Gisbert Lechner · Harald Naunheimer

Automotive Transmissions

Fundamentals, Selection, Design and Application

In Collaboration with Joachim Ryborz

With 370 Figures

Springer

ν	Speed ratio, kinematic viscosity
ρ	Density, angle of friction of the claws
σ	Direct stress
σ_b	Bending stress
σ_D	Fatigue strength
σ_H	Hertzian stresses
σ_v	Reference stress
τ	Torsional stress, torque increase with combustion engine
φ	Gear step, bending angle
φ_1	Basic step with progressive stepping
φ_2	Progression factor with progressive stepping
φ_{th}	Gear step with geometrical stepping
ω	Angular velocity

Subscripts

0	Nominal or initial state
1	Pinion (= small gear), input
2	Wheel (= large gear), output
1, 2, 3, ...	At point 1, 2, 3, ...
A	Offer, related to area, drive shaft, power train, moving off
B	Demand, brake
C	Clutch
CG	Constant gear
CS	Countershaft
D	Duration, fatigue-resistant, deficit, opening, direct drive
E	End
Ex	Excess
F	Vehicle, root, free-wheel
G	Gearbox
H	Adhesion, main, main gearbox, main shaft wheel, ring gear, high (= fast)
IS	Input shaft
L	Air, load, low (= slow)
L, L1, L2	At bearing point, at bearing point 1, 2
M	Engine, motor, model
MS	Main shaft
N	Rear-mounted range-change unit
OS	Output shaft
P	Pump, pump wheel, planetary step
PV	Pump test
Q	Transverse
R	Reverse gear, roll, slip, friction, wheel, range-change unit, reactor
Roll	Roll
S	Status, system, splitter unit
Sch	Pulsating (strength)
St	Gradient
T	Turbine wheel, drive
TC	Torque converter
U	Circumference
V	Front-mounted splitter unit, variator, loss, trial

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Translated from the German Edition "Fahrzeuggetriebe"

ISBN 3-540-65903-X Springer-Verlag Berlin Heidelberg New York

Library Congress Cataloging-in-Publication Data applied for
Die Deutsche Bibliothek - CIP-Einheitsaufnahme
Lechner, Gisbert: Automotive transmissions : fundamentals, selection, design and application / Gisbert Lechner ; Harald Naunheimer. In collab. with Joachim Ryborz. -Berlin; Heidelberg; New York; Barcelona; Hong Kong; London; Milan; Paris; Singapore; Tokyo: Springer, 1999
Einheitssacht.: Fahrzeuggetriebe <engl.>
ISBN 3-540-65903-X

Springer-Verlag Berlin Heidelberg New York
a member of BertelsmannSpringer Science+Business Media GmbH

Printed in Germany

Typesetting: camera-ready by authors
Cover design: medio GmbH, Berlin
62/3111-5 4 3 2 1 - Printed on acid-free paper

Preface to the English Edition

"Automotive Transmissions" was first published in German in May 1994. It was so well received that we decided to publish the book in English, especially in view of the trend to market globalisation.

This book gives a full account of the development process for automotive transmissions. It seeks to impart lines of reasoning, demonstrate approaches, and provide comprehensive data for the practical task of developing automotive transmissions. Much of the content is concerned with aspects of technology and production in the field of automotive transmissions that are of general validity, and hence of enduring relevance. The dynamics of the automotive transmission market since 1994 is reflected in numerous new types of transmission. Principal factors include the increasing use of electronics, light-weight construction, and the automation of manual gearboxes. Chapter 12 considers numerous production designs to illustrate the theoretical principles expounded in the earlier chapters, considering the main types of transmission and examining important detail solutions incorporated in specific mechanisms. Today's current design engineering is no longer state of the art tomorrow, with the next change of model. The designs presented here therefore claim to represent the different types of transmission design considered, rather than the latest production technology.

Certain aspects of the book relate to the situation in Germany, particularly as regards transport systems and the economic significance of motor vehicles.

This English language edition could not have come to fruition without the assistance of many contributors. We are particularly indebted to Dipl-Ing Joachim Ryborz as the manager and co-ordinator of the project, and to his assistants at the Institute of Machine Components (IMA), University of Stuttgart.

We would also like to thank Stephen Day of Übersetzungsbüro Herrera for his professional translation of this book, and Dr Ian Cole for proof-reading the final text.

We also wish to gratefully acknowledge the consistent financial support of the following companies: Audi AG, Robert Bosch GmbH, BMW AG, DaimlerChrysler AG, Ford Werke AG, GETRAG Getriebe- und Zahnradfabrik Hermann Hagenmeyer GmbH & Cie, Krupp Berco Bautechnik GmbH, Mannesmann Sachs AG, Adam Opel AG, Dr.-Ing. h.c. Porsche AG, Renk AG, Volkswagen AG, and ZF Friedrichshafen AG.

Stuttgart and Augsburg
February 1999

Gisbert Lechner

Harald Naunheimer

Preface to the German Edition

It was in 1953 that H. Reichenbächer wrote the first book on motor vehicle transmission engineering. At that time the German motor industry produced 490 581 vehicles including cars, vans, trucks, busses and tractor-trailer units. In 1992 production had reached 5.2 million. The technology at that time only required coverage of certain aspects, and Mr Reichenbächer's book accordingly restricted itself to basic types of gearbox, gear step selection, gear-sets with fixed axles, epicyclic systems, Föttinger couplings and hydrodynamic transmissions.

Automotive engineering and the technology of mechanism design have always been subject to evolution. The current state of the art is characterised by the following interrelations:

Environment ⇔ Traffic ⇔ Vehicle ⇔ Transmission.

Questions such as economy, environment and ease of use are paramount. The utility of a transmission is characterised by its impact on the traction available, on fuel consumption and reliability, service life, noise levels and the user-friendliness of the vehicle.

There are new techniques which now have to be taken into account, relating to development methodology, materials technology and notably strength calculation. Examples include serviceability calculations, the introduction of specific flank corrections, taking account of housing deformation, and the need for light-weight construction.

Transmission design engineering bas been enriched by numerous variants. The manual two-stage countershaft transmission, preferred for longitudinal engines, and the single-stage countershaft transmission preferred for transverse engines now have many subvariants, e.g. automatic transmissions, continuously variable transmissions, torque converter clutch transmissions, twin clutch transmissions, and transmissions for all-wheel drive.

The engine and transmission must increasingly be considered as one functional unit. The terms used are "power train matching" and "engine/transmission management". This can only be achieved by an integrated electronic management system covering the mechanical components in both engine and transmission.

The technique of Systematic Engineering developed in the 1960's, and the increasing use of computers for design, simulation and engineering (CAD), are resulting in ever-reducing development cycles. This trend is reinforced by competitive pressures. Systematic product planning is another significant factor in this regard.

It was therefore necessary to create an entirely new structure for the present book "Automotive Transmissions". Modern developments have to be taken into account. The great diversity and range of issues in developing transmissions made it difficult to select the material for this completely new version of "Automotive Transmissions", especially within the prevailing constraints. Not every topic could be covered in detail. In those places where there is an established literature, the authors have chosen to rely on it in the interests of brevity.

The purpose of this book is to describe the development of motor vehicle transmissions as an ongoing part of the vehicle development system. Only by actively taking this interaction into account is it possible to arrive at a fully viable transmission design. The aim is to highlight the basic interrelations between the drive unit, the vehicle and the transmission on the one hand, and their functional features such as appropriate gear selection, correct gear step, traction profile, fuel consumption, service life and reliability on the other. Of course another major concern was to represent the various engineering designs of modern vehicle transmissions in suitable design drawings.

The book is addressed to all engineers and students of automotive engineering, but especially to practitioners and senior engineers working in the field of transmission development. It is intended as a reference work for all information of importance to transmission development, and is also intended as a guide to further literature in the field.

Without the assistance of numerous people this book would not have been written. We would like to thank Dr Heidrun Schröpel, Mr Wolfgang Elser, Dr Ekkehard Krieg, Dr Winfried Richter, Mr Thomas Spörl, Mr Thilo Wagner, Dr Georg Weidner and Prof Lothar Winkler for researching and revising chapters. We also wish to acknowledge the contribution of numerous assistants and postgraduates for important work on specific aspects.

We wish to thank Christine Häbich for her professional editing. We would like to thank many employees and scientific assistants of the IMA (Institut für Maschinenelemente) for reviewing and checking various parts of the text.

Such a book cannot be published without current practical illustrations. The publishers wish to acknowledge their gratitude to numerous companies for making illustrations available: Audi AG, BMW AG, Eaton GmbH, Fichtel & Sachs AG, Ford Werke AG, GETRAG, Mercedes-Benz AG, Adam Opel AG, Dr.-Ing. h.c. Porsche AG, and Volkswagen AG. We are particularly indebted to ZF Friedrichshafen AG who have always been most forthcoming in responding to our numerous requests for graphic material.

We are also indebted to Springer-Verlag for publishing this book. We would particularly like to thank Mr M Hofmann, whose faith in our project never wavered, and whose gentle but firm persistence ensured that the book did indeed reach completion. Dr Merkle then prepared the work for printing. We must also thank the publisher of the "Design Engineering Books" series, Prof Gerhard Pahl for his patience and advice. Our thanks especially to our families for their understanding and support.

Stuttgart
May 1994

Gisbert Lechner

Harald Naunheimer

Contents

Terms and Symbols

A formula you cannot derive is a corpse in the brain /C. WEBER/

Physical variables are related by mathematical formulae. These can be expressed in two different ways:

- quantity equations,
- unit equations.

Quantity Equations

Quantity equations are independent of the unit used, and are of fundamental application. Every symbol represents a physical quantity, which can have different values:

Value of the quantity = numerical value x unit .

Example: Power P is generally defined by the formula

$$P = T\,\omega \ , \tag{1}$$

where T stands for torque and ω stands for angular velocity.

Unit Equations

If an equation recurs frequently or if it contains constants and material values, it is convenient to combine the units, in which case they are no longer freely selectable.

In unit equations the symbols incorporate only the numerical value of a variable. The units in unit equations must therefore be precisely prescribed.

Example: In order to calculate the effective power P in kW at a given rotational speed n in 1/min, the above equation (1) becomes the unit equation

$$P = \frac{T\,n}{9550} \ . \tag{2}$$

The unit equation (2) applies where the prescript P is expressed in kW, T in Nm and n in 1/min.

Terms and Symbols

(Only those which occur frequently; otherwise see text)

A	Surface area, transverse couple surface area = projection of vehicle front area
A_R	Synchroniser friction contact
B_{10}	System service life for a failure probability of 10%
B_x	System service life for a failure probability of x%
C	Pitch point, dynamic contact figure, constant
CC	Torque converter lock-up clutch
CG	Constant gear
CG_H	Front-mounted splitter constant high
CG_L	Front-mounted splitter constant low
CG_{main}	Main gear unit constant
CG_R	Range constant
D	Diameter
E	Modulus of elasticity
F	Force
F_a	Acceleration resistance, axial force
F_B	Braking force
F_H	Slope negative lift force
F_L	Air resistance, bearing force
F_n	Normal force
F_Q	Transversal force
F_R	Wheel resistance
F_r	Radial force
F_S	Lateral force
F_{St}	Gradient resistance
F_t	Tangential force
F_U	Circumferential force
F_Z	Traction
$F(t)$	Distribution function, failure probability
G_R	Wheel load
J	Mass moment of inertia
K_G	Gear characteristic value
L	Service life
M_b	Bending moment
M_t	Torsional moment
M_v	Reference moment
N	Fracture cycles
P	Power
P_A	Friction power related to area (synchroniser)
P_e	Effective power at engine output
P_m	Average friction work during synchroniser slipping times
$P_{Z,B}$	Demand power at wheel
Q	Lateral force, volume flow
R	Reaction force
R_a	Average peak-to-valley height

R_e	Yield point
R_m	Tensile strength
$R(t)$	Survivability, reliability
S	Safety factor, locking safety with synchronisers, slip, interlock value
S_B	Brake slip
S_H	Rear-mounted splitter unit high
S_L	Rear-mounted splitter unit low
S_T	Drive slip
T	Torque, characteristic service life
T_B	Acceleration torque (synchroniser), locking torque (differential)
T_L	Load torque
T_R	Friction torque (clutch, synchroniser), reactor torque (torque converter)
TC	Torque converter
TCC	Torque converter clutch
U	Revolutions
V_H	Total displacement
W	Moment of resistance, work, usable work, friction work
W_A	(Specific) friction work per unit area
W_b	Moment of resistance against deflection
W_t	Moment of resistance against torsion
a	Acceleration, axle base
b	Form parameter, failure gradient, overall length, width, fuel consumption
b_0	Size factor
b_e	Specific fuel consumption
b_S	Surface factor
b_s	Fuel consumption per unit of distance
c	Constant, rigidity, absolute speed
c_s	Tooth spring rigidity
c_u	Circumferential component of absolute speed
c_W	Drag coefficient
c_γ	Average value of tooth spring rigidity over time
d	Diameter
e	Eccentricity
f	Deflection
f_R	Coefficient of rolling resistance
$f(t)$	Density function
g	Acceleration due to gravity
h_i	Load cycle
i	Ratio
i_A	Power-train ratio (from engine to wheels)
i_E	Final ratio
i_G	Gear ratio
$i_{G, tot}$	Overall gear ratio, ratio spread
i_{CG}	Constant gear ratio
i_M	Centre gear ratio
i_N	Hub gear ratio
i_S	Moving-off element ratio
i_V	Variator ratio
j	Number of friction contacts
k	Wöhler curve equation exponent
$k(\nu)$	Characteristic value of a torque converter

m	Gear modulus, mass, linear scale (converter)
m_F	Vehicle mass
n	Rotational speed, quantity, stress reversals, number of bearings
n_M	Engine speed
p	Contact pressure, pressure, number of gear pairs
q	Gradient, surface load
q'	Gradient in %
r	Radius, degree of redundancy
r_{dyn}	Dynamic tyre radius
s	Travel, gearshift sleeve travel, fin pitch
s_{Fn}	Root thickness chord
t	Statistical variable, time
t_0	Time without failure
t_m	Mean of Weibull distribution
t_R	Slipping time, friction time
t_S	Shifting time
u	Gear ratio, circumferential speed
v	Speed, flow rate
v_F	Vehicle speed
v_{th}	Theoretical speed where $\lambda = 0$
v_W	Wind speed
w	Work input, relative wind speed
x, y, z	Co-ordinates
z	Number of speeds, number of teeth, number of load cycles
z_i	Number of teeth gear i

α	Meshing angle, taper angle of a taper synchroniser, viscosity pressure coefficient
α_{DK}	Throttle valve angle
α_K	Force meshing angle relative to tip edge
α_k	Statistical form factor
α_n	Normal meshing angle
α_{St}	Gradient angle
α_0	Strain ratio
β	Helix angle at pitch circle, aperture angle of claws
β_K	Dynamic beam stress rate
Δ	Interval, difference
ΔS	Wear path
ΔV	Wear
ε	Total contact
ε_α	Transverse contact ratio
ε_β	Overlap ratio
η	Efficiency, dynamic viscosity
ϑ	Temperature
λ	Performance coefficient (converter, retarder), drive slip, rotational inertia coefficient
$\lambda(t)$	Failure rate
μ	Torque conversion, coefficient of friction
μ_0	Stall torque ratio
μ_G	Coefficient of sliding friction
μ_H	Coefficient of bonding friction

ν	Speed ratio, kinematic viscosity
ρ	Density, angle of friction of the claws
σ	Direct stress
σ_b	Bending stress
σ_D	Fatigue strength
σ_H	Hertzian stresses
σ_v	Reference stress
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ω	Angular velocity

Subscripts

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2	Wheel (= large gear), output
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OS	Output shaft
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PV	Pump test
Q	Transverse
R	Reverse gear, roll, slip, friction, wheel, range-change unit, reactor
Roll	Roll
S	Status, system, splitter unit
Sch	Pulsating (strength)
St	Gradient
T	Turbine wheel, drive
TC	Torque converter
U	Circumference
V	Front-mounted splitter unit, variator, loss, trial

W	Reversing (strength)
Z	Traction, tensile load, intermediate gear, tooth, opening

a	Acceleration, axial, values at tip circle, tip of gear, outlet, external
abs	Absolute
b	Bending
dyn	Dynamic
e	Effective, inlet
fric	Friction
front	Front
fuel	Fuel
rear	Rear
i	Internal, control variable $i = 1, 2, 3, \ldots n$
id	Ideal
in	Input
i, j	At point i, j
j	Control variable
k	Control variable
kt	Beam stress
m	Mean, number of stress classes
main	Main
max	Maximum
min	Minimum
n	n-th speed, nominal, nominal operating point
oil	Oil
out	Output
perm	Permissible
r	Radial
red	Reduced
ref	Reference
rel	Relative
res	Resultant
spec	Specific
stat	Static
t	Torsion, time
th	Theoretical
tot	Total
twist	Twist
w	Reversing, pitch circle
x, y, z	In x, y, z direction, around x, y, z axis
z	Highest speed, number of speeds

1 Introduction

Every vehicle needs a transmission!

1.1 Preface

All forms of motorised transport, including vessels and aircraft, need transmissions to convert torque and rotation (Figure 1.1). There are distinctions between transmissions according to their function and intended use, for example selector gearboxes, steering boxes and power take-offs. This book deals only with road vehicle transmissions, or transmissions for vehicles for combined road and off-road use (outlined in bold in Figure 1.1).

Figure 1.2 gives an overview of transmission types in general current use. Further details are given in Chapter 6 "Vehicle Transmissions Systems: Basic Design Principles".

The function of a vehicle transmission is to adapt the traction available from the drive unit to suit the vehicle, the surface, the driver and the environment. The main parameters are technical and economic competitiveness. The transmission has a decisive effect on the reliability, fuel consumption, ease of use, road safety and transportation performance of passenger cars and commercial vehicles (Figure 1.3).

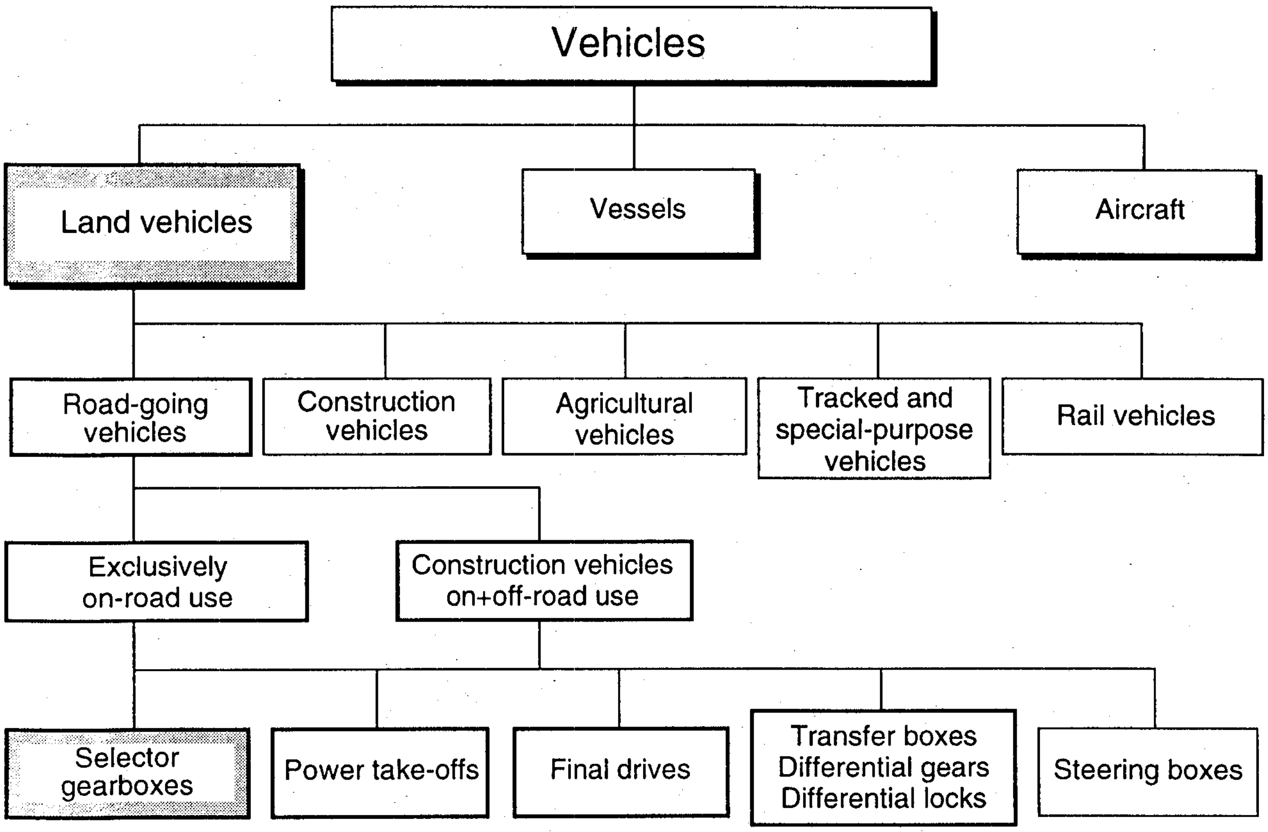


Figure 1.1. Definition of the term "automotive transmission" for the purposes of this book

