</td>
<td>Initial cond.</td>
<td>2000,000</td>
<td>1470,588</td>
<td>694,444</td>
<td>694,444</td>
</tr>
<tr>
<td>2.</td>
<td>SM3 R-&gt;N</td>
<td>2000,000</td>
<td>1470,588</td>
<td>694,444</td>
<td>694,444</td>
</tr>
<tr>
<td>3.</td>
<td>CS Braking</td>
<td>694,400</td>
<td>510,588</td>
<td>694,444</td>
<td>694,444</td>
</tr>
<tr>
<td>4.</td>
<td>SM2 R-&gt;L</td>
<td>694,400</td>
<td>510,588</td>
<td>694,444</td>
<td>694,444</td>
</tr>
<tr>
<td>5.</td>
<td>CS Accel.</td>
<td>1619,744</td>
<td>1190,988</td>
<td>351,741</td>
<td>1619,744</td>
</tr>
<tr>
<td>6.</td>
<td>SM4 N-&gt;L</td>
<td>1619,090</td>
<td>1190,508</td>
<td>351,741</td>
<td>1619,090</td>
</tr>
</tbody>
</table>

### 8->9

<table>
<thead>
<tr>
<th></th>
<th>DS</th>
<th>CS</th>
<th>HS</th>
<th>MS</th>
<th>PC</th>
</tr>
</thead>
<tbody>
<tr>
<td>1.</td>
<td>Initial cond.</td>
<td>2000,000</td>
<td>1470,588</td>
<td>434,492</td>
<td>2000,000</td>
</tr>
<tr>
<td>2.</td>
<td>SM4 L-&gt;N</td>
<td>2000,000</td>
<td>1470,588</td>
<td>434,492</td>
<td>2000,000</td>
</tr>
<tr>
<td>3.</td>
<td>CS Braking</td>
<td>1637,744</td>
<td>1204,223</td>
<td>568,661</td>
<td>1637,744</td>
</tr>
<tr>
<td>4.</td>
<td>SM3 N-&gt;R</td>
<td>1637,744</td>
<td>1204,223</td>
<td>568,661</td>
<td>1637,744</td>
</tr>
</tbody>
</table>

### 9->10

<table>
<thead>
<tr>
<th></th>
<th>DS</th>
<th>CS</th>
<th>HS</th>
<th>MS</th>
<th>PC</th>
</tr>
</thead>
<tbody>
<tr>
<td>1.</td>
<td>Initial cond.</td>
<td>2000,000</td>
<td>1470,588</td>
<td>694,444</td>
<td>2000,000</td>
</tr>
<tr>
<td>2.</td>
<td>SM3 R-&gt;N</td>
<td>2000,000</td>
<td>1470,588</td>
<td>694,444</td>
<td>2000,000</td>
</tr>
<tr>
<td>3.</td>
<td>CS Braking</td>
<td>1734,478</td>
<td>1275,351</td>
<td>792,786</td>
<td>1734,478</td>
</tr>
<tr>
<td>4.</td>
<td>SM3 N-&gt;L</td>
<td>1734,478</td>
<td>1275,351</td>
<td>792,786</td>
<td>1734,478</td>
</tr>
</tbody>
</table>

### 10->11

<table>
<thead>
<tr>
<th></th>
<th>DS</th>
<th>CS</th>
<th>HS</th>
<th>MS</th>
<th>PC</th>
</tr>
</thead>
<tbody>
<tr>
<td>1.</td>
<td>Initial cond.</td>
<td>2000,000</td>
<td>1470,588</td>
<td>914,149</td>
<td>2000,000</td>
</tr>
<tr>
<td>2.</td>
<td>SM3 L-&gt;N</td>
<td>2000,000</td>
<td>1470,588</td>
<td>914,149</td>
<td>2000,000</td>
</tr>
<tr>
<td>3.</td>
<td>SM1 L-&gt;R</td>
<td>2000,000</td>
<td>2500,000</td>
<td>914,149</td>
<td>2000,000</td>
</tr>
<tr>
<td>4.</td>
<td>CS Braking</td>
<td>1722,681</td>
<td>2153,351</td>
<td>1016,860</td>
<td>1722,681</td>
</tr>
<tr>
<td>5.</td>
<td>SM3 N-&gt;R</td>
<td>1722,681</td>
<td>2153,351</td>
<td>1016,860</td>
<td>1722,681</td>
</tr>
</tbody>
</table>