<table>
<tr><th colspan="11">Transmission types</th></tr>
<tr><td colspan="6">z-speed-transmissions
(geared transmissions with z speeds)</td><td colspan="5">Continuously variable transmissions</td></tr>
<tr><td>Constant-mesh transmission</td><td>Synchromesh transmission</td><td>Semi-automatic constant-mesh or synchromesh transmission</td><td>Fully automatic constant-mesh or synchromesh transmission</td><td>Conventional automatic transm.</td><td>Twin-clutch transmission</td><td>Pulley transmission</td><td>Toroidal transmission (friction gear)</td><td>Hydrostatic transmission</td><td>Electric transmission</td><td></td></tr>
<tr><td colspan="3">With power interruption</td><td colspan="8">Without power interruption (powershift)</td></tr>
<tr><td colspan="3">Moving off and disengagement with foot-operated clutch</td><td colspan="8">Automatic continuously variable moving off</td></tr>
<tr><td colspan="2">Manual gearshift</td><td>Semi-automatic gearshift</td><td colspan="3">Automatic gearshift</td><td colspan="5">Automatic torque and speed conversion</td></tr>
</table>

Figure 1.2. Systematic classification of vehicle transmissions

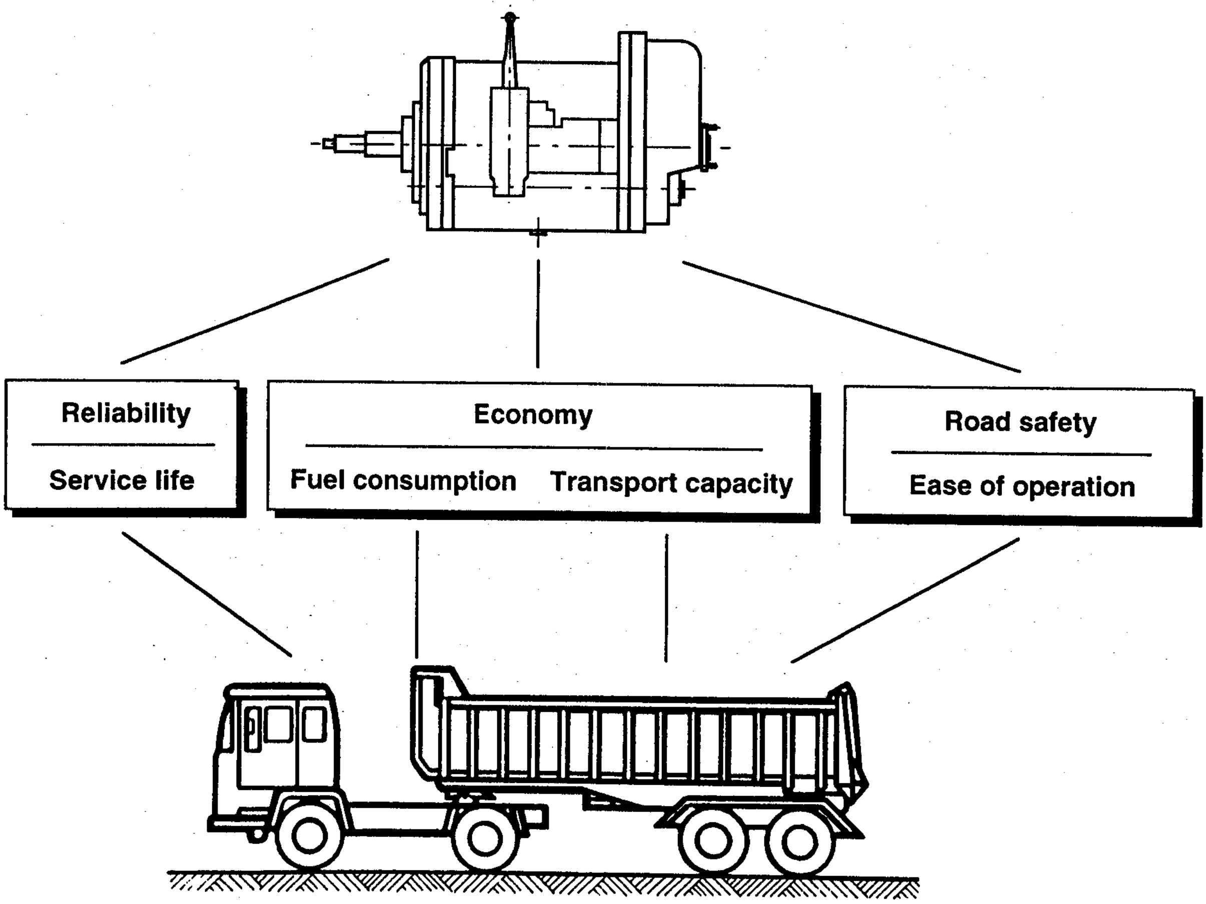


Figure 1.3. Effect of the transmission on key features of the vehicle

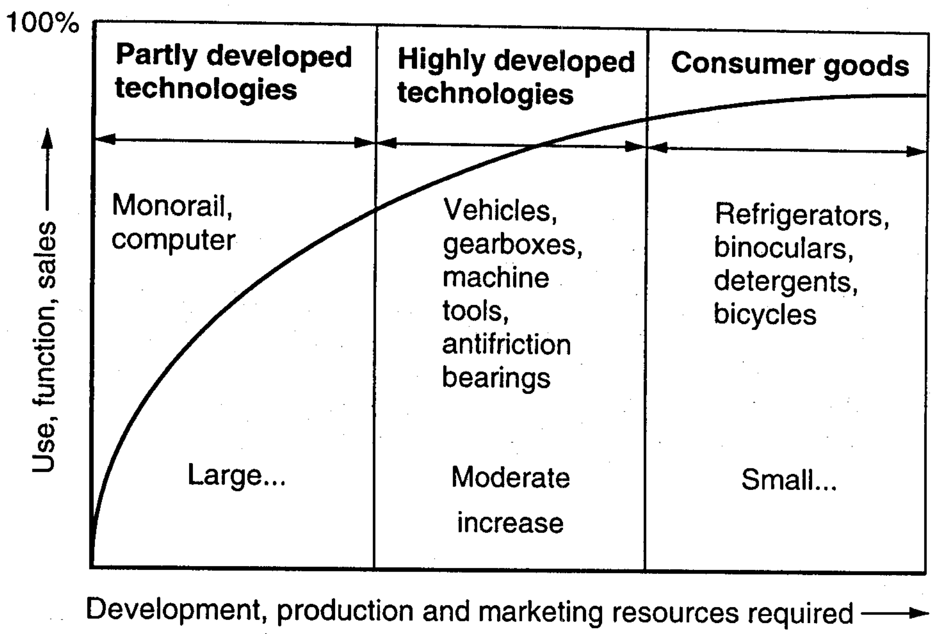
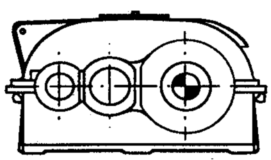
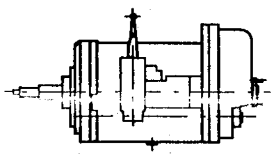


Figure 1.4. Achievable increase in utility of a product by additional development effort

Vehicle transmissions are technically and technologically highly mature mass-produced products. They are categorised as highly developed technologies (Figure 1.4). One notable feature is the specific power handling capacity P_{spec} in kW/kg of vehicle transmissions, which is more than twice that of industrial transmissions (Table 1.1), despite the fact that vehicle transmissions have more speeds. But industrial transmissions have to be designed for longer service life.

There are unlikely to be any further fundamental innovations in vehicle transmission technology. There is more likely to be a process of gradual evolution. The main trends are system thinking embracing the factors Environment ⇔ Traffic ⇔ Vehicle ⇔ Transmission, and greater use of electronics for control and monitoring processes. This defines the superordinate development goals for vehicle transmissions (Figure 1.5). Their development has to be fast and market-orientated. There has to be flexibility in adapting to customer preferences, especially in the case of commercial vehicles. Legal requirements also have to be taken into account, such as kW/t constraints and the maximum permissible noise level.

Table 1.1. Comparison of a vehicle transmission (commercial vehicle) with an industrial transmission

Transmission	Number of stages/ gears	Ratio i	Power P (kW)	Input torque T_1 (Nm)	Volume (m^3) Mass (kg)	Specific power P_{spec} (kW/kg)
Industrial use 	Two-stage 1 Gear	12.5	330	2100	0.2823 680	0.479 100%
Automotive use 	Two-stage or 3-stage 16 gears 1 reverse gear	13.8 in 1st gear	356	1700	0.159 335	1.06 221%

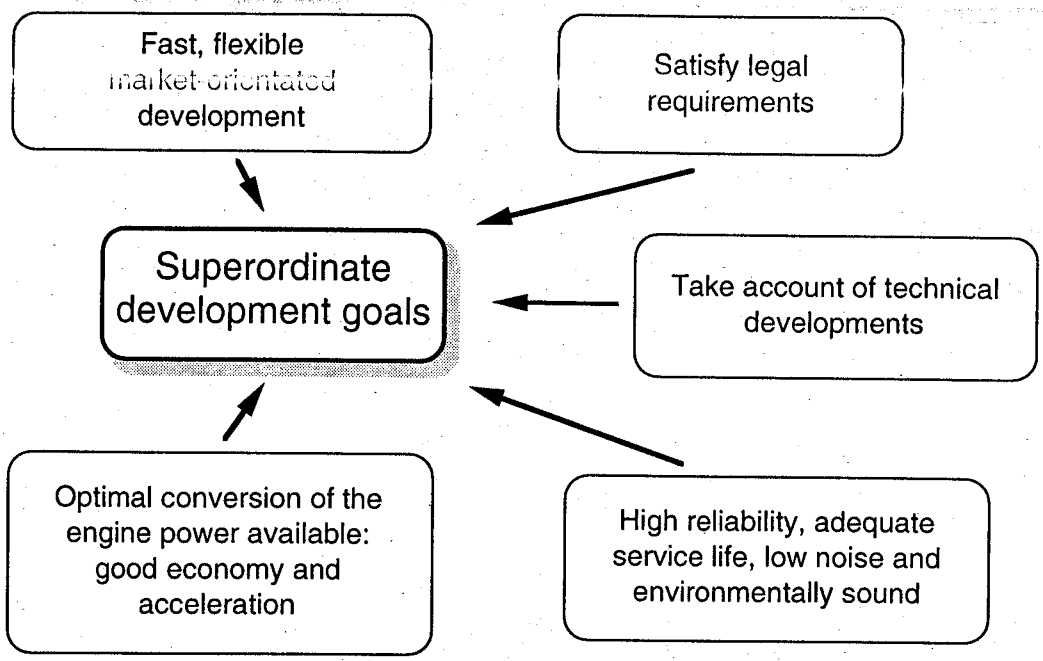


Figure 1.5. Superordinate development goals for vehicle transmissions

The main goal when developing a vehicle transmission is to convert the power from the engine into vehicle traction as efficiently as possible, over a wide range of road speeds. This has to be done ensuring a good compromise between the number of speeds, climbing performance, acceleration and fuel consumption of the vehicle. Technical and technological advances have to be taken into account, as do operational reliability and adequate service life. It is also essential to have regard for environmental and social considerations.

Vehicle transmissions always have to be developed within the framework of planning horizons for new vehicles (Figure 1.6). Transmissions have to be developed or adapted in parallel with the development phase of a vehicle. This also involves preparing and introducing new production technologies for mass production.

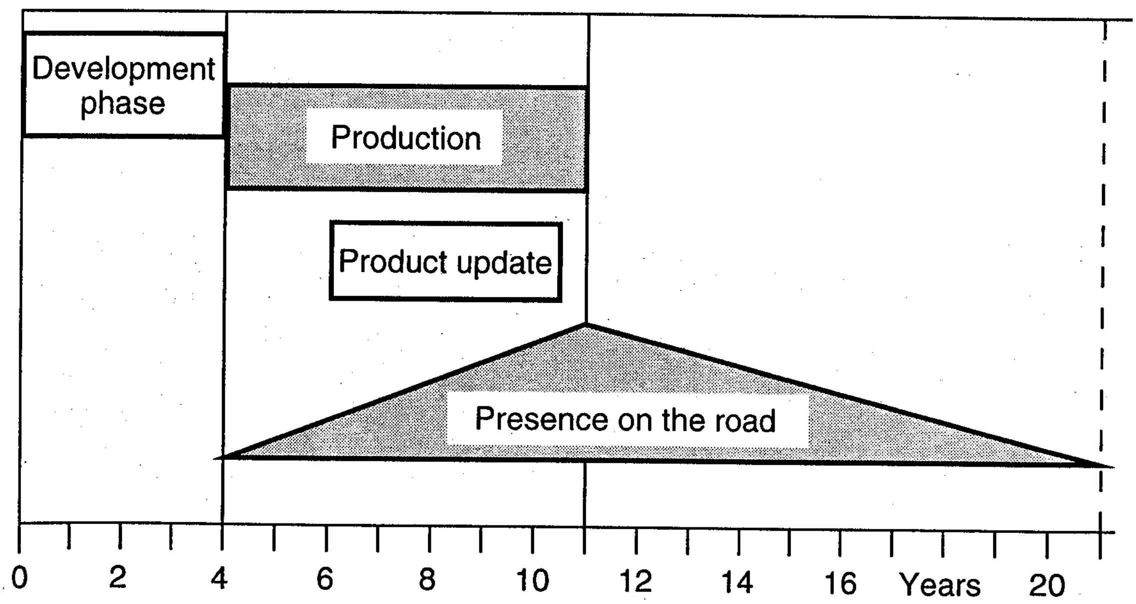


Figure 1.6. Time dimensions and planning horizons in the automotive industry as per [1.1]

This book sets out to present the development process for vehicle transmissions in its totality (Figure 1.7). The intention is to put the development of vehicle transmissions into its broader context. It is always necessary to determine the overall system within which the product is to be used, apart from just the product. A system overview is essential, and is presented in Chapter 2.

The critical constraints for the vehicle transmission designer are the vehicle, the engine and the operating environment. Basic knowledge of these factors is essential for meaningful development. Chapter 3 highlights the interaction between the power required and the power available. The first specific development task involving the gearbox is then selecting the range of ratios, or "overall gear ratio", to be covered. The functioning of vehicle and transmission as a system can then be assessed, and decisions made as to the number of speeds z, the ratio of the individual speeds, and the resultant gear stages. Taking into account the operating environment, the designer then has to decide whether the vehicle has adequate acceleration, the required climbing performance, and the top speed v_{max} stipulated in the specification. This also determines whether the transmission enables efficient motoring, especially in terms of fuel consumption. This issue is examined in detail in Chapters 4 and 5.

Creative design remains essential, and is now supported by systematic engineering design. This involves carrying out a functional analysis at the concept stage. Solutions must be found and assessed for the various individual functions, and then combined to form an overall solution. The necessary knowledge of vehicle transmission systems is given in Chapter 6.

Chapters 7 to 11 examine the design and construction of the main components of a transmission: gearwheels, shafts, bearings, synchronisers, and hydrodynamic clutches and torque converters. The sophisticated techniques now in use, such as the finite element method (FEM) and gearwheel calculation to German standard DIN 3990, are not examined in detail. An attempt is made to describe the fundamentals of design methodology and engineering processes.

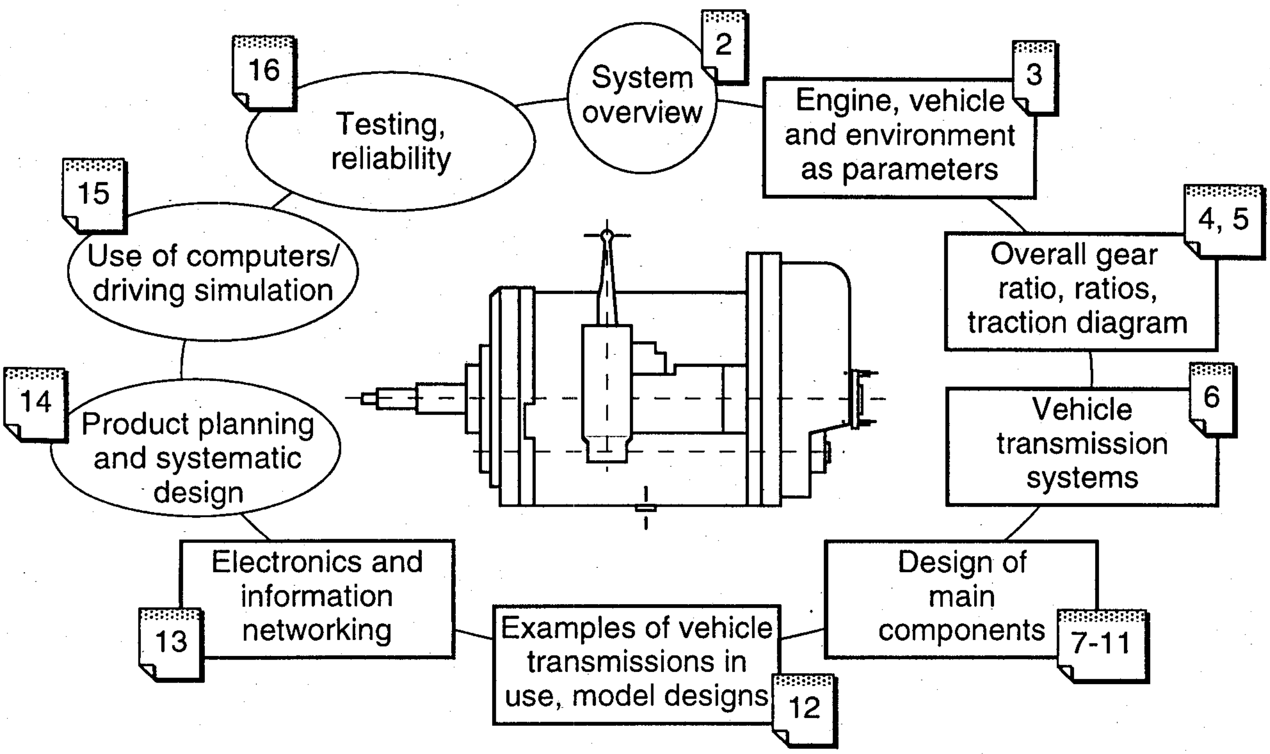


Figure 1.7. Map of the processes involved in developing vehicle transmissions, overview of chapters

Chapter 12 examines the construction of various types of transmission, with a detailed review of numerous existing designs and key design elements.

No treatment of the subject would be complete without considering the increasing use of electronics. Developments are leading to integrated engine/transmission management (Chapter 13). Electronics is also being applied to assist the shifting process in conventional manual transmissions and in automatic transmissions.

Important development tools for designing vehicle transmissions are dealt with in the latter part of this book. Product planning, project planning and systematic engineering are examined in Chapter 14. Chapter 15 considers the use of computer-aided design (CAD) and driving simulation for optimising transmission design.

Product reliability is becoming increasingly important. The customer is principally concerned with the reliability and service life of the system as a whole. Extensive testing on suitable test rigs is essential for determining service life and reliability. This topic is dealt with in Chapter 16.

A special attempt has been made in this book to show the user how to proceed, and to provide comprehensive data for practical transmission development work. As DUDECK has stated, *"One of the tasks of engineering science is to refine complicated models to the point of simplicity"*. This book strives towards that aim.

1.2 History of Vehicle Transmissions

Knowledge of the past and of the state of the Earth
affords the human spirit delight and sustenance
/LEONARDO DA VINCI/

Applying the lessons of the past! Development engineers and designers should have a grasp of the historical development of their products. They can then assess what yet remains to be achieved, and the level of technology represented by current product development. Such knowledge complements systematic design (see Chapter 14).

1.2.1 Fundamental Innovations

Fundamental innovations are discoveries, inventions and new developments without which the existing product could not have been developed. These seminal innovations inform subsequent discoveries, inventions, new developments and designs, leading to the creation of new products (Figure 1.8).

The aim of such development processes is to investigate and research certain phenomena to ensure reliable operation of the product. Table 1.2 is an attempt to trace the development of basic mechanical engineering innovations which lead to motor vehicles and thus to vehicle transmissions.

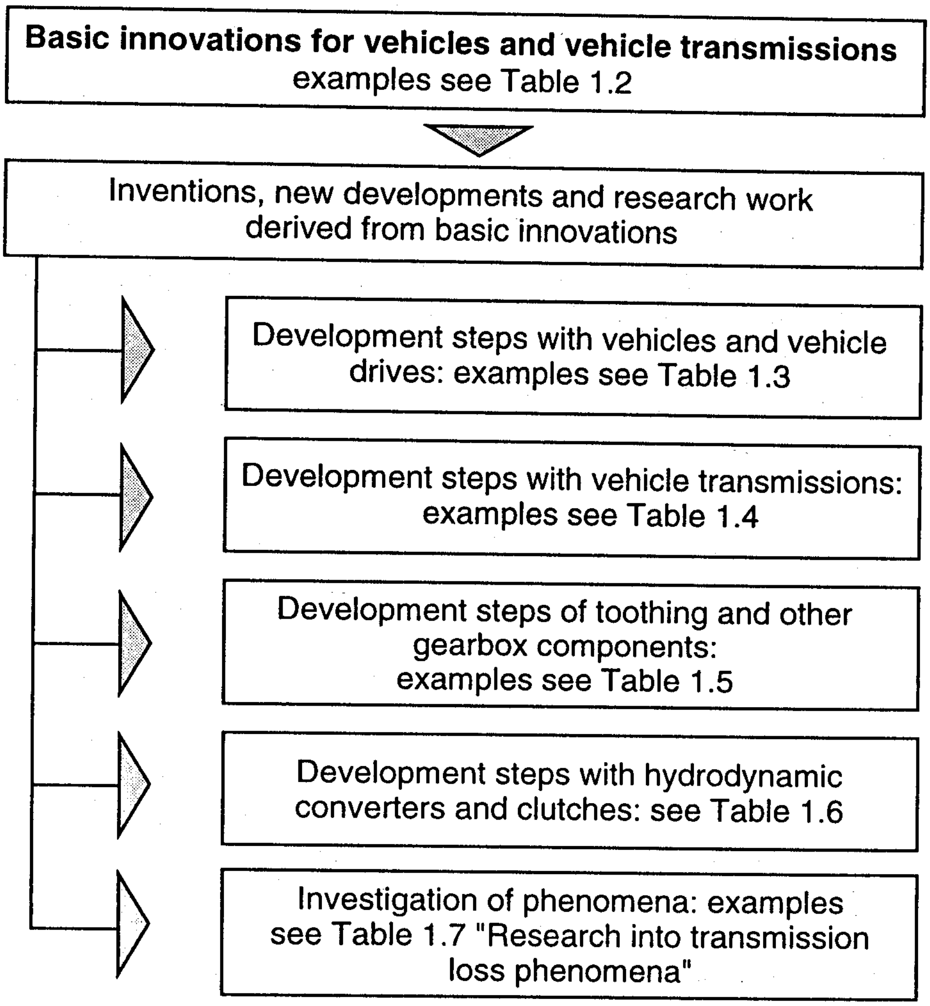


Figure 1.8. Products are developed on the basis of fundamental innovations!

Table 1.2. Examples of seminal innovations for vehicles and vehicle transmissions

4000 BC	Mesopotamian vase with a picture of a cart
2500 BC	Wheels made of two semicircular wooden discs, presumably with leather tyres
2000-1000 BC	Spur gears with pin wheel gear as drive element for water scoops (Sakie, Figure 1.10), worm gears for cotton gins
500 BC	Greek scholars discover the principles of mechanics
200 BC	Lever, crank, roller, wheel, hoist, worm gear and gearwheel are in use
1754	*Euler's* law of gears for gearwheels, involute toothing
1769	*Watt* Patent for steam engine
1784	*Watt* Gearbox with constant-mesh engagement
1829	*Stephenson* Rail vehicle, steam locomotive
1877	*Otto* Patent for 4-stroke gas engine with compression
1885	*Benz* Three-wheeler with internal combustion engine
1897	*Bosch* Magneto electric ignition
1905	*Föttinger* Hydrodynamic torque converter
1907	*Ford* Mass production of model T; the passenger car becomes a mass-produced item
1923	*Bosch* Injection pump
1925	*Rieseler* Automatic passenger car transmission with torque converter and planetary gear-set

1.2.2 Development of Vehicles and Drive Units

The idea of equipping the engine with a gear unit for adapting the engine's speed and torque to the power output required dates from 100 years before the official date of birth of the automobile in 1886. Another problem in the early years of the engine was the need to convert the reciprocating movement of the pistons into rotary movement. One solution is shown in Figure 1.9. The historical development of the transmission is thus closely linked to that of all engines.

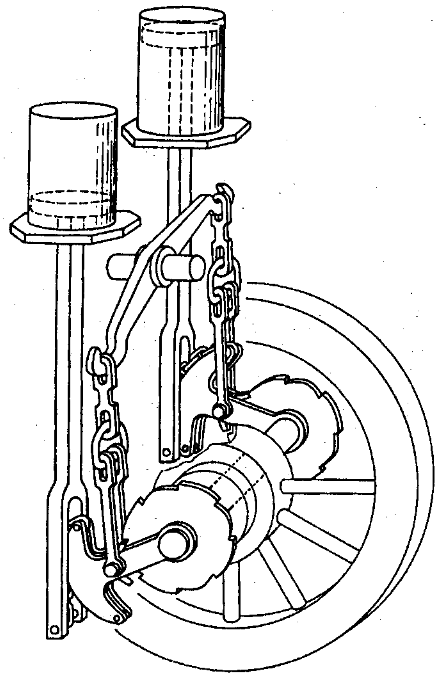


Figure 1.9. Conversion of reciprocating movement into rotary movement. Twin-cylinder power unit with opposed pistons in steam passenger car (CUGNOT, 1725 to 1804)

Table 1.3. Chronological development of vehicles and drive units

5000–500 BC	First technical inventions known: wheel, cart, gearwheel
1500	*Dürer* Sketch of a self-propelled vehicle
1690	*Papin* Designs an atmospheric steam engine with cylinder and pistons
1769	*Cugnot* Steam vehicle with rectifier transmission
1784	*Watt* Double-acting steam engine with rotary movement and flywheel
1800	*Trevithick* Patent for high-pressure steam engine
1801	*Trevithick* Use of steam vehicle to carry passengers
1801	*Artamonow* Metal bicycle with pedal cranks
1814	*Stephenson* First steam locomotive
1817	*Drais* Steerable road wheel
1832	*Pixii* Rotating alternating current generator
1845	*Thompson* Invention of the pneumatic tyre
1862	*Lenoir* Double-acting gas piston engine
1866	*Siemens* Discovery of the dynamo-electric principle and design of an operational dynamo
1877	*Otto* Patent for four-stroke gas engine with compression
1884	*Parsons* Patent for steam turbine
1885	*Benz* Three-wheeler with combustion engine
1885	*Daimler* Motorcycle
1886	*Daimler/Maybach* Four-wheel motor car
1888	*Dunlop* Pneumatic rubber tyre
1889	*Maybach-Daimler* Steel wheeled passenger car with open 2-speed transmission
1897	*Bosch* Controlled electric magneto ignition

1897 *Diesel* Diesel engine; heavy fuel engine with compression ignition
1903 *Wright brothers* Powered flight
1907 *Ford* Introduction of mass production line
1926 *Gregoire* Constant-velocity joint. The Tracta joint opens the door to mass-produced front-wheel drive
1934 *Porsche* Project draft of the Volkswagen
1970 *Thyssen Henschel* Transrapid maglev monorail
1980 *France* TGV high-speed trains
1990 *Bundesbahn* ICE high-speed trains

1.2.3 Stages in the Development of Vehicle Transmissions

Gear systems were undoubtedly used more than 1000 years ago to enhance the effectiveness of human and animal effort. Like the bullock gear systems, that are still in use in Egypt today, the two mating parts interlock by means of wooden pins or teeth (Figure 1.10).

The first drawings of gear systems are from the Middle Ages. Muscle power was used in the absence of mechanical power. The human "machines" had to do the hard work. This was the origin of the first "vehicle transmissions". In Albrecht Dürer's etching of a "muscle-powered vehicle" around 1500, the limited human power stroke is converted into propulsive force by means of a thrust crank, a bevel gear and a spur gear stage.

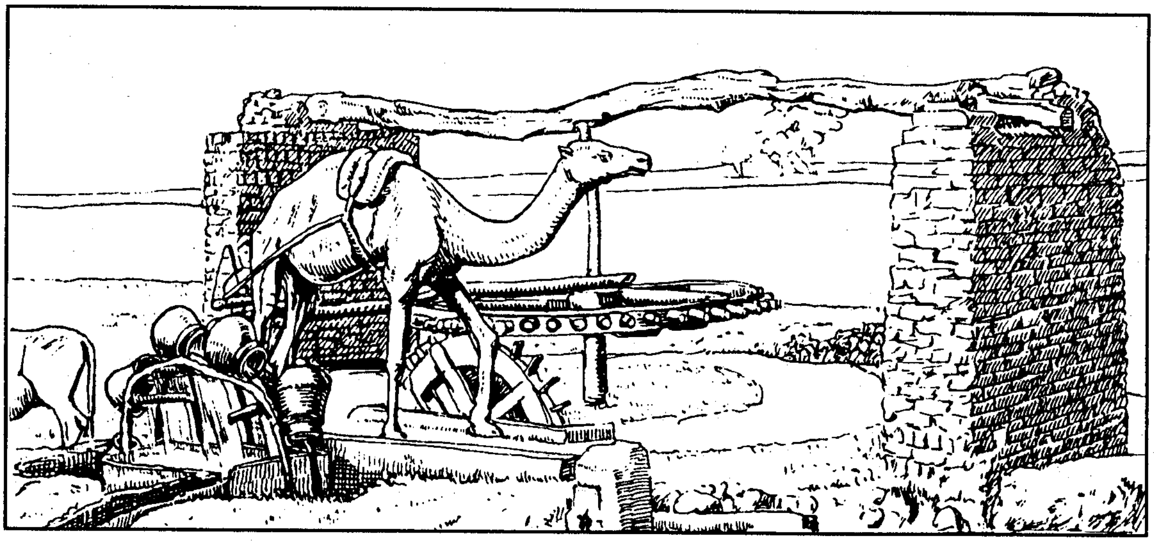


Figure 1.10. An early gear system! Egyptian water scoop (Sakie) in Luxor, approximately 2000 to 1000 BC

Table 1.4 gives examples of the important stages in the development of vehicle transmissions. It shows that all the main elements and design principles for vehicle transmissions had been developed by 1925. Developments since that date have been aimed at improving service life and performance, and/or reducing weight and noise, and improving ease of use. Four lines of development can be distinguished (Figure 1.11):

- mechanical z-speed geared transmission,
- semi-automatic or fully automatic z-speed geared transmission,
- conventional hydrodynamic/mechanical geared automatic transmission,
- mechanical, hydrodynamic, hydrostatic or electric continuously variable transmission.

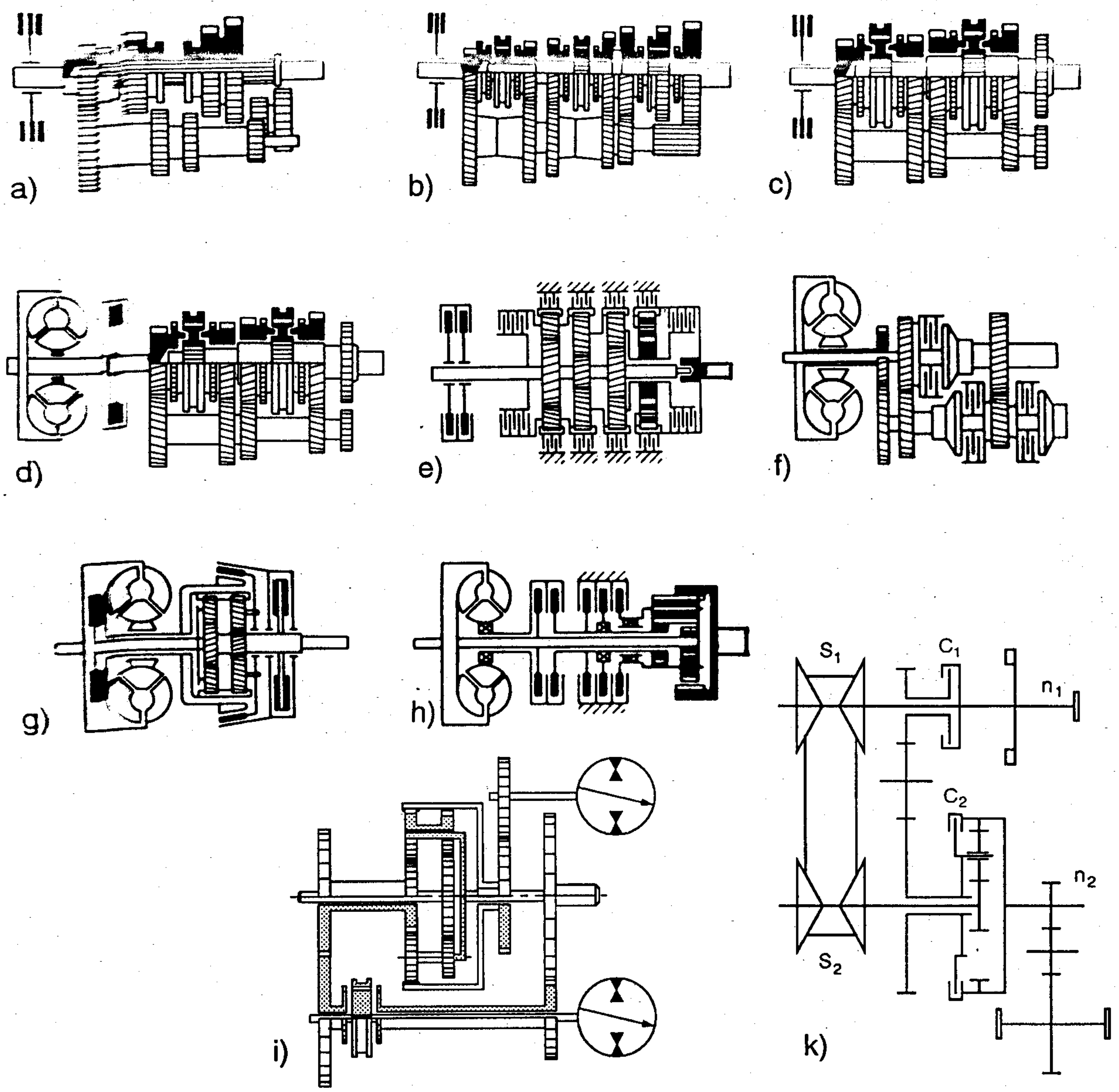


Figure 1.11. Development sequence of passenger car and commercial vehicle transmissions. **a)** Transmission with sliding gear engagement; **b)** Transmission with constant-mesh engagement; **c)** Synchromesh gearbox; **d)** Torque converter clutch gearbox, semi-automatic: converter + gearshifting clutch + synchromesh gearbox; **e)** Transmission with multiplate clutch shift; **f)** Transmission with converter and rear-mounted powershiftable countershaft transmission; **g)** Hydro-planetary transmission; **h)** Conventional fully automatic transmission; **i)** Hydrostatic continuously variable transmission with power split, fully automatic; **k)** Mechanical continuously variable pulley transmission

Table 1.4. Examples of important development steps in vehicle transmissions

1784	*Watt* stipulates that steam engines require additional ratios for road-going vehicles. *Watt* patents variable-speed gearbox with dog clutch engagement and constant mesh of gearwheels (Figure 1.12)
1821	*Griffith* 2-speed transmission with sliding gears (Figure 1.12)
1827	*Pequeur* First differential in a road-going vehicle (Figure 1.12)
1834	*Bodmer* Planetary transmission with stallable ring gear body using brake band

1849 *Napier/Anderson* 2-speed belt transmission (Figure 1.12)

1879 *Selden* Patent enclosed sliding gear transmission with reverse gear and clutch (Figure 1.12)

1885 *Marcus* Cone clutch for motor vehicles

1886 *Benz* Belt-driven bevel gear differential (Figure 1.12)

1889 *Maybach-Daimler* 4-speed transmission with sliding gears (Figure 1.13)

1890 *Peugeot* Complete power train with sliding gear drive (Figure 1.13)

1899 *Buchet* Continuously variable belt transmission with axially adjustable taper discs

1899 *Krauser/Schmidt* Continuously variable friction gear with taper discs

1899 *Darracq - Léon - Bollée* 5-stage variable-speed belt "transmission gearbox"

1899 *Oliverson - Killingsbeck* Continuously variable belt transmission with axially adjustable taper discs

1900 *Reeves - Pulley* Continuously variable belt expanding pulley transmission with thrust links and axially adjustable taper discs

1900 *Léo* 3-speed transmission with face dog clutch engagement, integral differential and chain drive reverse gear

1900 *Lang* 3-speed geared transmission with constant-mesh wheels and draw key shifting

1900 *Diamant Speed Gear Company* Helical gear transmission

1905 *Pittler* Hydraulic drive system with hydro pump and hydro motor

1906 *Renault* Pneumatic transmission with piston compressor and piston engine

1906 *Didier* Two-stage planetary gear transmission with shifting using brake band and clutch of the planetary gear via friction plate face clutch

1907 *Renault* Hydrostatic transmission with axial piston pump and axial piston motor

1907 *Ford* Mass production of the model T with 2-speed planetary gear

1915 *ZF-Soden transmission* 4-speed all constant-mesh transmission with constant-mesh gearwheels with preselector shifting and with synchronising aids

1925 *ZF* Commercial vehicle standard gearbox with spur toothed sliding gears

1925 *Rieseler* Automatic passenger car transmission with torque converter and planetary gear-set

1926 *Cotal* 3-speed planetary gear with automatic shifting via 3 electromagnetic clutches

1928 Development of the TRILOK converter – a precondition for modern hydro-mechanical "conventional" automatic transmissions

1928 *Maybach* Overdrive auxiliary gearbox for reducing engine speed; shifting by means of override face clutches, and ground helical cut gearwheels to reduce noise

1929 *ZF Aphon transmission* Helical cut 4-speed transmission with multiplate synchromesh

1931 *DKW F1* with driven front wheels. Transverse-mounted 2-cylinder 2-stroke engine

1932 *Wilson transmission* Multistage planetary coupling gear with identical ring gears that are alternately fixed against the housing by means of brake bands

1934 *ZF All-synchromesh gearbox* 4-speed gearbox, helical cut, all speeds synchronised

1939 *General Motors Hydra-Matic transmission* First mass-produced conventional automatic transmission: 13 million produced; hydrodynamic clutch, 4-speed planetary transmission, 2 belt brakes, 2 multiplate clutches

1939 *ZF* 4-speed transmission, helical cut, gearshift mechanism with electro-magnetic multiplate clutches

1948 *General Motors Dynaflow*-transmission with polyphase converter and 2-speed Ravigneaux planetary gear-set

1950 *Packard Ultramatic transmission* Conventional automatic transmission with torque converter lock-up clutch, 2-stage 2-phase converter and 2-speed planetary gear

1950 *Van Doorne "Variomatic"* Mass production of continuously variable V-belt transmission with axially adjustable taper discs (diameter adjustment)

1952 *Borg-Warner "Warner Gear" transmission* Conventional automatic transmission with TRILOK converter and 3-speed planetary gear-set

1953 *Borgward* Automatic transmission with converter and 3-speed spur gear drive with electro-hydraulic shifting

1953 *ZF* Hydromedia transmission for buses; 3-speed transmission with converter and hydraulically activated multiplate clutches

1958 *Smith* Magnetic-particle double clutch with rear-mounted 3-speed spur gear stage transmission and electrically activated dog clutches

1961 *ZF* 3-speed automatic transmission for passenger cars; converter without lock-up clutch, 3-stage planetary gear-set and hydraulic control

1961 *Daimler-Benz* 4-speed automatic transmission, of 2-range design with hydrodynamic clutch

1962 *ZF* 6-speed transmission series for commercial vehicles; dog clutch engagement or synchronised; optional 12-speed version with front-mounted splitter unit (2-unit design)

1962 *Eaton* 9-speed commercial vehicle transmission with power split to 2 countershafts for a shorter overall design length

1962 Commercial vehicle range change type transmission designs with 9 and more gears, especially with a rear-mounted range unit of planetary design start to become established

1967 *VW* Semi-automatic transmission with torque converter clutch and rear-mounted 3-speed geared transmission

1970 Various companies develop a torque converter clutch transmission for commercial vehicles with a torque converter lockup clutch and secondary 6–8 speed geared transmission

1971 *Sundstrand "Responder"* Mass produced hydrostatic commercial vehicle gearbox with power split through planetary gear-set

1972 *Turner* Commercial vehicle transmission with output constant gear and synchromesh on the countershaft to increase service life

1975 *Van Doorne* Continuously variable transmission for passenger cars with steel thrust chain and axially adjustable taper discs

1976 *ZF* 16-speed commercial vehicle transmission with integral front-mounted splitter and rear-mounted range unit

1978 5-speed passenger car gearboxes with increased overall gear ratio to reduce fuel consumption become established

1980 Converter with lock-up clutch in automatic passenger car transmissions

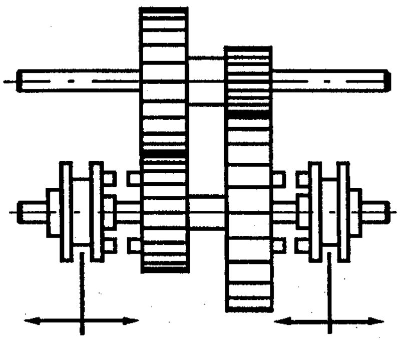
1983 *Eaton/Fuller Twin Splitter* 12-speed commercial vehicle transmission with 4-speed main gearbox and 2 rear-mounted splitter units

1985 *Porsche* Re-discovery of the twin clutch principle as an automatic transmission for passenger cars

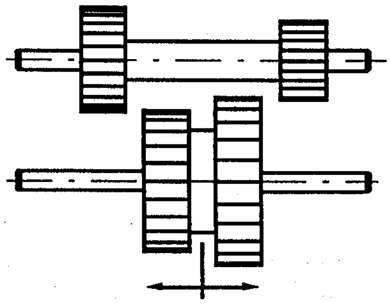
1990 Mass production of conventional automatic transmissions with torque converter, lock-up clutch, five speeds and electro-hydraulic shift