### 11->12

<table>
<thead>
<tr>
<th></th>
<th>DS</th>
<th>CS</th>
<th>HS</th>
<th>MS</th>
<th>PC</th>
</tr>
</thead>
<tbody>
<tr>
<td>1.</td>
<td>Initial cond.</td>
<td>2000,000</td>
<td>2500,000</td>
<td>1180,556</td>
<td>2000,000</td>
</tr>
<tr>
<td>2.</td>
<td>SM3 R-&gt;N</td>
<td>2000,000</td>
<td>2500,000</td>
<td>1180,556</td>
<td>2000,000</td>
</tr>
<tr>
<td>3.</td>
<td>CS Braking</td>
<td>1674,482</td>
<td>2093,103</td>
<td>1301,118</td>
<td>1674,482</td>
</tr>
<tr>
<td>4.</td>
<td>SM3 N-&gt;R</td>
<td>1674,482</td>
<td>2093,103</td>
<td>1301,118</td>
<td>1674,482</td>
</tr>
<tr>
<td></td>
<td>DS</td>
<td>CS</td>
<td>HS</td>
<td>MS</td>
<td>PC</td>
</tr>
<tr>
<td>---</td>
<td>----------</td>
<td>----------</td>
<td>----------</td>
<td>----------</td>
<td>----------</td>
</tr>
<tr>
<td>1</td>
<td>Initial cond.</td>
<td>2000,000</td>
<td>2500,000</td>
<td>1554,054</td>
<td>2000,000</td>
</tr>
<tr>
<td>2</td>
<td>SM3 L-&gt;N</td>
<td>2000,000</td>
<td>2500,000</td>
<td>1554,054</td>
<td>2000,000</td>
</tr>
<tr>
<td>3</td>
<td>CS Braking</td>
<td>1674,580</td>
<td>2093,225</td>
<td>1674,580</td>
<td>1674,580</td>
</tr>
<tr>
<td>4</td>
<td>SM2 L-&gt;N</td>
<td>1674,580</td>
<td>2093,225</td>
<td>1674,580</td>
<td>1674,580</td>
</tr>
<tr>
<td>5</td>
<td>SM1 R-&gt;N</td>
<td>1674,580</td>
<td>0,000</td>
<td>1674,580</td>
<td>1674,580</td>
</tr>
<tr>
<td>Table E.2: Double Up-Shifts</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>-----------------------------</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td><strong>1-&gt;3</strong></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>1. Initial cond.</td>
<td>2000,000</td>
<td>1470,588</td>
<td>434,492</td>
<td>434,492</td>
<td>117,430</td>
</tr>
<tr>
<td>2. SM4 L-&gt;N</td>
<td>2000,000</td>
<td>1470,588</td>
<td>434,492</td>
<td>434,492</td>
<td>117,430</td>
</tr>
<tr>
<td>3. CS Braking</td>
<td>950,593</td>
<td>698,965</td>
<td>434,492</td>
<td>434,492</td>
<td>117,430</td>
</tr>
<tr>
<td>4. SM3 N-&gt;L</td>
<td>950,593</td>
<td>698,965</td>
<td>434,492</td>
<td>434,492</td>
<td>117,430</td>
</tr>
<tr>
<td><strong>2-&gt;4</strong></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>1. Initial cond.</td>
<td>2000,000</td>
<td>1470,588</td>
<td>694,444</td>
<td>694,444</td>
<td>187,688</td>
</tr>
<tr>
<td>2. SM1 L-&gt;R</td>
<td>1176,471</td>
<td>1470,588</td>
<td>694,444</td>
<td>694,444</td>
<td>187,688</td>
</tr>
<tr>
<td><strong>3-&gt;5</strong></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>1. Initial cond.</td>
<td>2000,000</td>
<td>1470,588</td>
<td>914,149</td>
<td>914,149</td>
<td>247,067</td>
</tr>
<tr>
<td>2. SM1 L-&gt;R</td>
<td>1176,471</td>
<td>1470,588</td>
<td>914,149</td>
<td>914,149</td>
<td>247,067</td>
</tr>
<tr>
<td><strong>4-&gt;6</strong></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>1. Initial cond.</td>
<td>2000,000</td>
<td>2500,000</td>
<td>1180,556</td>
<td>1180,556</td>
<td>319,069</td>
</tr>
<tr>
<td>2. SM3 R-&gt;N</td>
<td>2000,000</td>
<td>2500,000</td>
<td>1180,556</td>
<td>1180,556</td>
<td>319,069</td>
</tr>
<tr>
<td>3. CS Braking</td>
<td>1180,800</td>
<td>1476,000</td>
<td>1180,556</td>
<td>1180,556</td>
<td>319,069</td>
</tr>
<tr>
<td>4. SM2 R-&gt;N</td>
<td>1180,556</td>
<td>1475,694</td>
<td>1180,556</td>
<td>1180,556</td>
<td>319,069</td>
</tr>
<tr>
<td>5. SM1 R-&gt;L</td>
<td>1180,556</td>
<td>868,056</td>
<td>1180,556</td>
<td>1180,556</td>
<td>319,069</td>
</tr>
<tr>
<td><strong>5-&gt;7</strong></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>1. Initial cond.</td>
<td>2000,000</td>
<td>2500,000</td>
<td>1554,054</td>
<td>1554,054</td>
<td>420,015</td>
</tr>
<tr>
<td>2. SM3 L-&gt;N</td>
<td>2000,000</td>
<td>2500,000</td>
<td>1554,054</td>
<td>1554,054</td>
<td>420,015</td>
</tr>
<tr>
<td>3.a SM1 R-&gt;L, SM5 R-&gt;L</td>
<td>2000,000</td>
<td>1470,588</td>
<td>420,015</td>
<td>420,015</td>
<td>420,015</td>
</tr>
<tr>
<td>3.b CS Braking</td>
<td>1299,840</td>
<td>889,588</td>
<td>420,015</td>
<td>420,015</td>
<td>420,015</td>
</tr>
<tr>
<td>4. SM3 N-&gt;R</td>
<td>1299,642</td>
<td>889,443</td>
<td>420,015</td>
<td>420,015</td>
<td>420,015</td>
</tr>
<tr>
<td><strong>6-&gt;8</strong></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>1. Initial cond.</td>
<td>2000,000</td>
<td>1470,588</td>
<td>2000,000</td>
<td>2000,000</td>
<td>540,541</td>
</tr>
<tr>
<td>2. SM2 N-&gt;L</td>
<td>2000,000</td>
<td>1470,588</td>
<td>2000,000</td>
<td>2000,000</td>
<td>540,541</td>
</tr>
<tr>
<td>3. SM5 R-&gt;L</td>
<td>2000,000</td>
<td>1470,588</td>
<td>0,000</td>
<td>2000,000</td>
<td>540,541</td>
</tr>
<tr>
<td>4. CS Braking</td>
<td>1260,586</td>
<td>926,901</td>
<td>273,857</td>
<td>1260,586</td>
<td>540,541</td>
</tr>
<tr>
<td>5. SM4 N-&gt;L</td>
<td>1260,586</td>
<td>926,901</td>
<td>273,857</td>
<td>1260,586</td>
<td>540,541</td>
</tr>
<tr>
<td><strong>7-&gt;9</strong></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>1. Initial cond.</td>
<td>2000,000</td>
<td>1470,588</td>
<td>694,444</td>
<td>694,444</td>
<td>694,444</td>
</tr>
<tr>
<td>2. SM3 R-&gt;N</td>
<td>2000,000</td>
<td>1470,588</td>
<td>694,444</td>
<td>694,444</td>
<td>694,444</td>
</tr>
<tr>
<td>3. CS Braking</td>
<td>694,400</td>
<td>510,588</td>
<td>694,444</td>
<td>694,444</td>
<td>694,444</td>
</tr>
<tr>
<td>4. SM2 R-&gt;L</td>
<td>694,400</td>
<td>510,588</td>
<td>694,444</td>
<td>694,444</td>
<td>694,444</td>
</tr>
<tr>
<td>5. CS Accel.</td>
<td>1326,165</td>
<td>975,121</td>
<td>460,474</td>
<td>1326,165</td>
<td>694,444</td>
</tr>
<tr>
<td>6. SM3 N-&gt;R</td>
<td>1326,165</td>
<td>975,121</td>
<td>460,474</td>
<td>1326,165</td>
<td>694,444</td>
</tr>
<tr>
<td></td>
<td>DS</td>
<td>CS</td>
<td>HS</td>
<td>MS</td>
<td>PC</td>
</tr>
<tr>
<td>-------</td>
<td>-------</td>
<td>-----</td>
<td>------</td>
<td>------</td>
<td>------</td>
</tr>
<tr>
<td>8-&gt;10</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>1. Initial cond.</td>
<td><strong>2000,000</strong></td>
<td>1470,588</td>
<td>434,492</td>
<td>2000,000</td>
<td>857,602</td>
</tr>
<tr>
<td>2. SM4 L-&gt;N</td>
<td>2000,000</td>
<td>1470,588</td>
<td>434,492</td>
<td>2000,000</td>
<td>857,602</td>
</tr>
<tr>
<td>3. CS Braking</td>
<td>1420,640</td>
<td>1044,588</td>
<td>649,190</td>
<td>1420,640</td>
<td>857,602</td>
</tr>
<tr>
<td>4. SM3 N-&gt;L</td>
<td>1420,315</td>
<td>1044,349</td>
<td>649,190</td>
<td>1420,315</td>
<td>857,602</td>
</tr>
<tr>
<td>9-&gt;11</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>1. Initial cond.</td>
<td><strong>2000,000</strong></td>
<td>1470,588</td>
<td>694,444</td>
<td>2000,000</td>
<td>1047,297</td>
</tr>
<tr>
<td>2. SM1 L-&gt;R</td>
<td>1493,976</td>
<td>1867,470</td>
<td>881,861</td>
<td>1493,976</td>
<td>1047,297</td>
</tr>
<tr>
<td>10-&gt;12</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>1. Initial cond.</td>
<td><strong>2000,000</strong></td>
<td>1470,588</td>
<td>914,149</td>
<td>2000,000</td>
<td>1207,623</td>
</tr>
<tr>
<td>2. SM1 L-&gt;R</td>
<td>1442,299</td>
<td>1802,874</td>
<td>1120,705</td>
<td>1442,299</td>
<td>1207,623</td>
</tr>
<tr>
<td>11-&gt;13</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>1. Initial cond.</td>
<td><strong>2000,000</strong></td>
<td>2500,000</td>
<td>1180,556</td>
<td>2000,000</td>
<td>1402,027</td>
</tr>
<tr>
<td>2. SM3 R-&gt;N</td>
<td>2000,000</td>
<td>2500,000</td>
<td>1180,556</td>
<td>2000,000</td>
<td>1402,027</td>
</tr>
<tr>
<td>3. CS Braking</td>
<td>1402,027</td>
<td>1752,000</td>
<td>1402,027</td>
<td>1402,027</td>
<td>1402,027</td>
</tr>
<tr>
<td>4. SM2 L-&gt;N</td>
<td>1402,027</td>
<td>1752,534</td>
<td>1402,027</td>
<td>1402,027</td>
<td>1402,027</td>
</tr>
<tr>
<td>5. SM1 R-&gt;N</td>
<td>1402,027</td>
<td>0,000</td>
<td>1402,027</td>
<td>1402,027</td>
<td>1402,027</td>
</tr>
</tbody>
</table>
Table E.3: Single Down-Shifts