1990 *Voith* Continuously variable hydrostatic power split transmission for buses. Possibility of braking energy recuperation with energy accumulator

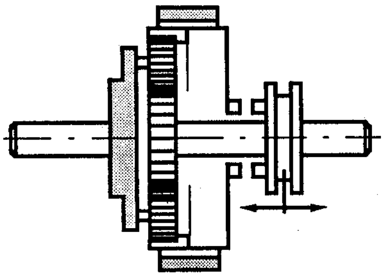
1991 Renewed interest in alternative power-train concepts: electrical and hybrid drives



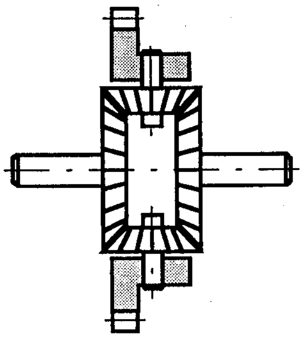
1784 *Watt* patent
2-speed gearbox with dog clutch engagement



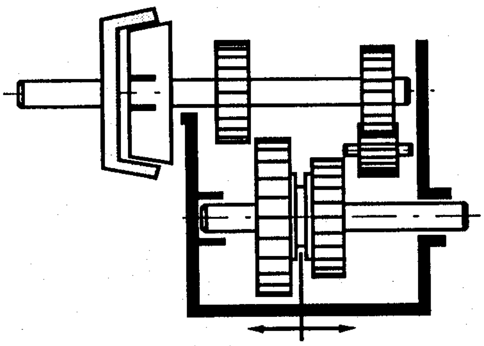
1821 *Griffith*
2-speed gearbox with sliding gears



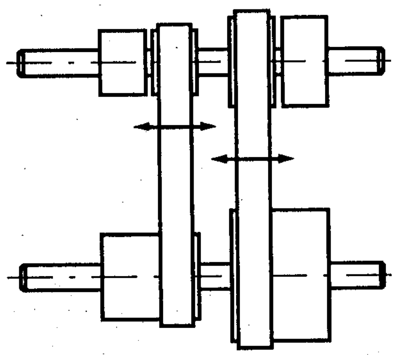
1834 *Bodmer*
Shiftable planetary gear



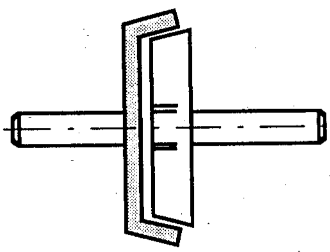
1827 *Pecqueur*
Differential gear



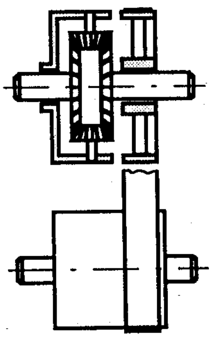
1879 *Selden*
Complete vehicle transmission with clutch, R gear and housing



1849 *Anderson*
Shiftable belt transmission



Around 1885 *Marcus*
Engaging cone clutch



1886 *Benz*
Belt-driven bevel gear differential

Figure 1.12. Early vehicle gear components and mechanisms

With the development of the steam engine, the need arose to adapt the engine power available to the intended use. The first steam-powered vehicles were driven by ratchet gears (Figure 1.9). Higher ratios were required to climb gradients than to drive on the flat. In 1784 JAMES WATT patented the constant-mesh gear with constantly meshing gearwheels (Figure 1.12), which is still in common use today. The variable-speed transmission was born. Production of road vehicles only really started several decades later. The steam vehicle builders EVANS and TREVITHICK, 1801, solved the problem of torque adaptation, but still by interchanging a gear pair.

There were a number of important inventions as early as the beginning of the 19th century (Figure 1.12). In 1821, GRIFFITH disclosed the sliding gear transmission system, which was extensively used as an inexpensive solution into the 20th century. In 1827, PECQUEUR succeeded in equalising wheel speeds when cornering, by means of a differential. In 1834, BODMER designed a partial power-shift planetary transmission. The change in gear ratio is achieved by disengaging the shifting dogs and tightening a brake band. In 1879, SELDEN patented a sliding gear drive with clutch and reverse gear as part of an overall patent for a piston engine vehicle.

It is striking that around the turn of the century there was already intensive effort devoted to the continuously variable transmission, which is ideally suited to the internal combustion engine. This involved considering not only mechanical solutions, but also hydrostatic and even pneumatic solutions (Table 1.4). But they did not gain acceptance, because of their low power rating or mechanical complexity. The hydrodynamic Föttinger torque converter (Table 1.6), invented in 1905 for ship propulsion systems, was not applied to vehicle power trains until 1925.

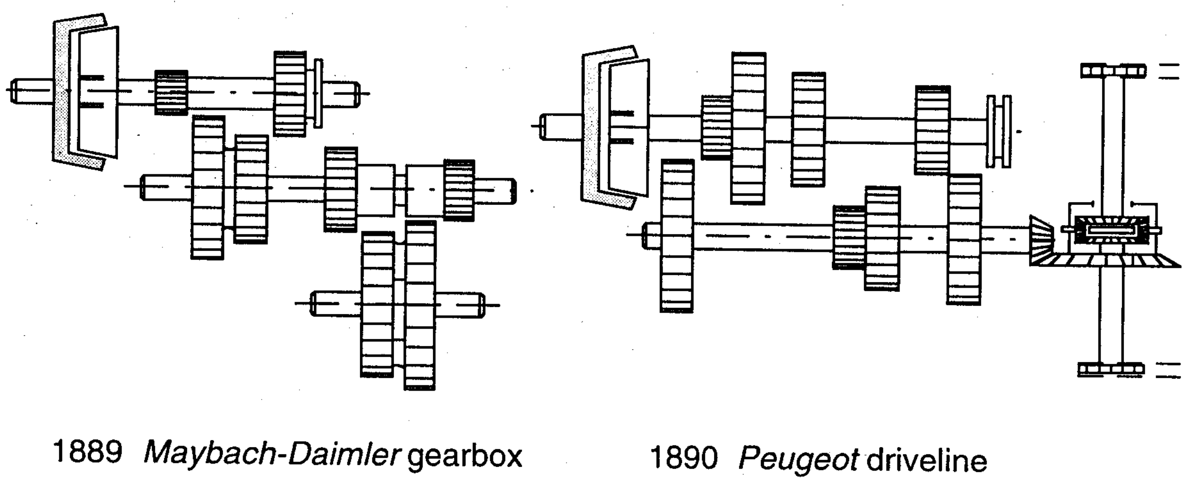


Figure 1.13. Early vehicle transmissions

Direct drive was another important development, with which BENZ created the classic countershaft transmission with coaxial input and output, which remains valid to this day. It is not yet incorporated in the exemplary 1890 Peugeot power train (Figure 1.13). This design of countershaft transmission with direct drive and four forward speeds proved effective in practice. The basic problems of ratio changing were solved.

Another phase of development started around 1920. In an effort to improve comfort and ease of use, development effort focused on ground and/or helical-cut spur gears, or reducing engine speed to reduce noise and make changing gear easier. Another important breakthrough was the standard gearbox (i.e. gearboxes which are structurally identical or which vary only in their ratios and connections) to facilitate efficient, cost-effective production (Table 1.4).

The first gearshifting aids date from the year 1915. The ZF Soden transmission had constant-mesh gearwheels, preselector and synchronising mechanisms. This transmission provided preselection, whereby the driver set a knob on the steering wheel to the required gear and pressed on the pedal. The clutch disengages. When the shift pedal is released, the pre-selected gear engages automatically. The advantage of almost effortless shifting could not make up for the disadvantages, such as the difficulty of adjusting the cable controls, and the complex gearbox design.

In a General Motors transmission, the shifting action and power transmission was effected by means of dogs with a taper synchroniser. In 1928, MAYBACH succeeded in substantially reducing vehicle noise with his auxiliary overdrive and helical-ground gears, by reducing gear hobbing faults and substantially improving the engine speed. At the same time the quiet-running ZF Aphon gearbox was produced, with three gears synchronised with plates. In the ZF fully synchromesh gearbox (1934), all the forward gears already had taper synchronisers.

The last striking changes in format without any change in design of mechanical passenger car transmissions occurred after the Second World War, when rear-wheel drive and then front-wheel drive with transverse engine became more prevalent on the market, a development which has now penetrated as far as upper mid-size vehicles. The direct gear and coaxial design were abandoned, and the engine, transmission and differential were combined in one unit to save space. From around 1978, 5-speed geared transmissions with an increased range of ratios and finer ratio stepping became increasingly common.

Gear shifting aids, leading up to automatic systems, represent an independent line of development. From around 1956, Fichtel & Sachs supplied DKW (now Audi) with an electrically controlled semi-automatic clutch, the SAXOMAT. The system consisted of a centrifugal master clutch and a vacuum-operated gearshifting clutch. When the gearshift lever is touched, a vacuum-controlled servo device opens the gearshifting clutch. When the gearshift lever is released, air is slowly released to the servomechanism through a nozzle, thus engaging it. Pressing the accelerator pedal increases the flow of air, and thus the engaging action. This represented a considerable advance in ease of operation compared to vehicles with a foot-operated clutch. In 1967 VW presented a semi-automatic 3-speed torque converter clutch transmission for passenger cars.

H. RIESELER designed an automatic transmission as early as 1925, consisting of a torque converter and rear-mounted planetary geared transmission, whose main components (torque converter with planetary gear shifted by means of clutches and brakes) are now typical for all automatic transmissions. Rieseler thus made an outstanding contribution, the advantages of which were not yet recognised by subsequent designers. They continued to merely replace the mechanical clutches with a fluid clutch. Conventional automatic transmissions began to establish themselves from 1939, comprising a torque converter (some with a clutch), three- or four-stage planetary gear-set and hydraulic control. The first mass-produced transmission of this type was the General Motors *Hydramatic*. These transmissions spread rapidly in the USA after the Second World War. They achieved a market share of around 85%. In Europe, conventional automatic transmissions still only reach a market share of around 15% even today. In 1953, Borgward developed the first automatic transmission design in Germany. It had a power-shift countershaft transmission with a front-mounted torque converter used only for moving off. After building under licence for some time, Daimler-Benz and ZF launched their own designs in 1961. Daimler-Benz still had an old design similar to the Hydramatic transmission, with planetary gear transmission and front-mounted fluid clutch. These automatic transmissions underwent constant development to reduce fuel consumption. A torque converter lockup clutch and a fourth and fifth gear to increase the range and adaptation of ratios became standard. The introduction of electrohydraulic controls made shifting easier.

The continuously variable transmission reappeared 50 years after the first development. The van Doorne *Variomatic* was developed in 1950, and in 1958 became the first mass-produced continuously variable transmission. The power was transmitted by rubber V-belts and V-belt pulleys whose diameter could be varied by axial displacement; the *Variomatic* used centrifugal weights and a membrane acted on by vacuum. On the output side, the pressure is applied by a spring. There is no need for a differential in this design with two parallel mounted belts. The difference in rotational speed is compensated by belt slip. The rubber V-belts placed a limit on power. The permissible input torque was around 100 Nm. The transmission was therefore only suitable for small passenger cars. Van Doorne then invented the "steel V-belt". The thrust link chain consists of a steel belt made up of thin belts, onto which the thrust links are pushed, linked to the V-belt pulleys. This transmission, developed from 1970 onwards, was ready for use in 1975. It went into mass production around 1987. This continuously variable transmission with steel link chain is not yet fully developed. It is hoped to increase its capability to make it suitable for mid-range passenger cars. It is not yet clear whether the continuously variable transmission can really reduce fuel consumption enough to justify introducing it on a large scale. At all events a variable-capacity pump and axial adjustment of the taper disc are necessary to improve the lower level of efficiency of such transmissions compared to toothed gear transmissions.

The only difference between passenger car gearboxes and commercial vehicle gearboxes until the Second World War was their size. There was then a fundamental change. Payloads increased as tyres with greater load-bearing capacity were developed, the truck moved into long-distance as opposed to local haulage, and the motorway network was expanded, etc., all of which meant a greater range of ratios (i.e. greater overall gear ratio), and thus higher number of speeds. The development sequence shown in Figure 1.11 also applies to commercial vehicles.

The development goals for mechanical geared transmissions for commercial vehicles were firstly low weight (= larger payload), reduced noise and improved ease of use with the introduction of synchronisers. One particular requirement was long service life of up to 1 million km. Initially five to six speeds were adequate, although these were fitted with front-mounted splitter units to give finer grading of the overall gear ratio. The 6-speed gearbox became a 12-speed gearbox. The increase in specific power output (kW/t) with commercial vehicles gave rise to the requirement for an increased overall gear ratio. Transmissions with nine and more speeds were developed. To provide greater economy or better performance, transmissions with twelve to sixteen speeds became established in the early seventies for heavy trucks. These were range type gearboxes (see Chapter 6). A modern 12-speed gearbox of this type is shown in Figure 1.14.

Synchromesh did not establish itself in commercial vehicle gearboxes to the same extent as in passenger car gearboxes, because of service life problems. But especially in Europe more and more commercial vehicle transmissions were fitted with synchromesh. Other approaches to improving ease of operation were also investigated. The SYMO selector was developed by Faun and Siemens from 1954 onwards. In this engine-based synchromesh, the gear is engaged under electronic control at the precise point when the element to be engaged is synchronised. The accelerator is also controlled by the electronic system when shifting. In critical situations, such as downhill slopes or on hills, equalisation of rotational speed by the engine may not alone be sufficient, or may be impossible if the electronic system fails, creating a dangerous situation for the driver, vehicle and load. Since this situation could never be completely excluded, the system never went into mass production. An attempt was made around 1970 to make commercial vehicle transmissions semi-automatic by developing torque converter clutch transmissions. The combination of a torque converter with a conventional gearshifting clutch and a 6- to 8-speed transmission made moving off with heavy tractor-trailer units easier. The torque converter increases the overall gear ratio without requiring more than six to eight speeds. Although transmissions of this type are in use, they have not proved popular, achieving a market share of only 1–2%. The reasons are principally the system's complexity and consequently its price and the increase in fuel consumption.

Automatic transmissions have not yet become common in trucks because of the questions of economy and reliability. Where commercial vehicles are exported to developing countries, reliability and ease of maintenance are prime considerations. Automatic transmissions are however standard equipment for city buses (Figure 1.11h). In 1971 the first production version of a continuously variable hydrostatic power split transmission (via a planetary gear-set) for city delivery vehicles, the Sundstrand *Responder*, did not prove successful, and production was stopped. Continuously variable bus transmissions were under development (at Voith, Heidenheim) in the early 1990's, with hydrostatic units and mechanical power split via a planetary gear, and the possibility of regenerative breaking. But these transmissions did not go into mass production either. Especially for city buses, the main thrust of development is currently towards electrical drives combined with fuel cells.

1.2.4 Development of Gear-Tooth Systems and other Transmission Components

Vehicle transmission components are now themselves undergoing a process of evolution. We examine below the development of components such as gearwheels, shafts, bearings, synchronisers and clutches, as well as electronic controls (Table 1.5).

The most important component is the gearwheel. The historical origins of the gearwheel cannot be precisely determined. But gear drives were in use at an early stage for increasing human or animal power, and for exploiting wind and water power. It is reasonable to assume that the use of wooden gearwheels with crossed axes, like the bullock gear systems still in use for irrigation in Egypt today, is one of the earliest examples of the use of the gearwheel (Figure 1.10). Derivants of this design are mill drives and serial-mounted geared drives to achieve greater transmission ratios, recorded in a great variety of forms in contemporary drawings. The use of gears for power transmission has been particularly beneficial too in mining and mill construction. The great artist and inventor, LEONARDO DA VINCI, laid the foundations for modern machine elements as early as the 15th century.

The scientific study of gear systems started in the late 17th century with the work of DE LA HIRE, continued by EULER, WILLIS and REULEAUX. The law of gears finally formulated by SAALSCHÜTZ in 1870 states:

> *There will be uniformity of transmission of motion between two meshing gearwheels where the common normal of both tooth curves passes through the pitch point C at any contact point of the flanks.*

The precondition for gear hobbing by machine was the use of mathematical, graphical methods to create theoretically correct flank profiles. The development of the rolling process opened the way for industrial gearwheel production (Table 1.5).

Whereas previously pinwheel and cycloid gears were the most important types of gear, today it is the involute. It can be accurately manufactured and measured because of its straight flanked tool, which meshes on the base circle. It also has the characteristic of being insensitive to changes in gear centre distance.

New perspectives in gear tooth manufacture have opened up since 1980. With numerically controlled tooth hobbing machines, the rotary movements and longitudinal movements necessary to produce the tooth profile are electronically controlled and synchronised. This means that now any required tooth profile can be produced that satisfies special requirements (e.g. for low-noise gear pumps) yet also fulfils the requirements of the law of gears.

Initially heat treatable materials were used for gearwheels. Case-hardened materials soon became necessary to increase performance and minimise weight. To achieve the quality necessary for noise reduction, the gearwheels had to be shaved after hobbing, or ground after hardening.

Table 1.5. Chronological development of gear tooth systems and other transmission components

2000–1000 BC	Spur gears with pinwheel gear, worm gears. Transport of heavy loads on rollers
230 BC	*Philon of Alexandria* Multi-lever wheel with gear rack
100 BC	Sun wheels and planetary gears in the astrolabe of Anticythera
1300	*Giovanni da Dondi* Astronomic clock with internal gearing and elliptical gearwheels
15th C.	Idea of helical gears Sprocket wheels for link chains
15th C.	*Leonardo da Vinci* "Book of movement", "Book of gravity", "Book of moment of force", principle of virtual speeds, principle of independent superposition of movements, principle of the potential lever
15th C.	Gearwheels for transmitting movement in windmills
1639	*Désargues* Cycloid profiled gearwheels
1694	*De La Hire* Founder of gearing science, point gearing: teeth paired with points or journals, involute circles
1733	*Camus* Pair gearing, teeth paired with teeth, cycloid toothing
1754	*Euler* Involute toothing
1765	*Euler* Curvature centre-points
1780	*Wasborough/Pickard* Thrust crank transmission
1820	Axial ball-bearing with cage as bearing for castors
1820	*Tredgold* Validation of gearwheel strength calculation
1850	*Willis* Systematic classification of gears: Moduli: possibility of combining any gearwheels from the same modulus
1856	*Schiele* Hobbing process useable with insertion of index gears
1857	Application and spread of ball and roller bearings in bicycles, first patented cup-and-cone bearing
1865	*Reuleaux* Description of "general gear hobbing"
1869	*Surirey* Ball bearing
1872	*Wagen-Thorn* Cutter shaping method
1876	*Reuleaux* Line of action
1881	*Hertz* Theory of contact and pressure of solid elastic bodies; Hertzian stress
1882	*Bilgram* Invention of bevel gear production
1883	*Petroff - Tower - Reynolds* Hydrodynamic lubricant film theory in plain bearings
1885	*Marcus* Cone clutch for automobiles (Figure 1.12)
1887	*Grant* Cutter shaping method for helical gears
1890	*Sachs* Patent on precision bicycle wheel hub
1895	*Maybach* Gate shift for automotive transmissions, grouping speeds in "gates"
1897	*Pfauter* Universal gearwheel milling machine for spur gears, worm gears and helical gears
1902	*Stribeck* Work on the chief characteristics of plain bearings and roller bearings

1903 First deep groove ball bearing
1907 *SKF* Self-aligning ball bearing
1908 *Norma* First useable cylindrical roller bearing
1912 *Humphrie* Synchromesh to make changing gear easier
1915 *Maag* Gear grinder
1916 *v. Soden* Patent application for synchromesh
1925 *Gleason* Hypoid gear
1927 *ZF* Bevel grinding
1930 *Palmgren* Method for calculating anti-friction bearings based on the concept of service life
1934 Determination of module series
1938 *ZF* Introduction of lock synchroniser
1956 *Fichtel und Sachs Saxomat* Electrically controlled semi-automatic clutch comprising centrifugal master clutch and vacuum-activated gear-shifting clutch
1955 *Novikov* Round-flank toothing for unhardened spur gears
1983 Free tooth formation according to the law of gears using numerically controlled gear hobbing machines

Other important transmission components such as ball-and-roller bearings, clutches and synchronisers were then developed in the second half of the 19th century and the early 20th century.

It should finally be noted that as a means of converting torque and rotational speed, toothed gearing has a better power/weight ratio than other converters such as belt or chain drive, hydrodynamic or hydrostatic transmission or the electric motor.

1.2.5 Development of Torque Converters and Clutches

The individual components of the automatic transmission were initially slow to develop, but then proceeded very rapidly, considering the complexity involved.

The foundations were laid by H. FÖTTINGER when he applied for a patent for a torque converter in 1905, and some time later for a hydrodynamic clutch. FÖTTINGER had designed this torque converter for use in ships, and never considered its use in automobiles. The development of the torque converter is a good example of the systematic development of a transmission component (Table 1.6 and Chapter 10).

Table 1.6. Chronological development of torque converters and clutches, and their use in conventional automatic transmissions

1900 Steam turbines start to replace steam engines. Ship propulsion systems require a reversible reduction gearbox approx. 1:4 for several 1000 hp between the turbine and the propeller
1902 *Föttinger* is commissioned by the "VULCAN" shipyard in Szczecin where he works to study this problem; the largest gearwheel transmissions deliver only 400 hp
1905 Föttinger's patent specification on 24 June, with the basic idea of hydrodynamic power transmission. Integration of pump and turbine to reduce losses, German Patent No. 221422
1910 German Patent No. 238804 for hydrodynamic clutch = converter without reactor
1917 Gearwheel transmissions catch up with and displace torque converters in marine engineering. But the significance of the hydrodynamic clutch continues to increase
1925 *Rieseler*, a colleague of Föttinger, constructs and tests an automatic vehicle transmission with torque converter and planetary gear unit
1928 The *TRILOK* consortium in Karlsruhe (Spannhake, previously a colleague of Föttinger, Kluge and van Sanden) develop the Trilok converter.

Both phases run in a single fluid circuit, first the torque phase ($\eta_{max} = 0.8$–0.9) and then the clutch phase ($\eta_{max} = 0.98$)

1939 *General Motors* develops the first mass-produced (10 million) fully automatic vehicle transmission, the *Hydramatic*, with hydrodynamic clutch

1948 Dynaflow transmission by *GMC* with 4-phase torque converter

1955 *Borgward* builds the first automatic mass-produced transmission in Germany, with hydrodynamic converter with lock-up clutch and rear-mounted 2-speed transmission

1961 The first in-house development by *Daimler-Benz*. Hydrodynamic clutch with rear-mounted 4-speed 2-unit planetary transmission

1963 3 HP 12 from the gear manufacturer *Zahnradfabrik Friedrichshafen AG*: Trilok pressed-steel converter with rear-mounted 3-speed Ravigneaux planetary gear-set

1965 Trilok converter with lock-up clutch for commercial vehicle torque converter clutch transmission. Cast pump, pressed steel turbine

1980 Trilok converter with lock-up clutch for automatic passenger car transmission

As an electrical engineer, FÖTTINGER recognised the potential of combining a hydrodynamic prime mover (pump) and machine (turbine), and initially developed them theoretically.

Thus it was almost two decades before the first attempts to use Föttinger torque converters and clutches for a vehicle transmission. The *Trilok converter* was produced by SPANNHAKE, KLUGE and VAN SANTEN, and combined the benefits of the torque converter with those of the more efficient clutch. By mounting the reactor in the housing by means of a freewheel unit, the reactor runs freely when the moment of reaction is removed, i.e. at the precise point when the output torque falls below the input torque. The torque converter becomes a clutch, and can thus exploit the high level of efficiency of the fluid clutch in the high speed range. This combination has long since established itself in conventional automatic transmissions world-wide. In 1925 RIESELER recognised the potential of the torque converter as a device for moving off and limited torque conversion in automatic vehicle transmissions. The pump and turbine have recently been fitted with a lockup clutch in the main driving ranges to bypass the slip necessary for power transmission in the *Trilok converter*.

1.2.6 Investigation of Phenomena: Transmission Losses and Efficiency

The successful and reliable application of vehicle transmissions requires the investigation of a great variety of phenomena. Hertzian stress, tooth root strength, elasto-hydrodynamic lubrication and serviceability are just a few examples.

Let us take the phenomenon of friction as an example of historical development. Friction generates heat in a transmission. Friction arises where tooth flanks and bearing parts have rolling or sliding contact from shifting and from circulating, flowing oil.

The generation of heat in transmissions therefore soon became a matter of concern. The importance of determining transmission losses (toothing, bearing and churning losses) increased. The question of the friction coefficient along the contact path became acute. Understanding the efficiency of the transmission, and its relation to design, load and speed of rotation is important in terms of fuel efficiency. The investigation of these phenomena is shown in Table 1.7.

Table 1.7. Chronological development of research into transmission loss phenomena

1869	*Reuleaux* First formulations to determine frictional work losses
1883	*Ernst* Losses in spur gears and perpetual screws
1886	*Lewis* Measurement of efficiency of worm gears
1911	*Rickli/Grob* Measuring loss in transmissions with a torque test rig. The reading is the actual loss, and no longer the input and output power
1946	*Hofer* Approximation formula supported by measurements for calculating the efficiency of a gear stage
1954	*Niemann* develops a formula for calculating efficiency $\eta = 1 - \frac{P_V}{P_1}$ $\eta = 1 - \frac{i \pm 1}{7\,i\,z_1}$
1960	*Niemann, Ohlendorf* Systematic experiments and calculations to determine transmission losses. Gear losses in the mixed friction area (power loss through dry friction), information on churning losses and bearing losses
1965	*Hill* investigates the connection between gearing geometry and efficiency; he calculates the transmission efficiency at a constant average coefficient of friction
1967	*Lechner* Scuffing resistance with spur gears made of steel. Heat generation in gearwheels. Investigation of the phenomenon of gear scuffing as a function of gearing geometry and operating conditions
1971	*Duda* Detailed analysis of the influences of tooth geometry on efficiency
1972	*Schouten* Rolling, sliding action as elasto-hydrodynamic problem
1975	*Rodermund* Elasto-hydrodynamic lubrication with involute gearwheels. Losses with variable coefficient of friction along the contact path
1980	*Lauster* Investigation and calculation of the thermal economy of mechanical transmissions
1982	*Walter* Investigation of splash lubrication of spur wheels at circumferential speeds of up to 60 m/s
1985	*Funk* Heat dissipation in transmissions under quasi static operating conditions
1988	*Mauz* Hydraulic losses of spur gear systems at circumferential speeds of up to 60 m/s
1990	*Greiner* Investigation of lubrication and cooling of injection-lubricated spur gear systems

1.2.7 Overview

The history of the development of vehicle transmissions can be divided into four stages.

Circa 1784 to 1884 Recognition that the torque/speed characteristic of steam engines and internal combustion engines in vehicles had to be adapted to the load by means of a transmission in order to extract the maximum power. The first solutions were variable-speed transmissions with sliding or constant-mesh gears.

Circa 1884 to 1914 Search for the correct torque/speed conversion principle. A great variety of transmission designs were tried in addition to toothed gearing, such as chain, belt and friction gears, electric, hydraulic and even pneumatic transmissions, geared transmissions and especially continuously variable transmissions were tried. Each transmission design was specially tailored to a particular vehicle.

Circa 1914 to 1980 Geared transmissions became popular because of their high power/weight ratio. The idea of a standard gearbox that can easily be adapted for use in different vehicles became established. Their development has continued through the subsequent decades up to the present time in terms of service life, reliability, noise level and ease of operation (synchromesh, conventional automatic transmission, shifting with uninterrupted traction, semi-automatic transmission with electronically controlled shift control). The number of speeds and overall gear ratio steadily increased.

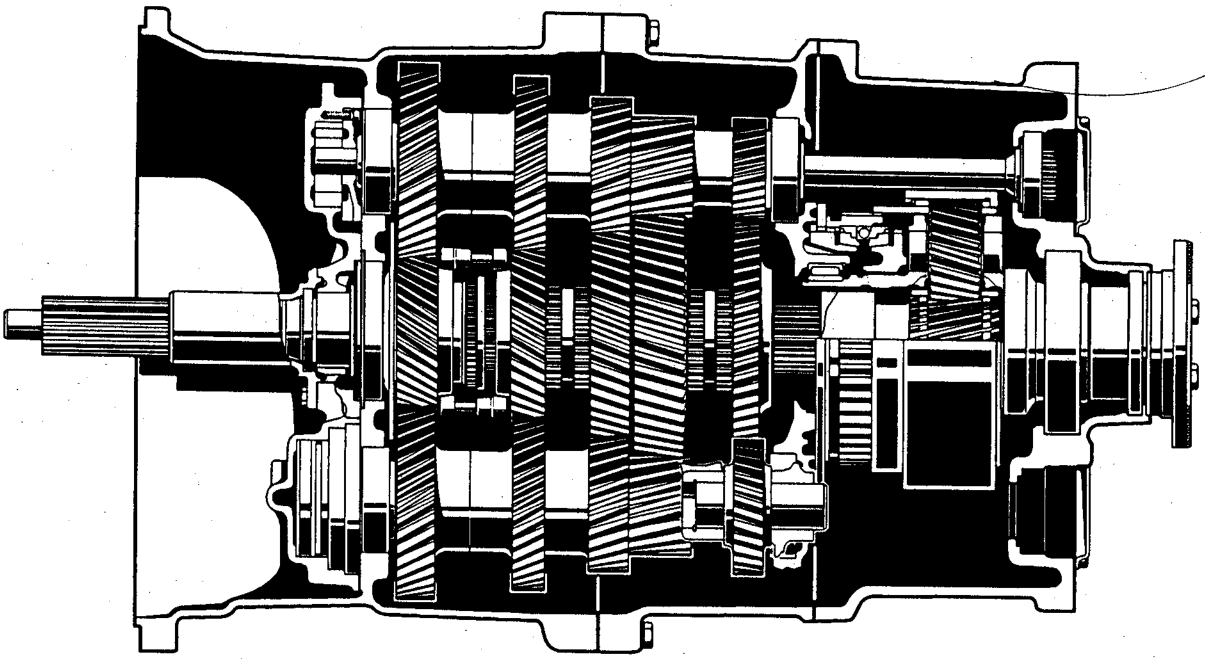


Figure 1.14. Modern 12-speed commercial vehicle transmission ZF-AS TRONiC 12 AS 1800. Double countershaft type, three-range transmission (2×3×2). Maximum transmission input torque T_1 = 1850 Nm; overall gear ratio $i_{G,\,tot}$ = 15.8

Circa 1980 to date Development effort is aimed principally at increasing fuel efficiency. Introduction of geared transmissions with five to six speeds for passenger cars, and twelve to sixteen speeds for trucks (Figure 1.14), and the greatest possible overall gear ratio. Various degrees of automation for manual gearboxes with power interruption are under development and coming onto the market for cars and commercial vehicles. Conventional automatic transmissions also acquire more speeds. Their torque converter is fitted with a lock-up clutch. There is a renaissance in the development of continuously variable transmissions for low- to medium-powered passenger cars. Work is carried out on engine/transmission management to minimise fuel consumption and emissions. Intensive effort is again devoted to alternative power-train designs with electric motor drive or hybrid drive.

2 Overview of the Traffic – Vehicle – Transmission System

Communication and mobility are essential to all human communal interaction! /WALTER KOCH, 1980/

2.1 Fundamental Principles of Traffic and Vehicle Engineering

Traffic and traffic engineering are closely and fundamentally interrelated with the economy as a whole. The *basic economic function* of transport processes is similar to that of money, without which a modern economy, based of the division of labour and with complex system processes, cannot function (Figure 2.1).

Vehicle transmissions are embedded in the "road traffic" transport system as a subsystem characterised by the following parameters:

$$Man \Leftrightarrow Vehicle \Leftrightarrow Road \Leftrightarrow Traffic \Leftrightarrow Cargo\,.$$

This formulation contains a conflict of interest (Figure 2.2). Increasing one's own quality of life is beneficial for the quality of life of society at large only in the short term. If each individual seeks to improve her quality of life without regard for others, the quality of life of the society in which she lives will suffer. This conflict of interest is starkly illustrated by the current problem of traffic and environment.

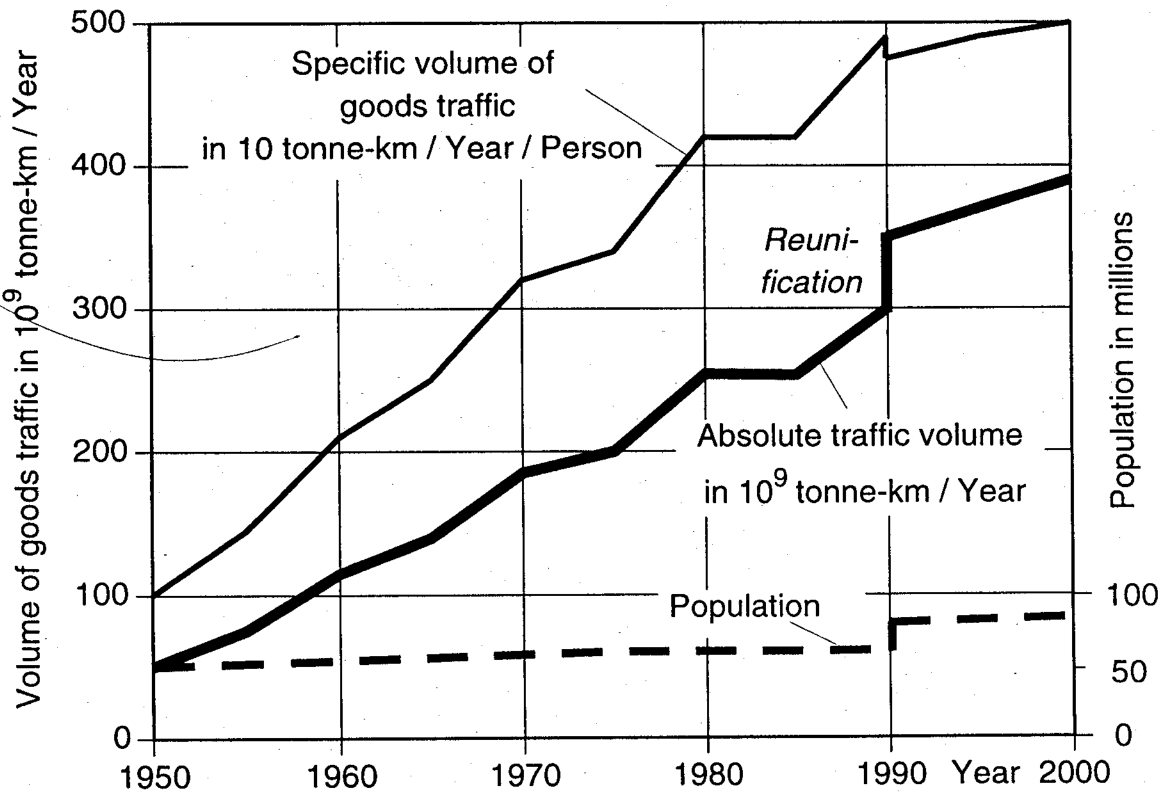


Figure 2.1. Example: Growth of goods traffic and population in Germany; figures for the whole of Germany from 1990 [2.1]

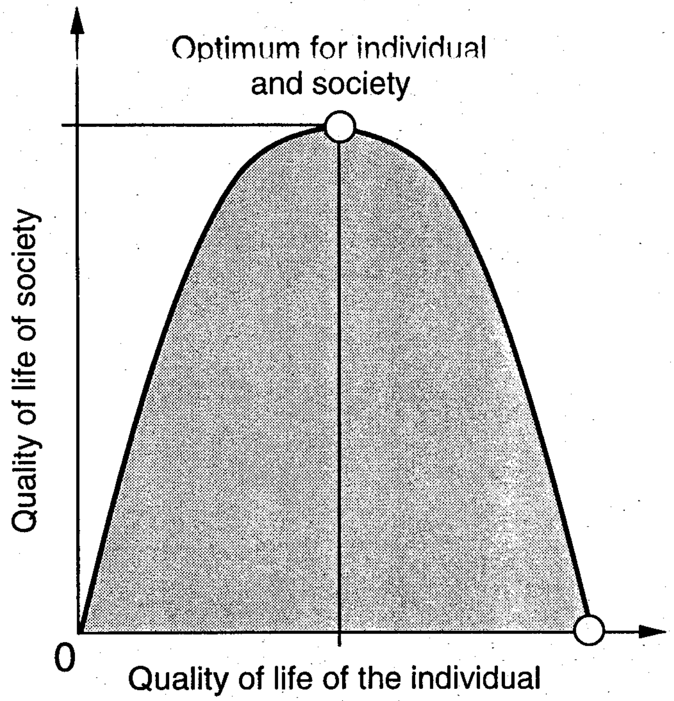


Figure 2.2. Relationship between the quality of life of a society and that of an individual

On the subject of the *Road Traffic Transport System*, H. J. FÖRSTER writes as follows [2.2]:

"Since MAN, with all his wishes and needs, far outweighs all other interests, optimising the system is not necessarily the same thing as optimising transportation performance. People using the traffic system also suffer from its ill effects, both as motorists and as citizens. Classic measures of transport effectiveness, such as transportation volume (passenger kilometres), and the cost and speed of travel, should therefore become secondary considerations. Priority has to be given to more complex human criteria such as journey quality, human satisfaction, and especially environmental impact. For goods traffic, economic factors such as transportation volume (tonne-km), transport costs (cost per tonne-km) and journey speed (km/h) continue to outweigh considerations of social and environmental impact."

2.1.1 The Significance of Motor Vehicles in our Mobile World

Mobility is an ancient basic human need. There are two factors influencing human choice of means of transport. One is actual satisfaction of objective needs, such as transportation performance, door-to-door access, and attainability of the destination. The other is satisfaction of putative subjective needs such as comfort, convenience, and freedom to decide the mode, destination and timing of the journey. Individual mobility by motor vehicle is also an expression of the freedom enshrined in our social order. Individual traffic is stochastic; it is neither determinable nor susceptible to a planned economy. Public transport is determinable. Its use can be planned.

HELLING [2.3] proposes sketching situations and development goals for road traffic by considering it as a black box (Figure 2.3) and comparing inputs and outputs. This simplified view represents the task as achieving the desired transportation output with the least negative side-effects and input of resources. The resources required to manufacture motor vehicles are shown as ambivalent to the extent that they contribute to adding value and creating jobs (Figure 2.4).

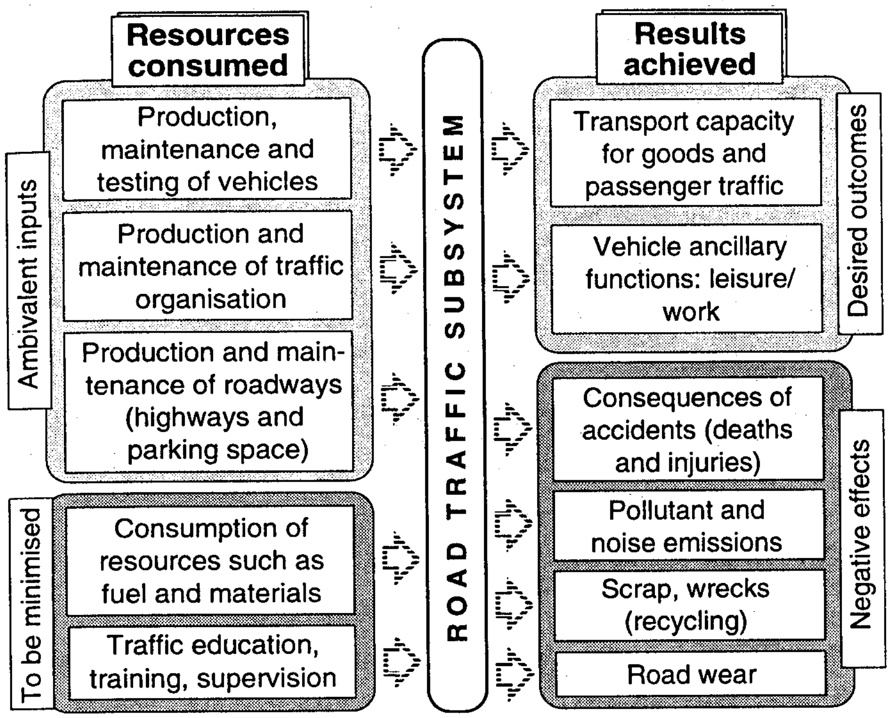


Figure 2.3. Resources consumed and results achieved by the road traffic system [2.3]

The motor industry is of enormous economic significance, both in terms of employment and in terms of satisfying human needs. For example in Germany approximately one in seven of the population earns his living from the automobile industry! The turnover of the vehicle industry is in Germany twelve times that of the machine tool industry. For a company producing predominantly motor vehicles or products for the motor vehicle industry there is no other product with the same production volume to provide anything like as many jobs. The motor vehicle has acquired great significance for the individual human being, and should serve to improve his quality of life.

There is no alternative to the motor vehicle industry in sight! The trend to the automobile continues despite impending gridlock (Figure 2.5). There is no alternative system available or under development to indicate that the motor vehicle could be displaced in the foreseeable future.