<table>
<thead>
<tr>
<th></th>
<th>DS</th>
<th>CS</th>
<th>HS</th>
<th>MS</th>
<th>PC</th>
</tr>
</thead>
<tbody>
<tr>
<td>13-&gt;12</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>1. Initial Cond.</td>
<td>800,000</td>
<td>0,000</td>
<td>800,000</td>
<td>800,000</td>
<td>800,000</td>
</tr>
<tr>
<td>2. SM1 N-&gt;R</td>
<td>800,000</td>
<td>1000,000</td>
<td>800,000</td>
<td>800,000</td>
<td>800,000</td>
</tr>
<tr>
<td>3. SM2 N-&gt;L</td>
<td>800,000</td>
<td>1000,000</td>
<td>800,000</td>
<td>800,000</td>
<td>800,000</td>
</tr>
<tr>
<td>4. CS Accel.</td>
<td>955,463</td>
<td>1194,329</td>
<td>742,421</td>
<td>955,463</td>
<td>800,000</td>
</tr>
<tr>
<td>5. SM3 N-&gt;L</td>
<td>955,463</td>
<td>1194,329</td>
<td>742,421</td>
<td>955,463</td>
<td>800,000</td>
</tr>
</tbody>
</table>

| 12->11|          |        |        |        |        |
| 1. Initial Cond. | 800,000 | 1000,000 | 621,622 | 800,000 | 669,832 |
| 2. SM3 L->N | 800,000 | 1000,000 | 621,622 | 800,000 | 669,832 |
| 3. CS Accel. | 955,519 | 1194,399 | 564,022 | 955,519 | 669,832 |
| 4. SM3 N->R | 955,519 | 1194,399 | 564,022 | 955,519 | 669,832 |

| 11->10|          |        |        |        |        |
| 1. Initial Cond. | 800,000 | 1000,000 | 472,222 | 800,000 | 560,811 |
| 2. SM1 R->L | 1360,000 | 1000,000 | 472,222 | 800,000 | 560,811 |
| 3. SM3 R->N | 1360,000 | 1000,000 | 472,222 | 800,000 | 560,811 |
| 4. CS Braking | 928,785 | 682,930 | 424,524 | 928,785 | 560,811 |
| 5. SM3 N->L | 928,785 | 682,930 | 424,524 | 928,785 | 560,811 |

| 10->9|          |        |        |        |        |
| 1. Initial Cond. | 800,000 | 588,235 | 365,660 | 800,000 | 483,049 |
| 2. SM3 L->N | 800,000 | 588,235 | 365,660 | 800,000 | 483,049 |
| 3. CS Accel. | 922,468 | 678,285 | 320,301 | 922,468 | 483,049 |
| 4. SM3 N->R | 922,468 | 678,285 | 320,301 | 922,468 | 483,049 |

| 9->8|          |        |        |        |        |
| 1. Initial Cond. | 800,000 | 588,235 | 277,778 | 800,000 | 418,919 |
| 2. SM3 R->N | 800,000 | 588,235 | 277,778 | 800,000 | 418,919 |
| 3. CS Accel. | 976,954 | 718,348 | 212,239 | 976,954 | 418,919 |
| 4. SM4 N->L | 976,954 | 718,348 | 212,239 | 976,954 | 418,919 |