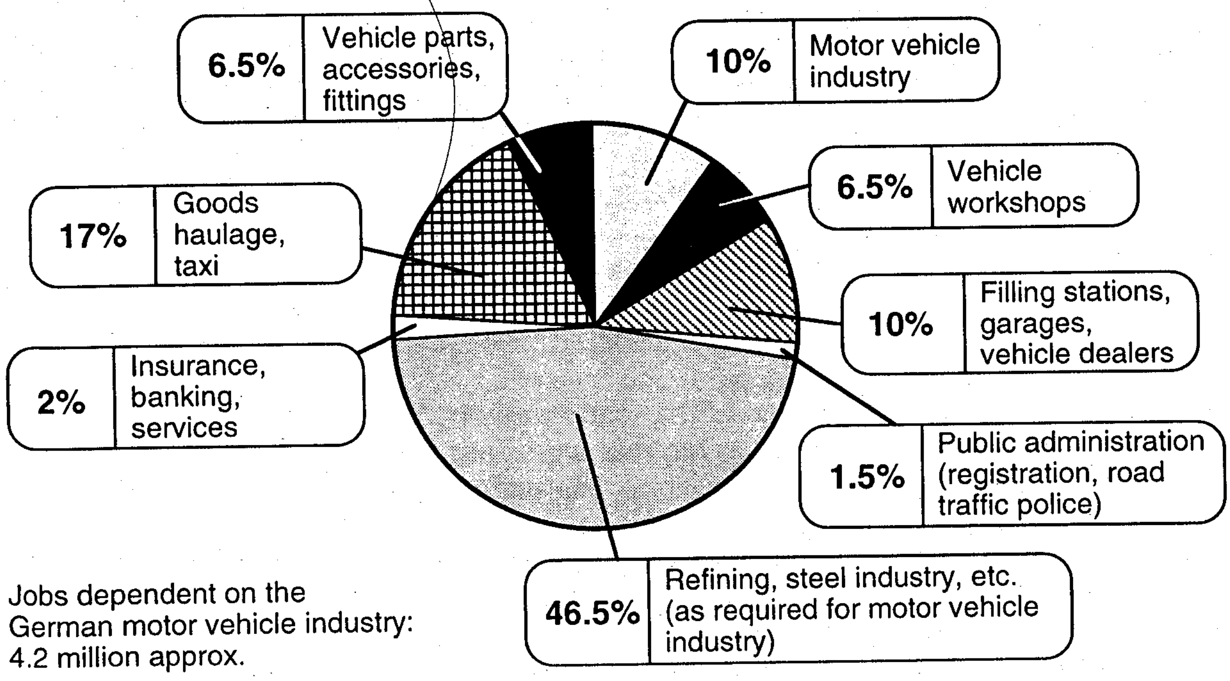


Figure 2.4. Breakdown of jobs dependent on the motor transport industry in Germany

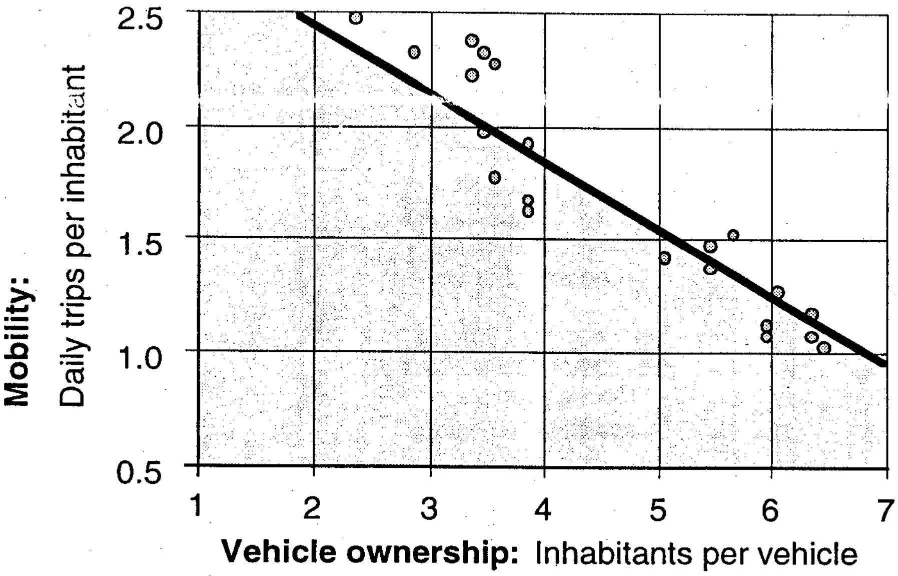


Figure 2.5. Relationship between mobility and vehicle ownership [2.4]

There is an upsurge of entrepreneurial spirit in the railways, with high-speed trains and rail trailer shipment for long-distance haulage representing steps in the right direction. But for the time being the motor vehicle fulfils the basic human need for mobility, and also enables door-to-door transport of people and goods. The motor vehicle plays a dominant role in both passenger and goods transport. The various types of vehicle are shown in Figure 2.6.

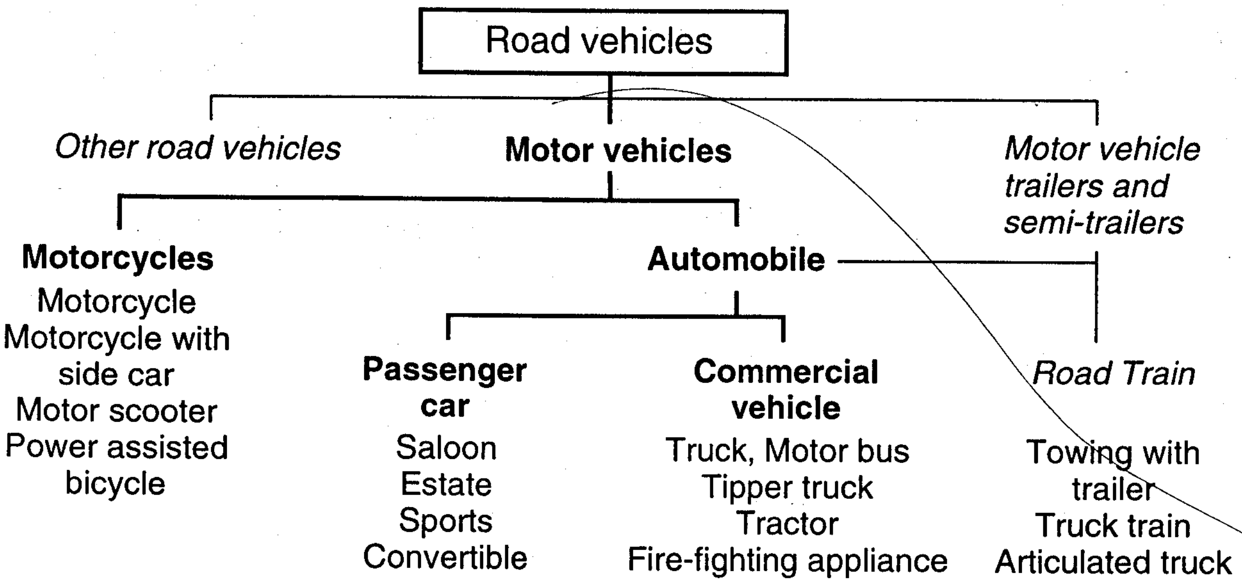


Figure 2.6. Classification of road vehicles and motor vehicles to German standard DIN 70010

Statistics confirm the relentless rise of the road transport! Environmental destruction and the threat of gridlock fail to deter, given the overriding desire for mobility. Since 1946 the number of motor vehicles in the world has increased at the rate of 10% per year (Figure 2.7).

The average annual rate of increase in Germany during the period 1907 to 1990 (Reunification) was 9%, despite the slump caused by the World Wars (Figure 2.8). During the same period, the number of motor vehicles per head of population increased from 0.00044 to 0.52345. Thus by 1990 roughly 1 in 2 inhabitants in Western Germany owned a motor vehicle (Figure 2.9).

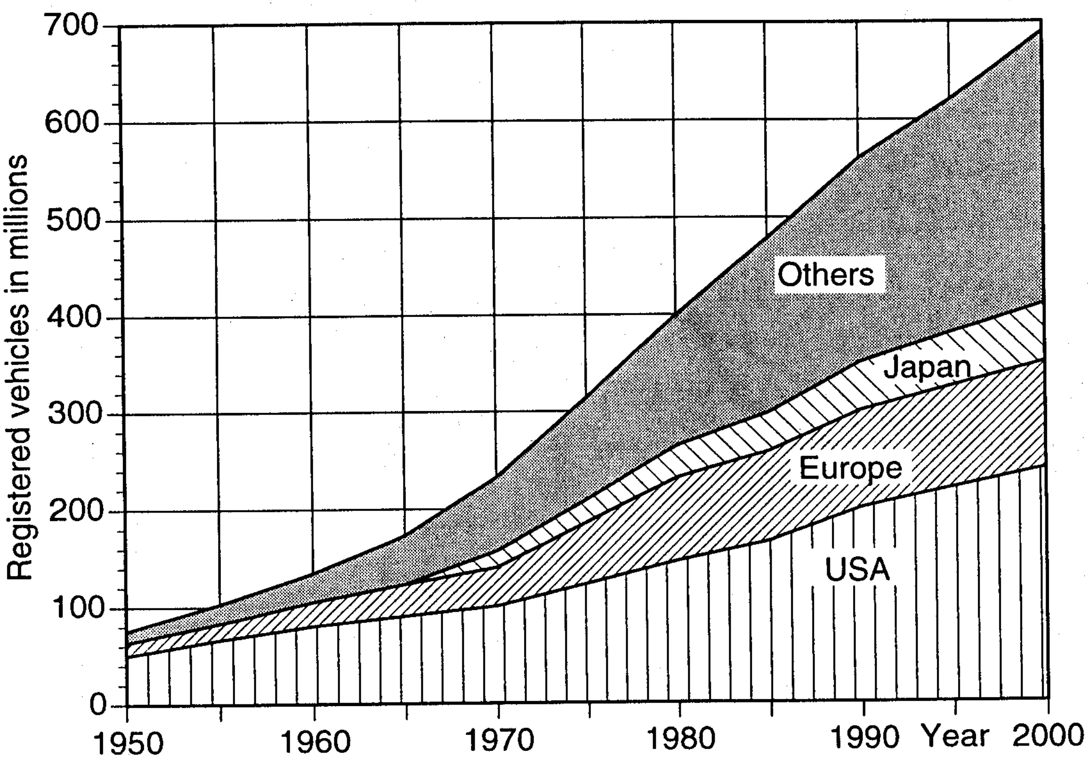


Figure 2.7. Increase in the number of motor vehicles world-wide. Source: Scientific American, September 1990

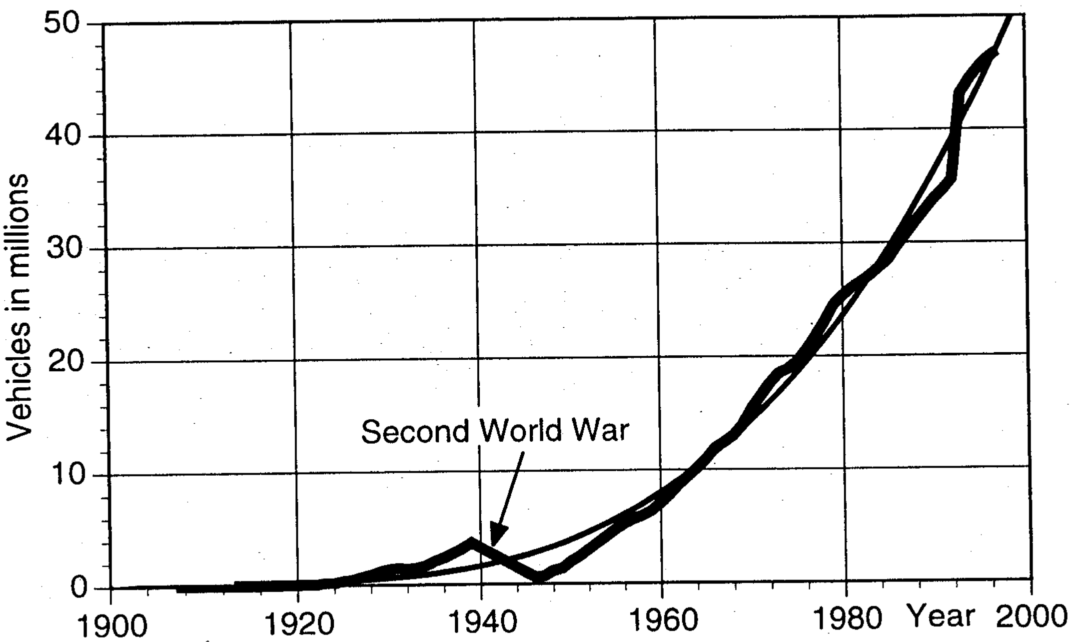


Figure 2.8. Growth of the number of vehicles in Germany [2.5]

Even with intensive promotion of mass transit facilities, the demand for motor vehicles will probably continue to increase, especially in Eastern Europe and in developing countries, although not at the rate that has pertained in industrialised countries. It is therefore important that vehicles exported to and produced in these areas are as efficient and economical as possible.

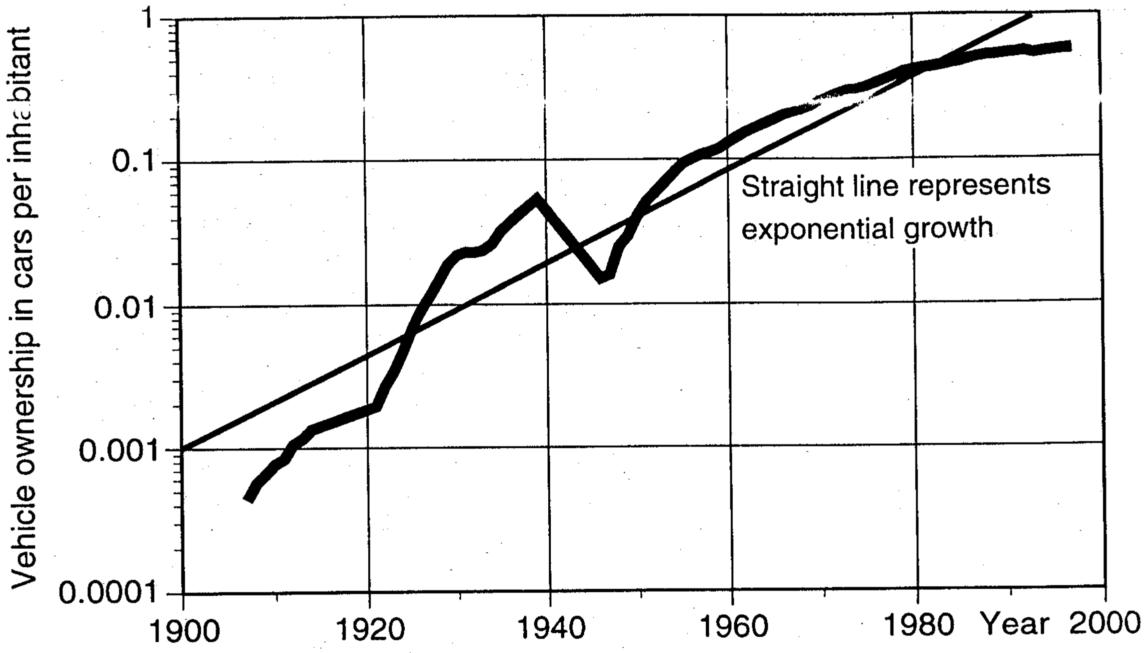


Figure 2.9. Growth in vehicle ownership in Germany [2.5]

The prognosis for the *Road Traffic Transport System* is thus:

> *There will be no long-term challenge to the pre-eminence of the motor vehicle, despite some shifts within some areas of transport. Road transport meets the demand for personal mobility and flexible movement of goods, and infrastructure spending will permit only gradual change because of the complex established structure of traffic systems [2.2].*

2.1.2 Trends in Transport Engineering

Transport is the sum of all the processes serving to overcome constraints of distance, including relocation of people, goods and information. The environment of vehicle transmissions as a product is determined by the transport system. There are five categories of transport:

- local transport: urban transport,
- regional transport: transport in conurbations,
- transport between conurbations: long-distance transport,
- continental traffic: long-distance transport,
- inter-continental transport.

As regards passenger car and commercial vehicle gearboxes, a useful distinction can be made between local and long-distance transport; in the case of buses there is a threefold distinction between urban, local and long-distance traffic (coaches). This transport structure in turn influences the design and development of vehicle transmissions.

The factors relating to transport performance are specified in Table 2.1. The most striking figures are the level of vehicle ownership, and passenger kilometres and tonne-kilometres per year as measures of passenger and goods transportation performance.

Haulage performance is the measure for goods transport (Table 2.1/5b). The gradient of increase of specific transportation performance has always been greater than the gradient of population growth throughout all stages of the historical development of road transport, i.e. the per capita consumption of goods and the haulage distance have always grown more rapidly than the population.

Table 2.1. Measures of transportation effectiveness

Name	Definition	Calculation	Unit
1. Vehicle population *MVP*	Total number of vehicles circulating in a region or state	MVP	Motor vehicle
2. Level of vehicle ownership *MVO*	Number of vehicles per head of population in a region or state	$MVO = \frac{MVP}{Inh}$	$\frac{\text{Motor vehicles}}{\text{Inhabitants}}$
3. Total volume of traffic *VT*	Number of vehicle journeys in a particular period	$VT = \sum_{i=1}^{n} f_i = \sum_{i \in I} f_i$	$\frac{\text{Journeys}}{\text{Year}}$
3a. Passenger traffic VT_P	Number of car and bus trips in a particular period	$VT_P = \sum_{i \in I_{Car}} f_i + \sum_{i \in I_{Bus}} f_i$	$\frac{\text{Journeys}}{\text{Year}}$
3b. Individual traffic VT_{In}	Number of car trips in a particular period	$VT_{In} = \sum_{i \in I_{Car}} f_i$	$\frac{\text{Journeys}}{\text{Year}}$
3c. Goods traffic VT_G	Number of truck trips in a particular period	$VT_G = \sum_{i \in I_{Com.\ veh.}} f_i$	$\frac{\text{Journeys}}{\text{Year}}$
4. Transport volume *TV*	Weight of transported goods in a particular period	$TV = \sum_{i \in I_{Com.\ veh.}} \sum_{k=1}^{f_i} g_{ik}$	$\frac{\text{tonne}}{\text{Year}}$
5. Total transportation performance *TP*	Total number of km travelled by all vehicles in a particular period	$TP = \sum_{i \in I} \sum_{k=1}^{f_i} s_{ik}$	$\frac{\text{km}}{\text{Year}}$
5a. Passenger traffic TP_P	Km travelled by passenger vehicles multiplied by the number of occupants	$TP_P = \sum_{i \in I_{Car} \cup I_{Bus}} \sum_{k=1}^{f_i} s_{ik}\, p_{ik}$	$\frac{\text{Pkm}}{\text{Year}}$
5b. Goods traffic TP_G	Km travelled by goods vehicles multiplied by the weight of the load	$TP_G = \sum_{i \in I_{Com.\ veh.}} \sum_{k=1}^{f_i} s_{ik}\, g_{ik}$	$\frac{\text{tonne-km}}{\text{Year}}$
6. Total spec. transportation performance	Transportation performance as above, but related to the number of inhabitants	$TP_{spec} = \frac{1}{Inh} \sum_{i \in 1} \sum_{k=1}^{f_i} s_{ik}$	$\frac{\text{km}}{\text{Inhabit. x Year}}$
7. Transport flow *TF*	Effective volume per hour of a traffic conduit	TF	$\frac{m^3}{h}$
8. Specific transport flow TF_{spec}	Transport flow related to the cross-sectional area required by the traffic conduit	$TF_{spec} = \frac{TF}{A}$	$\frac{m}{h}$

Observations: Time interval considered $\Delta t = 1$ year

MVP : Motor vehicle $i \in 1, \ldots, MVP = I$ where $I = I_{Car} \cup I_{Bus} \cup I_{Com.\ veh.}$

Inh : Inhabitants $j \in 1, \ldots, Inh$

f_i : Number of trips of the i-th motor vehicle / Δt $k \in 1, \ldots, f_i$

s_{ik} : Journey length of the k-th journey of the i-th vehicle

p_{ik} : Number of people of the k-th trip of the i-th vehicle

g_{ik} : Weight of the load of the k-th trip of the i-th vehicle

The trends in modern transportation engineering relate to the solution of four main problems:

- satisfying all transport needs,
- reducing the environmental impact of transportation,
- reducing primary and secondary energy consumption,
- exploiting the potential of electronic communication.

The various means of transport such as road, rail, canal and pipeline can be categorised according to their purpose or their technology. A transport system consists of:

- means of transport:
  - transport medium (vehicle),
  - transport infrastructure (road, track, and rail),
- transport organisation (operational control, administration).

Vehicle transmissions are thus components of a transport system. The factors affecting this system are environmental constraints, market needs, legislation, and individual customer requirements.

2.1.3 Passenger and Goods Transport Systems

The purpose of transport engineering is to develop and provide dependable, acceptable transport systems.

A distinction is made between passenger and goods transportation. The most important means of *transport for passengers* are walking, bicycle, motorcycle, private car, taxi (passenger car on demand), bus, railway, aircraft and ship. Roller-skates and walkways can be added to the list as exotic items.