<p>| 8-&gt;7|          |        |        |        |        |
| 1. Initial Cond. | 800,000 | 588,235 | 173,797 | 800,000 | 343,041 |
| 2. SM4 L-&gt;N | 800,000 | 588,235 | 173,797 | 800,000 | 343,041 |
| 3. CS Braking | 343,040 | 252,235 | 343,040 | 343,040 | 343,040 |
| 4. SM2 L-&gt;R | 343,040 | 252,235 | 343,040 | 343,040 | 343,040 |
| 5. Beschl. | 987,958 | 726,440 | 343,041 | 343,041 | 343,041 |
| 6. SM3 N-&gt;R | 987,958 | 726,440 | 343,041 | 343,041 | 343,041 |</p>
<table>
<thead>
<tr>
<th></th>
<th>DS</th>
<th>CS</th>
<th>HS</th>
<th>MS</th>
<th>PC</th>
</tr>
</thead>
<tbody>
<tr>
<td>7→6</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>1.</td>
<td>Initial Cond.</td>
<td>800,000</td>
<td>588,235</td>
<td>277,778</td>
<td>277,778</td>
</tr>
<tr>
<td>2.</td>
<td>SM3 R→N</td>
<td>800,000</td>
<td>588,235</td>
<td>277,778</td>
<td>277,778</td>
</tr>
<tr>
<td>3.</td>
<td>SM5 L→R</td>
<td>800,000</td>
<td>588,235</td>
<td>1027,778</td>
<td>1027,778</td>
</tr>
<tr>
<td>4.</td>
<td>CS Accel.</td>
<td>1027,778</td>
<td>755,719</td>
<td>1027,778</td>
<td>1027,778</td>
</tr>
<tr>
<td>5.</td>
<td>SM2 R→N</td>
<td>1027,778</td>
<td>755,719</td>
<td>1027,778</td>
<td>1027,778</td>
</tr>
<tr>
<td>6→5</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>1.</td>
<td>Initial Cond.</td>
<td>800,000</td>
<td>588,235</td>
<td>800,000</td>
<td>800,000</td>
</tr>
<tr>
<td>2.</td>
<td>SM2 N→R</td>
<td>800,000</td>
<td>588,235</td>
<td>800,000</td>
<td>800,000</td>
</tr>
<tr>
<td>3.</td>
<td>SM1 L→R</td>
<td>800,000</td>
<td>1000,000</td>
<td>800,000</td>
<td>800,000</td>
</tr>
<tr>
<td>4.</td>
<td>CS Accel.</td>
<td>1029,565</td>
<td>1286,957</td>
<td>800,000</td>
<td>800,000</td>
</tr>
<tr>
<td>5.</td>
<td>SM3 N→L</td>
<td>1029,565</td>
<td>1286,957</td>
<td>800,000</td>
<td>800,000</td>
</tr>
<tr>
<td>5→4</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>1.</td>
<td>Initial Cond.</td>
<td>800,000</td>
<td>1000,000</td>
<td>621,622</td>
<td>621,622</td>
</tr>
<tr>
<td>2.</td>
<td>SM3 L→N</td>
<td>800,000</td>
<td>1000,000</td>
<td>621,622</td>
<td>621,622</td>
</tr>
<tr>
<td>3.</td>
<td>CS Accel.</td>
<td>1053,100</td>
<td>1316,375</td>
<td>621,622</td>
<td>621,622</td>
</tr>
<tr>
<td>4.</td>
<td>SM3 N→R</td>
<td>1053,100</td>
<td>1316,375</td>
<td>621,622</td>
<td>621,622</td>
</tr>
<tr>
<td>4→3</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>1.</td>
<td>Initial Cond.</td>
<td>800,000</td>
<td>1000,000</td>
<td>472,222</td>
<td>472,222</td>
</tr>
<tr>
<td>2.</td>
<td>SM3 R→N</td>
<td>800,000</td>
<td>1000,000</td>
<td>472,222</td>
<td>472,222</td>
</tr>
<tr>
<td>3.</td>
<td>SM1 R→L</td>
<td>800,000</td>
<td>588,235</td>
<td>472,222</td>
<td>472,222</td>
</tr>
<tr>
<td>4.</td>
<td>CS Accel.</td>
<td>1033,140</td>
<td>759,662</td>
<td>472,222</td>
<td>472,222</td>
</tr>
<tr>
<td>5.</td>
<td>SM3 N→L</td>
<td>1033,140</td>
<td>759,662</td>
<td>472,222</td>
<td>472,222</td>
</tr>
<tr>
<td>3→2</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>1.</td>
<td>Initial Cond.</td>
<td>800,000</td>
<td>588,235</td>
<td>365,660</td>
<td>365,660</td>
</tr>
<tr>
<td>2.</td>
<td>SM3 L→N</td>
<td>800,000</td>
<td>588,235</td>
<td>365,660</td>
<td>365,660</td>
</tr>
<tr>
<td>3.</td>
<td>CS Accel.</td>
<td>1053,100</td>
<td>774,338</td>
<td>365,660</td>
<td>365,660</td>
</tr>
<tr>
<td>4.</td>
<td>SM3 N→R</td>
<td>1053,100</td>
<td>774,338</td>
<td>365,660</td>
<td>365,660</td>
</tr>
<tr>
<td>2→1</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>1.</td>
<td>Initial Cond.</td>
<td>800,000</td>
<td>588,235</td>
<td>277,778</td>
<td>277,778</td>
</tr>
<tr>
<td>2.</td>
<td>SM3 R→N</td>
<td>800,000</td>
<td>588,235</td>
<td>277,778</td>
<td>277,778</td>
</tr>
<tr>
<td>3.</td>
<td>CS Accel.</td>
<td>1278,632</td>
<td>940,171</td>
<td>277,778</td>
<td>277,778</td>
</tr>
<tr>
<td>4.</td>
<td>SM4 N→L</td>
<td>1278,632</td>
<td>940,171</td>
<td>277,778</td>
<td>277,778</td>
</tr>
</tbody>
</table>
Table E.4: Double Down-Shifts

<table>
<thead>
<tr>
<th>13-&gt;11</th>
<th>DS</th>
<th>CS</th>
<th>HS</th>
<th>MS</th>
<th>PC</th>
</tr>
</thead>
<tbody>
<tr>
<td>1. Initial Cond.</td>
<td>800,000</td>
<td>0,000</td>
<td>800,000</td>
<td>800,000</td>
<td>800,000</td>
</tr>
<tr>
<td>2. SM1 N-&gt;R</td>
<td>800,000</td>
<td>1000,000</td>
<td>800,000</td>
<td>800,000</td>
<td>800,000</td>
</tr>
<tr>
<td>3. SM2 N-&gt;L</td>
<td>800,000</td>
<td>1000,000</td>
<td>800,000</td>
<td>800,000</td>
<td>800,000</td>
</tr>
<tr>
<td>4. CS Accel.</td>
<td>1141,205</td>
<td>1426,506</td>
<td>673,628</td>
<td>1141,205</td>
<td>800,000</td>
</tr>
<tr>
<td>5. SM3 N-&gt;R</td>
<td>1141,205</td>
<td>1426,506</td>
<td>673,628</td>
<td>1141,205</td>
<td>800,000</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>12-&gt;10</th>
<th>DS</th>
<th>CS</th>
<th>HS</th>
<th>MS</th>
<th>PC</th>
</tr>
</thead>
<tbody>
<tr>
<td>1. Initial Cond.</td>
<td>800,000</td>
<td>1000,000</td>
<td>621,622</td>
<td>800,000</td>
<td>669,832</td>
</tr>
<tr>
<td>2. SM1 R-&gt;L</td>
<td>1109,340</td>
<td>815,691</td>
<td>507,051</td>
<td>1109,340</td>
<td>669,832</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>11-&gt;9</th>
<th>DS</th>
<th>CS</th>
<th>HS</th>
<th>MS</th>
<th>PC</th>
</tr>
</thead>
<tbody>
<tr>
<td>1. Initial Cond.</td>
<td>800,000</td>
<td>1000,000</td>
<td>472,222</td>
<td>800,000</td>
<td>560,811</td>
</tr>
<tr>
<td>2. SM1 R-&gt;L</td>
<td>1070,968</td>
<td>787,476</td>
<td>371,864</td>
<td>1070,968</td>
<td>560,811</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>10-&gt;8</th>
<th>DS</th>
<th>CS</th>
<th>HS</th>
<th>MS</th>
<th>PC</th>
</tr>
</thead>
<tbody>
<tr>
<td>1. Initial Cond.</td>
<td>800,000</td>
<td>588,235</td>
<td>365,660</td>
<td>800,000</td>
<td>483,049</td>
</tr>
<tr>
<td>2. SM3 L-&gt;N</td>
<td>800,000</td>
<td>588,235</td>
<td>365,660</td>
<td>800,000</td>
<td>483,049</td>
</tr>
<tr>
<td>3. CS Accel.</td>
<td>1126,511</td>
<td>828,317</td>
<td>244,730</td>
<td>1126,511</td>
<td>483,049</td>
</tr>
<tr>
<td>4. SM3 N-&gt;R</td>
<td>1126,511</td>
<td>828,317</td>
<td>244,730</td>
<td>1126,511</td>
<td>483,049</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>9-&gt;7</th>
<th>DS</th>
<th>CS</th>
<th>HS</th>
<th>MS</th>
<th>PC</th>
</tr>
</thead>
<tbody>
<tr>
<td>1. Initial Cond.</td>
<td>800,000</td>
<td>588,235</td>
<td>277,778</td>
<td>800,000</td>
<td>418,919</td>
</tr>
<tr>
<td>2. SM3 R-&gt;N</td>
<td>800,000</td>
<td>588,235</td>
<td>277,778</td>
<td>800,000</td>
<td>418,919</td>
</tr>
<tr>
<td>3. CS Braking</td>
<td>418,919</td>
<td>308,029</td>
<td>418,919</td>
<td>418,919</td>
<td>418,919</td>
</tr>
<tr>
<td>4. SM2 L-&gt;R</td>
<td>418,919</td>
<td>308,029</td>
<td>418,919</td>
<td>418,919</td>
<td>418,919</td>
</tr>
<tr>
<td>5. CS Accel.</td>
<td>1206,486</td>
<td>887,122</td>
<td>418,919</td>
<td>1206,486</td>
<td>418,919</td>
</tr>
<tr>
<td>6. SM3 N-&gt;R</td>
<td>1206,486</td>
<td>887,122</td>
<td>418,919</td>
<td>1206,486</td>
<td>418,919</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>8-&gt;6</th>
<th>DS</th>
<th>CS</th>
<th>HS</th>
<th>MS</th>
<th>PC</th>
</tr>
</thead>
<tbody>
<tr>
<td>1. Initial Cond.</td>
<td>800,000</td>
<td>588,235</td>
<td>173,797</td>
<td>800,000</td>
<td>343,041</td>
</tr>
<tr>
<td>2. SM4 L-&gt;N</td>
<td>800,000</td>
<td>588,235</td>
<td>173,797</td>
<td>800,000</td>
<td>343,041</td>
</tr>
<tr>
<td>3. CS Braking</td>
<td>343,041</td>
<td>252,236</td>
<td>343,041</td>
<td>343,041</td>
<td>343,041</td>
</tr>
<tr>
<td>4. SM2 L-&gt;N</td>
<td>343,041</td>
<td>252,236</td>
<td>343,041</td>
<td>343,041</td>
<td>343,041</td>
</tr>
<tr>
<td>5. SM5 L-&gt;R</td>
<td>1269,251</td>
<td>933,273</td>
<td>1269,251</td>
<td>1269,251</td>
<td>343,041</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>7-&gt;5</th>
<th>DS</th>
<th>CS</th>
<th>HS</th>
<th>MS</th>
<th>PC</th>
</tr>
</thead>
<tbody>
<tr>
<td>1. Initial Cond.</td>
<td>800,000</td>
<td>588,235</td>
<td>277,778</td>
<td>277,778</td>
<td>277,778</td>
</tr>
<tr>
<td>2. SM3 R-&gt;N</td>
<td>800,000</td>
<td>588,235</td>
<td>277,778</td>
<td>277,778</td>
<td>277,778</td>
</tr>
<tr>
<td>3a SM5 L-&gt;R</td>
<td>800,000</td>
<td>1000,000</td>
<td>1027,778</td>
<td>1027,778</td>
<td>277,778</td>
</tr>
<tr>
<td>3b SM1 L-&gt;R</td>
<td>1322,705</td>
<td>1653,382</td>
<td>1027,778</td>
<td>1027,778</td>
<td>277,778</td>
</tr>
<tr>
<td>4. CS Accel.</td>
<td>1322,705</td>
<td>1653,382</td>
<td>1027,778</td>
<td>1027,778</td>
<td>277,778</td>
</tr>
<tr>
<td>5. SM3 N-&gt;L</td>
<td>1322,705</td>
<td>1653,382</td>
<td>1027,778</td>
<td>1027,778</td>
<td>277,778</td>
</tr>
<tr>
<td></td>
<td>DS</td>
<td>CS</td>
<td>HS</td>
<td>MS</td>
<td>PC</td>
</tr>
<tr>
<td>---</td>
<td>------</td>
<td>-----</td>
<td>------</td>
<td>------</td>
<td>-------</td>
</tr>
<tr>
<td>6→4</td>
<td>Initial Cond.</td>
<td>800,000</td>
<td>588,235</td>
<td>800,000</td>
<td>800,000</td>
</tr>
<tr>
<td></td>
<td>SM2 N→R</td>
<td>800,000</td>
<td>588,235</td>
<td>800,000</td>
<td>800,000</td>
</tr>
<tr>
<td></td>
<td>SM1 L→R</td>
<td>800,000</td>
<td>1000,000</td>
<td>800,000</td>
<td>800,000</td>
</tr>
<tr>
<td></td>
<td>CS Accel.</td>
<td>1355,294</td>
<td>1694,118</td>
<td>800,000</td>
<td>800,000</td>
</tr>
<tr>
<td></td>
<td>SM3 N→R</td>
<td>1355,294</td>
<td>1694,118</td>
<td>800,000</td>
<td>800,000</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th></th>
<th>DS</th>
<th>CS</th>
<th>HS</th>
<th>MS</th>
<th>PC</th>
</tr>
</thead>
<tbody>
<tr>
<td>5→3</td>
<td>Initial Cond.</td>
<td>800,000</td>
<td>1000,000</td>
<td>621,622</td>
<td>621,622</td>
</tr>
<tr>
<td></td>
<td>SM1 R→L</td>
<td>1360,000</td>
<td>1000,000</td>
<td>621,622</td>
<td>621,622</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th></th>
<th>DS</th>
<th>CS</th>
<th>HS</th>
<th>MS</th>
<th>PC</th>
</tr>
</thead>
<tbody>
<tr>
<td>4→2</td>
<td>Initial Cond.</td>
<td>800,000</td>
<td>1000,000</td>
<td>472,222</td>
<td>472,222</td>
</tr>
<tr>
<td></td>
<td>SM1 R→L</td>
<td>1360,000</td>
<td>1000,000</td>
<td>472,222</td>
<td>472,222</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th></th>
<th>DS</th>
<th>CS</th>
<th>HS</th>
<th>MS</th>
<th>PC</th>
</tr>
</thead>
<tbody>
<tr>
<td>3→1</td>
<td>Initial Cond.</td>
<td>800,000</td>
<td>588,235</td>
<td>365,660</td>
<td>365,660</td>
</tr>
<tr>
<td></td>
<td>SM3 L→N</td>
<td>800,000</td>
<td>588,235</td>
<td>365,660</td>
<td>365,660</td>
</tr>
<tr>
<td></td>
<td>CS Accel.</td>
<td>1683,160</td>
<td>1237,618</td>
<td>365,660</td>
<td>365,660</td>
</tr>
<tr>
<td></td>
<td>SM4 N→L</td>
<td>1683,160</td>
<td>1237,618</td>
<td>365,660</td>
<td>365,660</td>
</tr>
</tbody>
</table>
Appendix F.  Efficiency