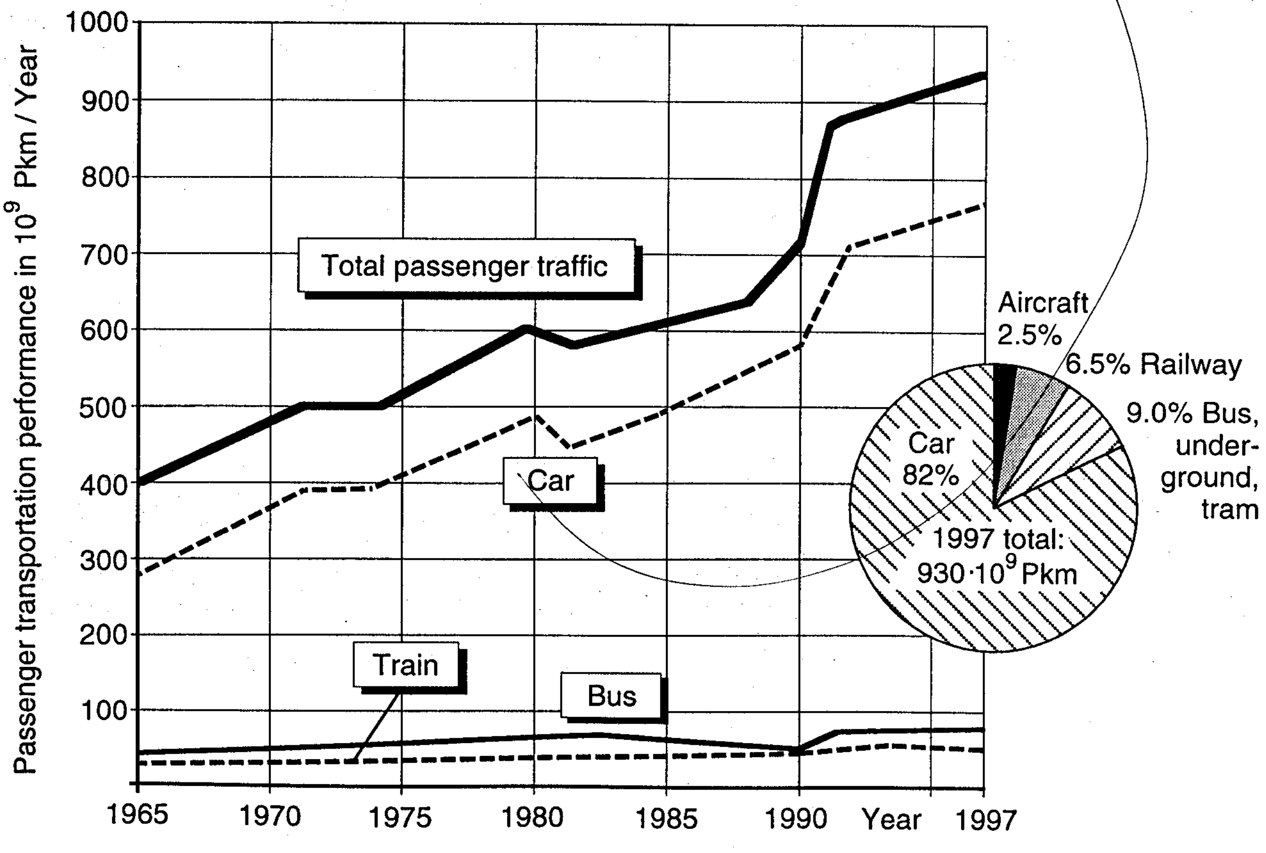


Figure 2.10. Growth of passenger traffic in Germany

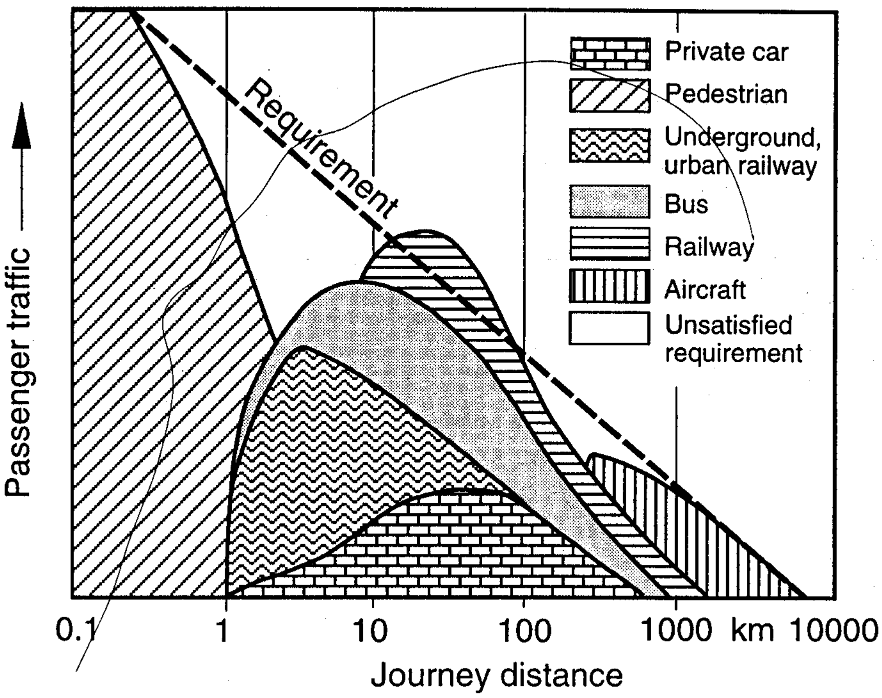


Figure 2.11. Comparison of passenger transport supply and demand related to length of journey [2.3]

Figure 2.10 shows that the largest share of passenger traffic is carried by the passenger car, far exceeding passenger transport by bus and by rail. If length of journey and transportation performance of the various means of transport are compared (Figure 2.11), it is evident that there is a transportation shortfall in the range 1–10 km and in the range 100–1000 km.

Figure 2.12 shows the travel times for various means of passenger transport related to length of journey. For short passenger trips, the passenger car and taxi are the fastest means of transport.

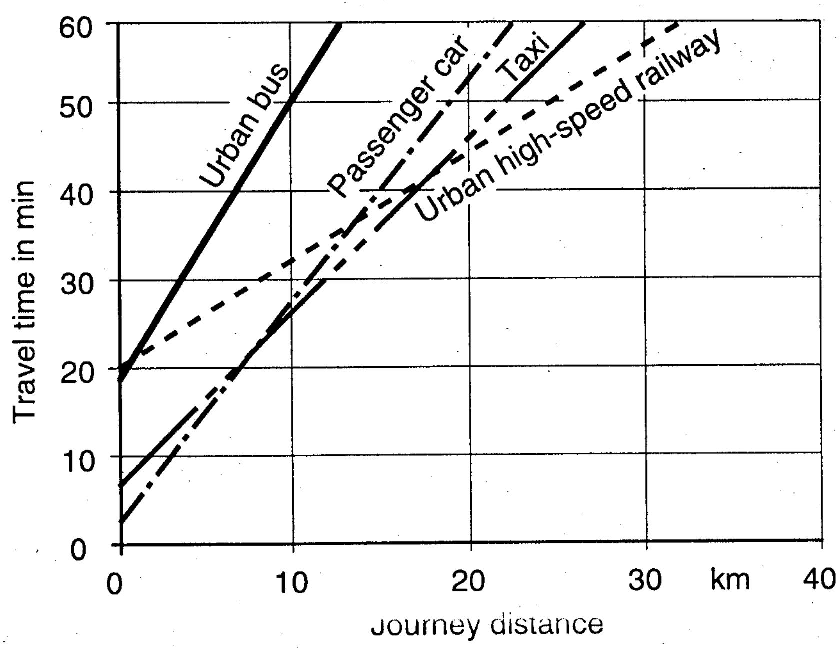


Figure 2.12. Local passenger traffic journey times [2.3]

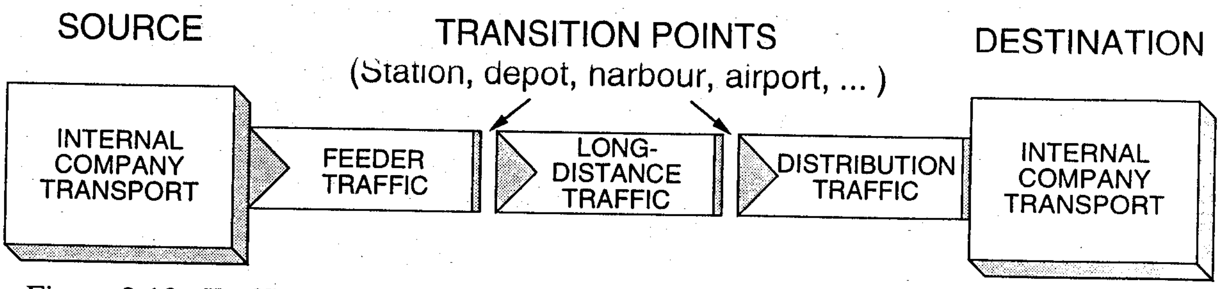


Figure 2.13. Traffic distances: Diagram of a goods transport chain

Because of their lower average speed, scheduled service buses are significantly slower than high-speed urban trains, given the same idle time. For longer trips of around 17 km and more, high-speed urban railways offer shorter travel times than passenger car or taxi. There are five different means of *transport for goods*:

- railway,
- commercial vehicle (road transport),
- ship (canal, sea freight),
- aircraft (air freight),
- pipeline.

These means of transport often form a transport chain (Figure 2.13). New approaches are urgently needed to reduce the amount of goods traffic on the roads. These means of transport can be compared by various features such as speed of transport, transport flow, space requirement, and transport flow related to space requirement.

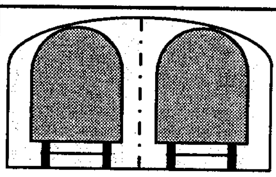
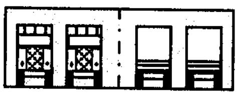
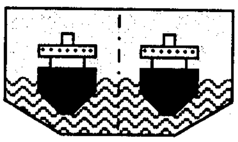
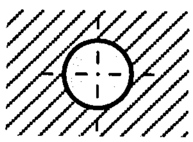
Means of trans-port	**Cross-sectional profile**	**Transport speed** v (km/h)	**Transport flow** TF (m^3/h)	**Profile surface** A (m^2)	**Specific transport flow** $TF_{spec} = TF / A$ (m/h)
Rail-way		50	20 000	37	541
Motor-way		50	14 500	115	126
Canal		12	6 250	470	13.3
Pipe-line		7.2	2 850	0.4	7125

Figure 2.14. Comparison of goods transport alternatives: rail, motorway, canal, pipeline [2.4]

If these various means of transport are compared in terms of transport speed and transport flow (Figure 2.14) the railway emerges as particularly effective, followed by the truck. The pipeline does not fare well in this comparison. The relative transport flow offers an interesting comparison, showing how well utilised the transport facility is relative to the cross sectional area it requires. On this measure the pipeline is well in the lead, followed by the railway and motorway. Canal navigation emerges unfavourably from this comparison.

Figure 2.15 shows the efficiency of various means of transport. The relationship between overall weight and payload is best with the pipeline, followed by the barge, railway and truck. The payload ratio is much less favourable for aircraft.

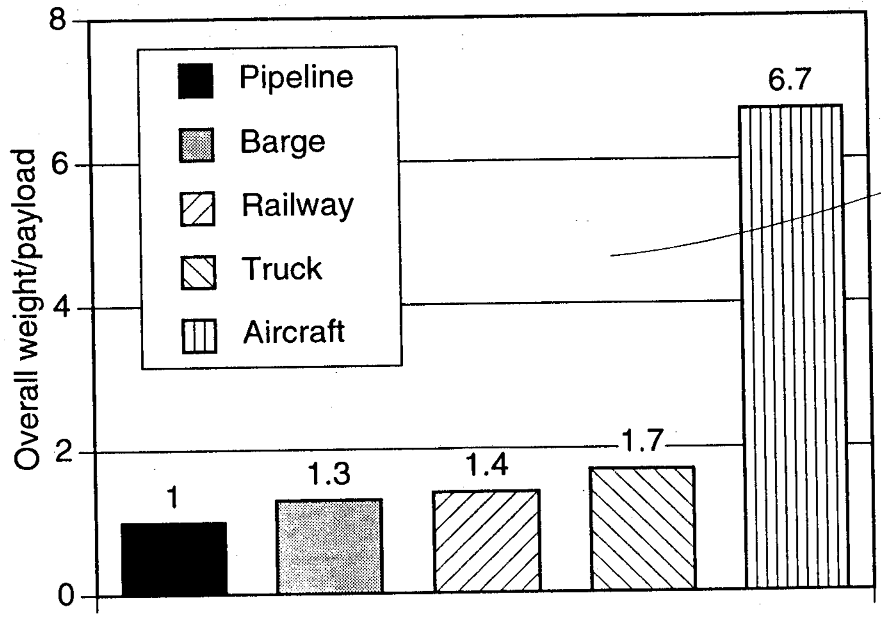


Figure 2.15. Relationship between total weight and payload for various means of transport [2.4]

Commercial vehicles carry most of the annual volume of goods traffic (Figure 2.16), with rail and barge well behind. There is no immediate prospect of sufficiently expanding rail freight to noticeably relieve the burden of goods traffic on our roads.

A key feature for a goods transport system is door-to-door access – enabling goods to be transported by the same means of transport without transhipment. The reason for the enormous increase in the number of trucks is door-to-door transport, speed and economic efficiency, and just-in-time delivery to assembly lines. Transport systems have to be assessed on the basis of satisfying transportation needs, environmental impact, and energy efficiency. First of all certain structural adjustments have to be made. In particular road and rail must be treated equally in financial terms. In this respect rail is at a disadvantage to road traffic. Using the road as a cheap storage facility in the just-in-time system of delivery is not economically viable in the long term. It contributes to traffic congestion.

2.1.4 Alternative Transport Concepts

Innovative mass transit systems have been under consideration since around 1960. There is a distinction between local transport within conurbations, and high-speed links for easing the burden of long-distance traffic on the roads. Prototypes of such concepts are in existence, some of them using new technologies such as maglev or air cushion technology. Some experimental tracks have been constructed.

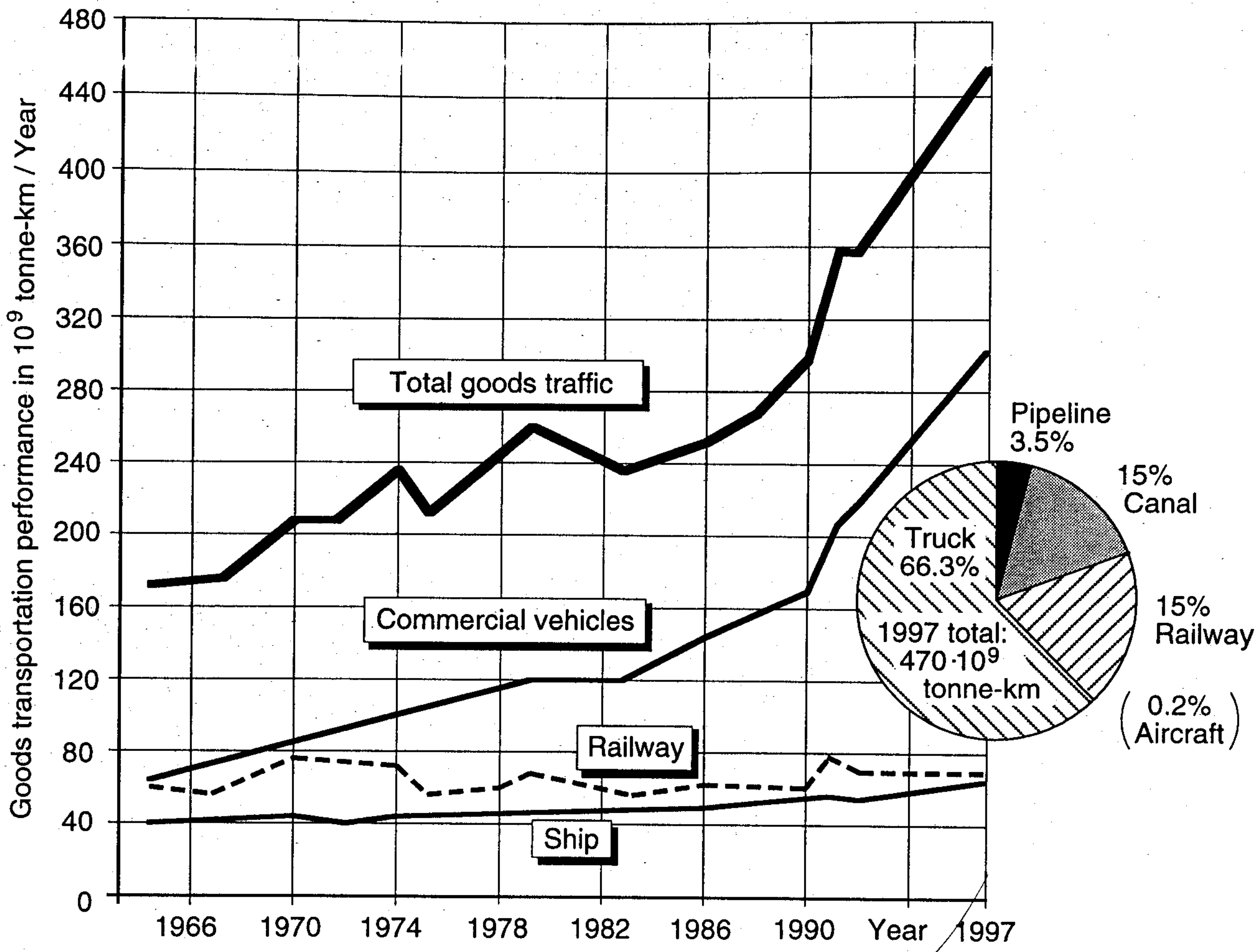


Figure 2.16. Growth of goods traffic in Germany

These concepts have not so far achieved a breakthrough, offering only minor advantages over conventional transport systems. This applies all the more since the railway also has further development potential.

The only one of these systems which appears to be establishing itself is the TRANSRAPID system [2.6]. This is a maglev monorail with linear motor drive. Designing and developing such systems is expensive, and has to be co-ordinated internationally to make it viable. Their success depends on legislation and market acceptance.

Both conventional and innovative vehicles and transport systems for local transport can be categorised in a morphological table by control system and type of use (Table 2.2). Methodical analysis of systems along these lines can help to develop proposals for innovative transport systems [2.7–2.14].

Buses are considered to have better than average development prospects for local and regional transport because of their flexibility in use, the low level of investment required and relatively low energy requirements. The significance of buses has been reflected in intensive development of automatic transmissions for buses.

An interesting new development is buses that are partly guided along tracks, operating in "dual mode" (Figure 2.17). This means buses can operate both freely on conventional roads and also under guidance along special tracks. The benefits of such systems are reduced driver stress, tracks that are easy to build and less environmentally damaging, and minimal tunnel diameters. "Demand buses" are mini-buses which can be called to bus stops. A process computer optimises transport routes within the service network and notifies the passenger of his time of departure.

Table 2.2. Classification of transport systems by control system and type of use. (Individual utility decreases from top to bottom) [2.3]

Control	Use		
	Individual	On demand	Planned
Free	Passenger car Commercial vehicle Motorcycle	Taxi Demand bus	Schedule service bus
Dual mode	Automated motorway	Dual-mode taxi	Dual-mode bus
Track-bound	Transport belt	Cabin taxi	Railway

Having lost the initiative to road traffic for decades, railways such as Germany's ICE, France's TGV, and Japan's SHINKANSEN are using high-speed trains which substantially reduce rail journey times. A European network of high-speed trains is emerging, with speeds of 250 km/h and more. These developments are characterised by high transportation performance, virtual door-to-door access, and a high level of passenger comfort.

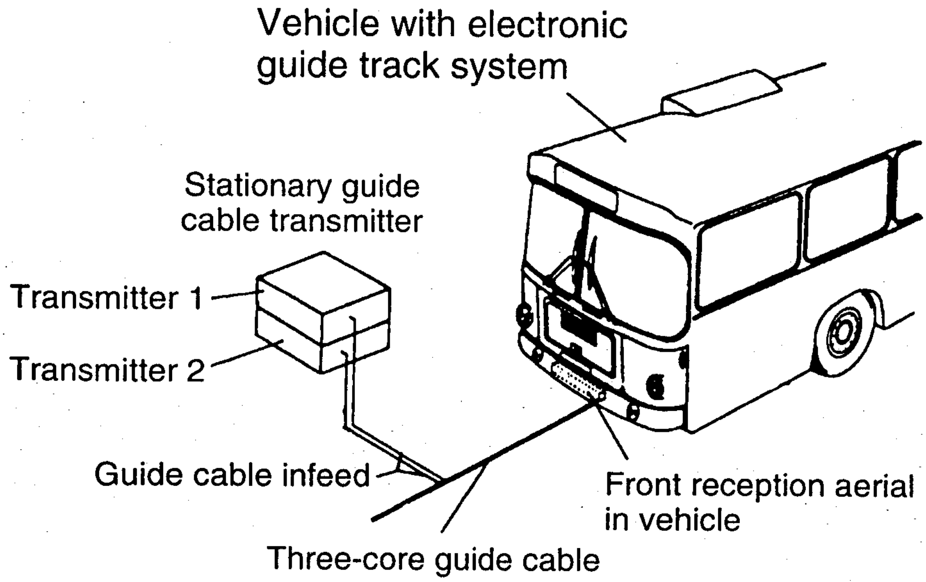


Figure 2.17. Bus with automatic electronic guide-rail control. Source: MAN

2.2 The Market and Development Situation for Vehicles, Gearboxes and Components

A progressive vehicle and gearbox development process must be not only technically sophisticated but also market-orientated. Vehicles and vehicle transmissions are developed cyclically, and have a relatively extended product and production life-cycle. Vehicle transmissions generally only require redevelopment after some 10–15 years. The transmission developer must therefore be familiar with the market situation, and be able to assess the market and changing values in society in the long term. This requires continuous observation of the market and of technological developments, as well as project planning, implementation and analysis of "futuristic" projects. Incorrect product development decisions generally lead to serious financial loss.