**Forward Gear**

\[ T_A = +1 \]

\[ T_B = -i \cdot \eta \cdot T_A = -10,43 \cdot \eta \cdot 1 = -10,43 \eta \]

\[ T_S = -T_B = +10,43 \eta \]

\[ T_1 = \frac{-T_S}{1 - i_0 \cdot \eta_0} = \frac{-10,43 \eta}{1 - (-2,7) \cdot 0,976} = -2,87 \eta \]

\[ T_3 = -i_0 \cdot \eta_0 \cdot T_1 = (-2,7) \cdot 0,976 \cdot -2,87 \eta = -7,56 \eta \]

\[ T_4 = -T_5 = +7,56 \eta \]

\[ T_5 = -\frac{1}{i_{45}} \cdot \eta_{\text{gear-set}} \cdot T_4 = -\frac{1}{3,077} \cdot 0,99^2 \cdot +7,56 \eta = +2,41 \eta \]

\[ T_6 = -T_5 = -2,41 \eta \]

\[ T_7 = -\frac{1}{i_{67}} \cdot \eta_{\text{gear-set}} \cdot T_6 = -\frac{1}{1,36} \cdot 0,99^4 \cdot -2,51 \mu = +1,75 \eta \]

\[ T_7 = -(T_A + T_1) = -(1 - 2,87 \eta) \]

\[ \eta = 0,896 \]

**Backward Gear**

\[ T_A = +1 \]

\[ T_B = -i \cdot \eta \cdot T_A = -(38,19) \cdot \eta \cdot 1 = +38,19 \eta \]

\[ T_S = -T_B = -38,19 \eta \]

\[ T_1 = \frac{-T_S}{1 - i_0 \cdot \eta_0} = \frac{38,19 \eta}{1 - (-2,7) \cdot 0,976} = +10,51 \eta \]

\[ T_3 = -i_0 \cdot \eta_0 \cdot T_1 = (-2,7) \cdot 0,976 \cdot 10,51 \eta = +27,68 \eta \]

\[ T_4 = -T_3 = -27,68 \eta \]

\[ T_5 = -\frac{1}{i_{45}} \cdot \frac{1}{\eta_{\text{gear-set}}} \cdot T_4 = -\frac{1}{3,077} \cdot 0,99^2 \cdot -27,68 \eta = -9,18 \eta \]

\[ T_6 = -T_5 = +9,18 \eta \]

\[ T_7 = -\frac{1}{i_{67}} \cdot \frac{1}{\eta_{\text{gear-set}}} \cdot T_6 = -\frac{1}{0,80} \cdot 0,99^4 \cdot 9,18 \mu = -11,59 \eta \]

\[ T_7 = -(T_A + T_1) = -(1 + 10,51 \eta) \]

\[ \eta = 0,922 \]
## Appendix G. Bearing Characteristics

<table>
<thead>
<tr>
<th>Bearing Type</th>
<th>Radial Load</th>
<th>Axial Load in Both Directions</th>
<th>Length Adjustment Inside the Bearing</th>
<th>Dismountable Bearing</th>
<th>Increased Accuracy</th>
<th>Suitability for High Number of Revolutions</th>
<th>Low-noise</th>
<th>High Stiffness</th>
<th>Low Friction</th>
<th>Fixed Bearing</th>
<th>Floating Bearing</th>
</tr>
</thead>
<tbody>
<tr>
<td>Grooved Ball Bearing</td>
<td>Very good</td>
<td>Good</td>
<td>n.a.</td>
<td>n.a.</td>
<td>Possible</td>
<td>Very good</td>
<td>Very Good</td>
<td>Possible</td>
<td>Good</td>
<td>Possible</td>
<td>Possible</td>
</tr>
<tr>
<td>Axial Grooved Ball Bearing</td>
<td>n.a.</td>
<td>Good</td>
<td>n.a.</td>
<td>Very good</td>
<td>Good</td>
<td>Possible</td>
<td>With modification</td>
<td>Possible</td>
<td>Good</td>
<td>Possible</td>
<td>n.a.</td>
</tr>
<tr>
<td>Cylinder Roller Bearing</td>
<td>Very good</td>
<td>n.a.</td>
<td>Very good</td>
<td>Very good</td>
<td>Possible</td>
<td>Good</td>
<td>With modification</td>
<td>Good</td>
<td>Good</td>
<td>n.a.</td>
<td>n.a.</td>
</tr>
<tr>
<td>Bearing Type</td>
<td>Radial Load</td>
<td>Axial Load</td>
<td>Length Adjustment</td>
<td>Dismountable Bearing</td>
<td>Increased Accuracy</td>
<td>Suitability for High Number of Revolutions</td>
<td>Low-noise</td>
<td>High Stiffness</td>
<td>Low Friction</td>
<td>Fixed Bearing</td>
<td>Floating Bearing</td>
</tr>
<tr>
<td>----------------------------</td>
<td>----------------------</td>
<td>------------</td>
<td>-------------------</td>
<td>----------------------</td>
<td>-------------------</td>
<td>--------------------------------------------</td>
<td>-----------</td>
<td>---------------</td>
<td>--------------</td>
<td>---------------</td>
<td>------------------</td>
</tr>
<tr>
<td><strong>Taper Roller Bearing</strong></td>
<td>Very good</td>
<td>Very good in one direction</td>
<td>n.a.</td>
<td>Good</td>
<td>Good</td>
<td>Possible</td>
<td>With modification</td>
<td>Very good</td>
<td>Good</td>
<td>Very good</td>
<td>Good</td>
</tr>
<tr>
<td><strong>Spherical Roller Bearing</strong></td>
<td>Very good</td>
<td>Possible</td>
<td>n.a.</td>
<td>n.a.</td>
<td>n.a.</td>
<td>Possible</td>
<td>With modification</td>
<td>Good</td>
<td>Possible</td>
<td>Good</td>
<td>n.a.</td>
</tr>
<tr>
<td><strong>Axial Cylinder Roller Bearing</strong></td>
<td>n.a.</td>
<td>Very good both directions</td>
<td>n.a.</td>
<td>n.a.</td>
<td>Possible</td>
<td>With modification</td>
<td>n.a.</td>
<td>Good</td>
<td>n.a.</td>
<td>Good</td>
<td>n.a.</td>
</tr>
</tbody>
</table>
Appendix H. 2D Constructions of Concepts

Figure H.1: Concept A
Figure H.2: Concept C
Figure H.3: Concept D
Figure H.4: Concept E
Figure H.5: Concept F
Appendix I.  Pressure in Split- and Range-Group

The pressure applied to the cylinder piston is equal for both the split group and the range group. The system pressure will be about 8.5 bar which is then adjusted by pressure valves in order to get the right pressure on the piston.

Since the area of both sides of the piston isn’t of the same size, the pressure will be higher for the smaller piston area. The pressure can now be extracted from Figure I.1 and Figure I.2.

The figures are test values provided by DaimlerChrysler and show the complete course of the synchromesh in the range group.

Figure I.1: Cylinder pressure on large piston area

The resulting pressure that translates the cylinder piston is the difference between the applied pressure and the counter-pressure. In Figure I.1 the applied pressure is denoted by “Druck1 NG Range S” and the counter-pressure is denoted by “Druck2 NG Range L”. The maximum difference between these curves will be the pressure that is used for synchronizing, and is in this case about $\Delta p = 3.0$ bar for the large piston area.
The value for the pressure on the small piston area is extracted in exactly the same way, but then from Figure I.2. The value of this pressure is about $\Delta p = 5.5$ bar.

![Figure I.2: Cylinder pressure on small piston area](image)

There also exists a ratio between the piston force and the force actually applied to the sliding collar. This ratio enlarges the piston force, although by such a small fraction that for the simplicity of the calculations the ratio is set to one.
Appendix J. Radio Frequency Identification

RFID can have different functions in the automotive industry, functions like car-tracking, measuring tire pressure, key-identification, toll roads and part-tracking. Of course there can be thought of many more applications, but the most interesting for the construction of this gearbox is part-tracking.

The use of RFID as a part-tracking device can have the following fields of application:
- Inventory Management
- Theft Control
- Assembly
- Brand Authentication
- Maintenance
- Recall
- Recycling

In order to have the ability of tracking the part, a Tag needs to be attached to the part. This can be done in two ways, either the Tag is attached to the part’s packaging or carrier, or the Tag is attached to the part itself. In this case the latter is the most interesting method because that way the field of application is much larger.

So with the Tag directly attached to the part in combination with the new gearbox the fields of application can be analysed for their functionality. Only the fields that are interesting are pointed out in the following section.

Assembly

There are three implementation options of RFID:
1. RFID makes it possible to accomplish full automation of the assembly process. An average truck consists of about 13,000 parts and if Tags are attached to all parts, the system will be able to check if different parts fit together.
2. Build-to-Order (BTO) Management is very common in truck production, because every customer has its own wishes. A disadvantage is that it increases the number of components that have to be available and the potential for incorrect assemblies. If Tags are attached to parts that are typically different from truck to truck, this problem will be solved.
3. Work-in-Progress Tracking. RFID makes it possible of knowing the status of every product in the assembly process can help the planning of the output, improve the capacity utilization and the product quality.

Maintenance
The technician can see what parts are used and what the history of the parts is. For example spark plugs: The technician can see how often they get changed in a certain time period. Such kind of history can predict how the maintenance should take place in future.

Recall
If parts have to be recalled due to a malfunction it is important to know exactly on hat trucks the wrong part exists. RFID makes this possible and that way makes the recall much faster and quiet, because not all cars have to be recalled.

Recycling
Recycling means tearing the truck down and sorting all parts which is a process that every manufacturer has to deal with in future. Each part must be classified in order to be sold or recycled. The latter is done by part composition.
Today the sorting is done manually, but with RFID it is possible to do this automatically.

Finally there is a field of application which is not mentioned in the list above, but certainly can be very interesting for future truck production.
It is known that Active Tags can be coupled to sensors and process the data gathered by the sensor. These sensors can be applied a variety of parts like for example the gear wheels. The sensor in combination with the Tag notices when the gear wheel is damaged and gives the driver a notification so he can make the appropriate decision. As a result of the quick notice of a problem the driver can stop the truck directly and possibly avoid extra damage to the gearbox.

There are still a lot of challenges in RFID technology, for example:
- The lack of industrial standards.
- The interference of RFID with metallic objects.
- Little integration of RFID technology in standard software.
- Little information on practical experiences, because the technology is still new.
- Benefits and costs are distributed unequally between the members of the value chain.
Conclusion
RFID technology will definitely be implemented in a few years, because there is more to know about every part.
The main advantages in the direct future will be the part-tracking in assemblies and the combination with sensors. But of course also inventory management is a field of application that shouldn’t be forgotten.
Bibliography


[22] Marciszko, F. (2004), Torque Sensor Based Powertrain Control


List of Figures

Figure 2.1: Part of the Product Development Process .......................................................... 9
Figure 2.2: Basis .................................................................................................................. 12
Figure 2.3: 2D-Construction ............................................................................................. 14
Figure 2.4: Part Improvements .......................................................................................... 15
Figure 2.5: Design for mass production .......................................................................... 16
Figure 3.1: Conventional 16-Speed Transmission ............................................................... 24
Figure 3.2: Schematical representation of the power-split transmission ......................... 26
Figure 5.1: Engine Characteristic of OM 502 LA ............................................................... 35
Figure 5.2: Transmission ratios of the forward and the reverse gears ......................... 38
Figure 5.3: Shift diagram: Vehicle speed vs. engine speed resulting .............................. 39
Figure 5.4: Vehicle Force vs. Vehicle Speed ................................................................. 41
Figure 5.5: Speed of the shafts during single up-shifts ................................................... 43
Figure 5.6: Left: Speed of the shafts during “uneven” double up-shifts. Right: Speed of the
       shafts during “even” double up-shifts ........................................................................ 43
Figure 5.7: Speed of the shafts during single down-shifts .............................................. 44
Figure 5.8: Left: Speed of the shafts during “uneven” double up-shifts. Right: Speed of the
       shafts during “even” double up-shifts........................................................................ 45
Figure 6.1: Power Flow through a Conventional Transmission ........................................ 49
Figure 6.2: Power Flow through the Power-Split Transmission ....................................... 50
Figure 6.3: Forward-gear with back-flow over the Gear Wheels ..................................... 51
Figure 6.4: Backward-gear with back-flow over the main- and driving Shaft ............... 53
Figure 6.5: Notations for torque calculations .................................................................... 57
Figure 7.1: Bearing Types ............................................................................................... 66
Figure 7.2: Concept A - One Countershaft; Small Distance Fixation ............................ 68
Figure 7.3: Concept A - One Countershaft; Large Distance Fixation ............................. 69
Figure 7.4: Concept B - One Countershaft ..................................................................... 70
Figure 7.5: Concept C - One Countershaft ..................................................................... 71
Figure 7.6: Concept D - Two Countershafts .................................................................... 72
Figure 7.7: Concept E - Two Countershafts .................................................................... 73
Figure 7.8: Concept F - Two Countershafts .................................................................... 74
Figure 8.1: 2D Construction of Concept A; Large Distance Fixation ............................ 78
Figure 8.2: 2D Construction of Concept C ...................................................................... 79
Figure 8.3: Bearing directly connected the main shaft ..................................................... 80
List of Tables

Table 2.1: The concept selection matrix .............................................................................. 11
Table 3.1: Comparison of direct-gear and overdrive .......................................................... 22
Table 5.1: Number of teeth and gear ratios of the gear wheels and the planetary gear ...... 37
Table 5.2: Vehicle Parameters ............................................................................................. 39
Table 6.1: Power-flow through the main shaft (P₁), and though the gear wheels (P₁₁)........ 51
Table 6.2: Moments at the transmission parts in first gear .................................................. 52
Table 6.3: Efficiency in Power-Split Situation ..................................................................... 56
Table 6.4: Efficiency Comparison ....................................................................................... 56
Table 6.5: Efficiency of the transmission with back-flow of the power ............................... 58
Table 6.6: Gear ratios and transmission ratios for a crawler .............................................. 59
Table 6.7: Power-flow through the driving shaft and main shaft (P₁), and through the gear
wheels (P₁₁), in forward-gear ........................................................................................ 59
Table 6.8: Efficiencies of the forward-gear at different gear ratios ...................................... 60
Table 6.9: Reverse transmission ratios at the different gear ratios of Table 6.6 ................. 60
Table 6.10: Reverse Vehicle Speed ..................................................................................... 61
Table 8.1: Selection matrix for bearing concepts ............................................................... 85
Table 9.1: Mass Moments of Inertia ................................................................................... 89
Table 9.2: DaimlerChrysler specifications of a standard counter shaft brake. ................. 92
Table 9.3: Specifications of the standard split-group synchromesh .................................. 98
Table 9.4: Specifications of the standard range-group synchromesh .............................. 101
Table 11.1: Selection table for coupling teeth .................................................................. 117
Table E.1: Single Up-Shifts ............................................................................................... 156
Table E.2: Double Up-Shifts .............................................................................................. 159
Table E.3: Single Down-Shifts .......................................................................................... 161
Table E.4: Double Down-Shifts ....................................................................................... 163
**Glossary of Terms**

<table>
<thead>
<tr>
<th>Term</th>
<th>Definition</th>
</tr>
</thead>
<tbody>
<tr>
<td>Backward Gear</td>
<td>Use of the intermediate gear in power-split situation, providing an extra reverse gear.</td>
</tr>
<tr>
<td>Bearing Layout</td>
<td>The way bearings are arranged in the transmission.</td>
</tr>
<tr>
<td>Connection Teeth</td>
<td>The teeth securing the connection between two or more rotating parts.</td>
</tr>
<tr>
<td>Constant Mesh</td>
<td>Device that is able to connect two rotating parts of the transmission.</td>
</tr>
<tr>
<td>Direct Gear</td>
<td>A final gear that has a transmission ratio of exactly $i = 1$.</td>
</tr>
<tr>
<td>Final Drive</td>
<td>Gear ratio at the axles of the driven wheels.</td>
</tr>
<tr>
<td>Forward Gear</td>
<td>Use of the intermediate gear in power-split situation, providing an extra forward driving gear.</td>
</tr>
<tr>
<td>Gear Ratio</td>
<td>Ratio between two gearwheels</td>
</tr>
<tr>
<td>Group Transmission</td>
<td>Transmission consisting of two or more groups of gearwheels.</td>
</tr>
<tr>
<td>Main-Group</td>
<td>Set of gearwheels that provide the main gears of a transmission.</td>
</tr>
<tr>
<td>Overdrive</td>
<td>A final gear that has a transmission ratio smaller than $i = 1$.</td>
</tr>
<tr>
<td>Power-Split Transmission</td>
<td>Transmission that splits the input power over two or more branches in the transmission. Finally the power will unite again and exit at the transmission output.</td>
</tr>
<tr>
<td>Range-Group</td>
<td>Set of gearwheels that add an additional ratio to the ratios of the main-group.</td>
</tr>
<tr>
<td>Reverse Gear</td>
<td>Use of the intermediate gear in conventional situation, making the vehicle move in reverse direction</td>
</tr>
<tr>
<td>Split-Group</td>
<td>Set of gearwheels that splits the ratios of the main-group in low-speed and high-speed.</td>
</tr>
<tr>
<td>Synchromesh</td>
<td>Device that is able to synchronize and connect to rotation parts of the transmission.</td>
</tr>
<tr>
<td>Transmission Ratio</td>
<td>Ratio between the transmission input and output</td>
</tr>
<tr>
<td>Three-Shaft Constant Mesh</td>
<td>Constant mesh connecting the driving shaft, main shaft and hollow shaft.</td>
</tr>
<tr>
<td>Three-Shaft Synchromesh</td>
<td>Synchromesh connecting the driving shaft, main shaft and hollow shaft.</td>
</tr>
</tbody>
</table>