2.2.1 Market Situation and Production Figures

The vehicle industry is an important factor in the global economy. 53.8 million motor vehicles were produced world-wide in 1997 (Figures 2.18 and 2.19). This figure comprises 38.6 million passenger cars and 15.2 million commercial vehicles [2.5].

Definitions:

Passenger car:	Motor vehicle designed and equipped mainly for transporting people, with a maximum of nine seats.
Commercial vehicle:	Motor vehicle designed for the purpose of: transporting people – *Bus*; for transporting goods and pulling trailers – *Truck*; or just for pulling trailers – *Tractor*. This excludes passenger cars.

There are three competing centres of motor vehicle development: Europe, the USA and Japan/South Korea.

Europe is the largest producer of passenger cars (Figure 2.18). The proportion of passenger car production accounted for by sub-compacts (up to approximately 1500 cc), mid-size passenger cars (up to approximately 2500 cc) and luxury passenger cars varies greatly in the various European producer countries. Whereas France and Italy produce mostly sub-compacts and mid-range passenger cars, Germany produces a larger proportion of mid-range and luxury passenger cars. Germany produces more luxury passenger cars than the rest of Europe put together. The USA and Canada are the largest producers of trucks over 2 t (Figure 2.19).

Each market has specific conditions dictated largely by the economic and social circumstances of customers, social values, geographical factors and, importantly, legislation. Motor vehicles must satisfy market requirements to be successful. This affects the gearbox in particular, as the link between the engine and the road. Whereas in the USA more than 80% of passenger cars are fitted with automatic transmissions, the figure in Europe is only 15% (Germany 20%). In economically prosperous countries there is a market for ever better equipped passenger cars, for example with power assisted steering, automatic gearbox, air conditioning, airbag, antilock braking system, etc.

Gearboxes for commercial vehicles over 4 t gross weight are selected specially for the particular application. There are often different numbers of speeds and different methods of operation (manual, semi-automatic or fully automatic) available for a commercial vehicle gearbox from different manufacturers. The spectrum of types of transmission for commercial vehicles is as broad as the spectrum of applications.

In the USA for example, constant-mesh gearboxes are mostly used for trucks weighing over 16 t. For long stretches where no shifting is necessary, the driver is equipped with the less convenient unsynchronised constant-mesh gearbox. This also applies in many developing countries, where driver comfort is of less concern than the longer service life of the constant-mesh transmission. In Europe on the other hand synchromesh gearboxes dominate for heavy trucks as well, accounting for about 60%. The trend is towards semi-automatic or fully automatic synchromesh gearboxes.

Much European truck production is in the class up to 4 t gross weight rating (Figure 2.19). These vehicles normally have 4-speed or 5-speed synchromesh gearboxes. These are often similar to passenger car gearboxes, or are modified passenger car gearboxes.

Assuming that 10% more gearboxes than vehicles are produced, to allow for spare parts, it is possible to estimate the number of transmission components comprising gears and synchroniser packs. In Europe in 1997 approximately 160 million gearwheels and 37 million synchroniser packs were manufactured for passenger car synchromesh gearboxes. Approximately 30 million gearwheels and 6 million synchroniser packs were produced for commercial vehicle constant-mesh transmissions and synchromesh gearboxes.

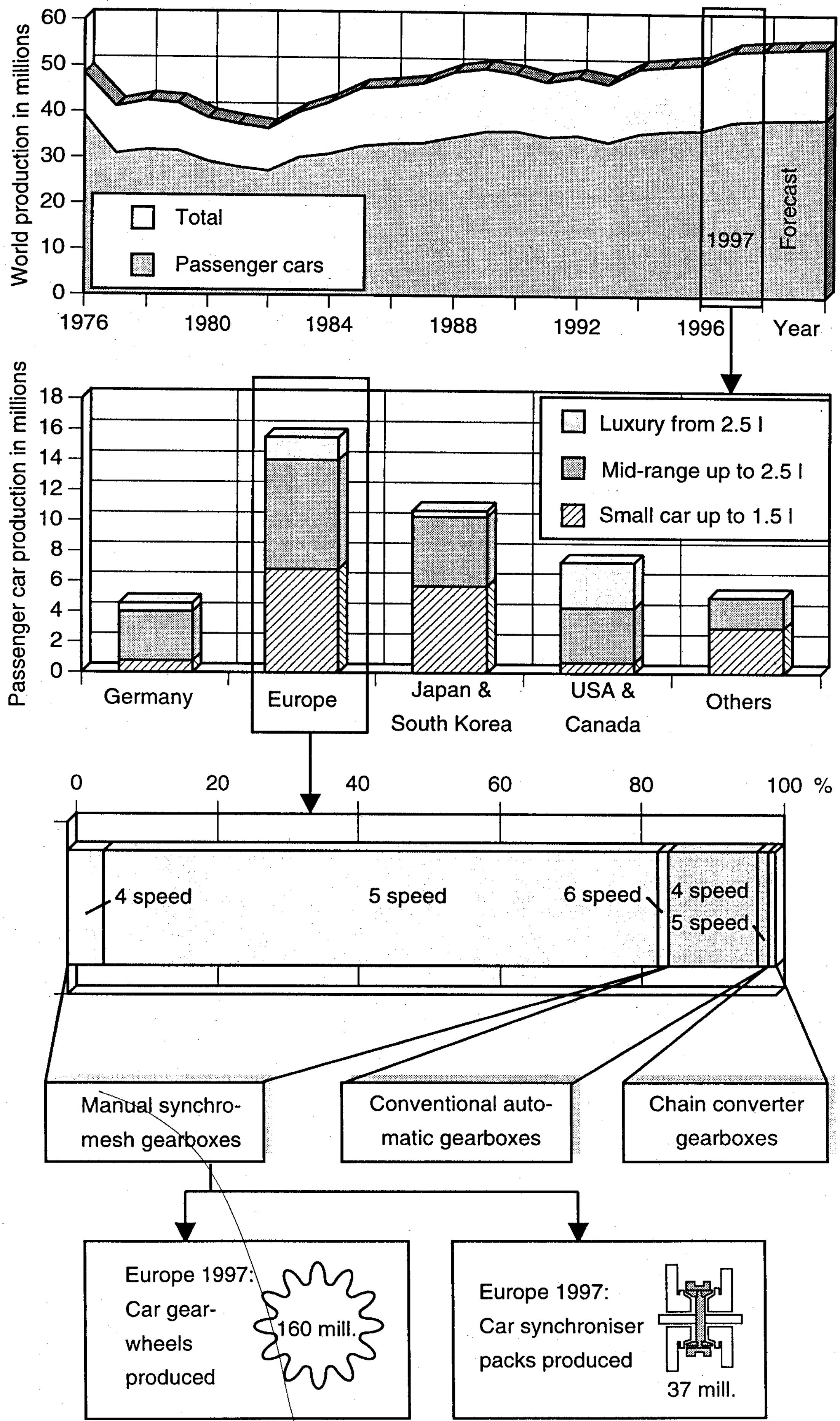


Figure 2.18. Production figures for passenger cars

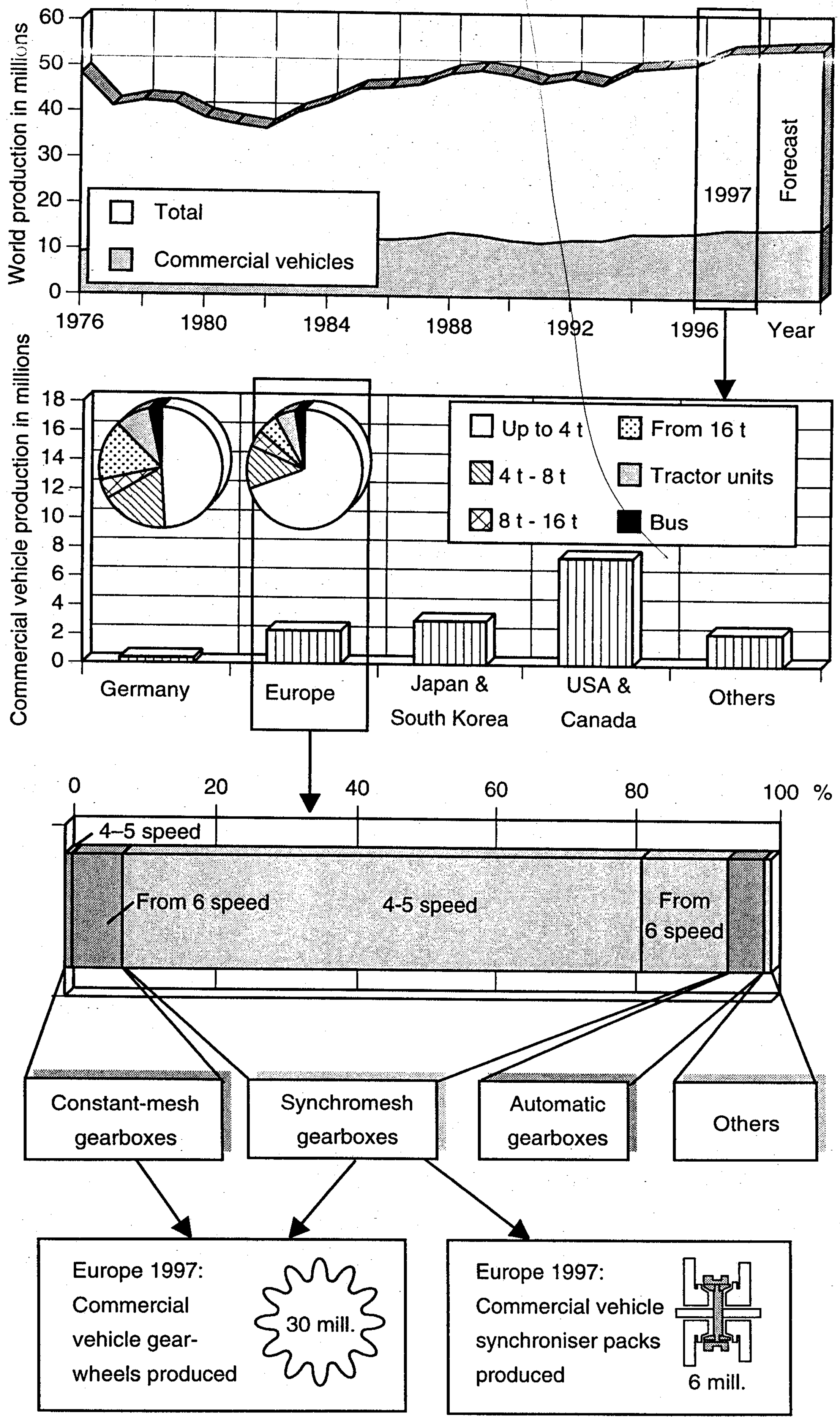


Figure 2.19. Production figures for commercial vehicles

These figures illustrate the enormous economic importance of the motor vehicle. No other product supports the production of such technically sophisticated components in such quantities. There is no product in sight, which could replace the motor vehicle as the engine of the economy.

Table 2.3 shows Germany's balance of trade in motor vehicles in recent years. 3.04 million motor vehicles were exported and 2.15 million imported in 1997.

Table 2.3. Germany's balance of trade in motor vehicles [2.5]

	Quantity in millions	1991	1992	1993	1994	1995	1996	1997
EXPORT	Passenger cars	2.20	2.57	2.06	2.27	2.47	2.65	2.82
	Commercial vehicles	0.16	0.16	0.12	0.14	0.17	0.19	0.22
	Total	2.36	2.73	2.18	2.41	2.64	2.84	3.04
IMPORT	Passenger cars	2.52	2.18	1.64	1.64	1.77	1.85	1.93
	Commercial vehicles	0.22	0.21	0.11	0.15	0.16	0.19	0.22
	Total	2.74	2.39	1.75	1.79	1.93	2.04	2.15

Table 2.4 lists the main independent manufacturers of motor vehicle transmissions. Vehicle transmissions, especially mass-produced passenger car gearboxes, are mostly produced by the motor vehicle manufacturers themselves.

Table 2.4. Some independent manufacturers of vehicle transmissions

	Passenger car mechanical	**Passenger car automatic**	**Commercial vehicle mechanical**	**Commercial vehicle automatic**
Western Europe	– GETRAG – ZF	– ZF – BORG-WARNER – GENERAL MOTORS	– ZF – EATON	– ZF – VOITH – RENK
USA	– AISIN – NEW VENTURE GEAR	– AISIN – BORG-WARNER	– DANA – EATON – NEW VENTURE GEAR – CLARK	– ALLISON (GM) – TWIN DISC
Japan	– AISIN	– AISIN – JATCO	– AISIN	– AISIN – JATCO

2.2.2 Development Situation

The pace of technological development has accelerated in recent years. Microelectronics continues to find new applications in vehicles, vehicle transmissions and in their development. The pace of product development is increasingly becoming an important competitive factor for individual companies.

Modern development processes such as "simultaneous engineering" or "rapid prototyping" are used. The aim is to reduce characteristic development times for vehicle gearboxes, which are shown in Table 2.5.

Modern development strategies in Europe are required in the face of the challenge from the Far East. European industry must accordingly make better use of existing potential, and assess its own depth of development and production [2.15].

The general principles of vehicle gearbox development are set out in Chapter 14 "Overview of the Development Process, Product Planning and Systematic Design".

Table 2.5. Typical development lead times for vehicle transmissions

	Passenger cars		**Commercial vehicles**	
Development phase	Synchromesh transmission	Automatic transmission	Synchromesh and constant-mesh transm.	Automatic transmission
Concept phase	4	5	6	9
Design and development	5	5	6	12
Prototype production	6	9	9	12
Testing	12	15	15	15
Pre-production development	9	12	12	18
Σ Months	36	46	48	66

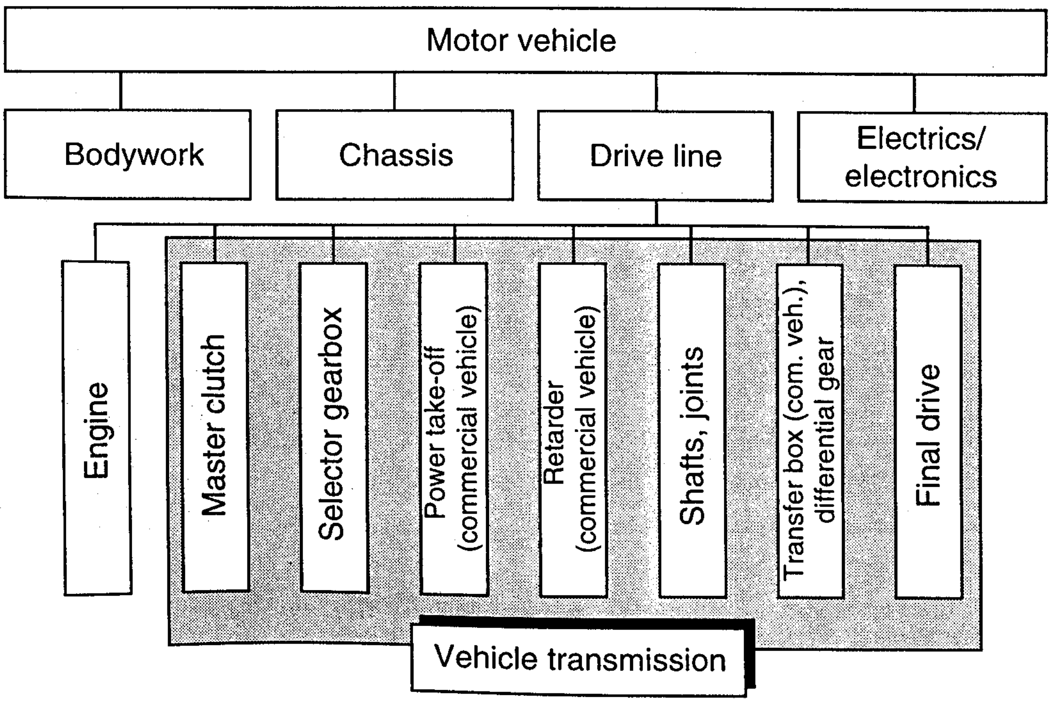


Figure 2.20. Definition of the subject area of vehicle transmissions

2.3 Basic Elements of Vehicle and Transmission Engineering

It is essential to clearly define the vehicle and its intended use as the starting point for targeted vehicle gearbox development. Definitions and basic physical elements of automotive and transmission technology are explained below. They are the basis for the observations in the following chapter.

The topic of "Automotive Transmissions" in this book covers all components of the power-train assembly with the exception of the engine (Figure 2.20).

In the development of vehicle transmissions a distinction must be made between variables which the designer can influence (internal factors) and those he cannot influence (external factors). These factors are set out in Table 2.6.

Table 2.6. Internal and external factors affecting the development of vehicle transmissions

Internal factors which can be influenced by the design engineer	External factors which cannot be influenced by the design engineer
❍ Bodywork	❍ Road profile
❍ Chassis	❍ Driving style
❍ Electrics/electronics	❍ Payload
❍ Engine	❍ Traffic conditions
❍ Vehicle transmission (see Figure 2.20)	❍ Weather conditions

2.3.1 Systematic Classification of Vehicles and Vehicle Use

The development of a vehicle transmission relates to the type of vehicle, its power unit and its intended use. A classification of vehicles oriented to transmission development assists systematic analysis.

Table 2.7 shows a transmission-oriented classification of vehicles which has proved effective in practice. Vehicles are first divided into passenger cars, commercial vehicles, construction vehicles, tractors and special vehicles. Passenger cars are split into two main groups by engine size: up to 75 kW and over 75 kW.

The commercial vehicle category is split into buses and trucks. The truck category is further broken down by gross weight.

The bus category can conveniently be broken down by function into urban and local buses, and long-distance coaches.

A further relevant dimension is the type of use intended. There are three basic types of use in a transmission-orientated classification of automobiles:

- ❍ *On-road use.*
- ❍ *On/off-road use*, e.g. construction vehicles. This combined type of use, which is typical e.g. for dump trucks, means the transmission must provide economical propulsion both on and off-road.
- ❍ *Off-road use*. Vehicles move predominantly off-road, possibly with occasional on-road use. This category includes tracked vehicles or extremely heavy special vehicles not permitted on normal roads, such as landfill vehicles or mining vehicles.

Table 2.7. Transmission-oriented classification of motor vehicles by type of vehicle and type of use; GVW: Gross vehicle weight. *) Another feature is the number of seats

<table>
<tr><th colspan="3">Type of vehicle</th><th colspan="3">Type of use</th></tr>
<tr><td colspan="2" rowspan="2">Pass. cars</td><td>Power $P < 75$ kW</td><td rowspan="11">On-road</td><td rowspan="11">On/off-road (building sites)</td><td rowspan="11">Off-road</td></tr>
<tr><td>Power $P > 75$ kW</td></tr>
<tr><td rowspan="6">Commercial vehicles</td><td rowspan="3">Trucks</td><td>Light commercial vehicles: GVW < 7.5 t</td></tr>
<tr><td>Medium-duty commercial vehicles: GVW < 16 t</td></tr>
<tr><td>Heavy commercial vehicles: GVW > 16 t</td></tr>
<tr><td rowspan="3">Buses *)</td><td>Urban</td></tr>
<tr><td>Local</td></tr>
<tr><td>Long-distance (coaches)</td></tr>
<tr><td colspan="3">Agricultural tractors</td></tr>
<tr><td colspan="3">Construction vehicles</td></tr>
<tr><td colspan="3">Special vehicles</td></tr>
</table>

2.3.2 Why do Vehicles Need Gearboxes?

Almost all automobiles in use today are driven by internal combustion engines with cyclical combustion, working on the spark-ignition or diesel principle. The factors determining the power output and performance characteristics of internal combustion engines are explained in Section 3.3.

In addition to the many advantages of the internal combustion engine, such as high power-to-weight ratio, relatively good efficiency and relatively compact energy storage, it has three fundamental disadvantages:

- Unlike steam engines or electric motors, the combustion engine is incapable of producing torque from rest (zero engine speed), see Figure 3.12.
- An internal combustion engine only produces maximum power at a certain engine speed (Figure 3.12).
- The efficiency of the engine, i.e. its fuel consumption, is very much dependent on the operating point in the engine's performance map (Figure 3.17, engine performance map).

With a maximum available engine power P_{max} and a road speed v, the so-called "ideal traction hyperbola" $F_{Z,\,Aid}$ and the effective traction hyperbola $F_{Z,\,Ae}$ can be calculated as follows:

$$F_{Z,\,Aid} = \frac{P_{max}}{v} \quad \text{or} \quad F_{Z,\,Ae} = \frac{P_{max}}{v}\,\eta_{tot}\,. \tag{2.1}$$

Thus if the full-load engine power P_{max} were available over the whole speed range, the traction hyperbolas shown in Figure 2.21 would result. But for the internal combustion engine the traction profile also shown in Figure 2.21 would result. The maximum traction between tyres and road is limited by the adhesion limit.

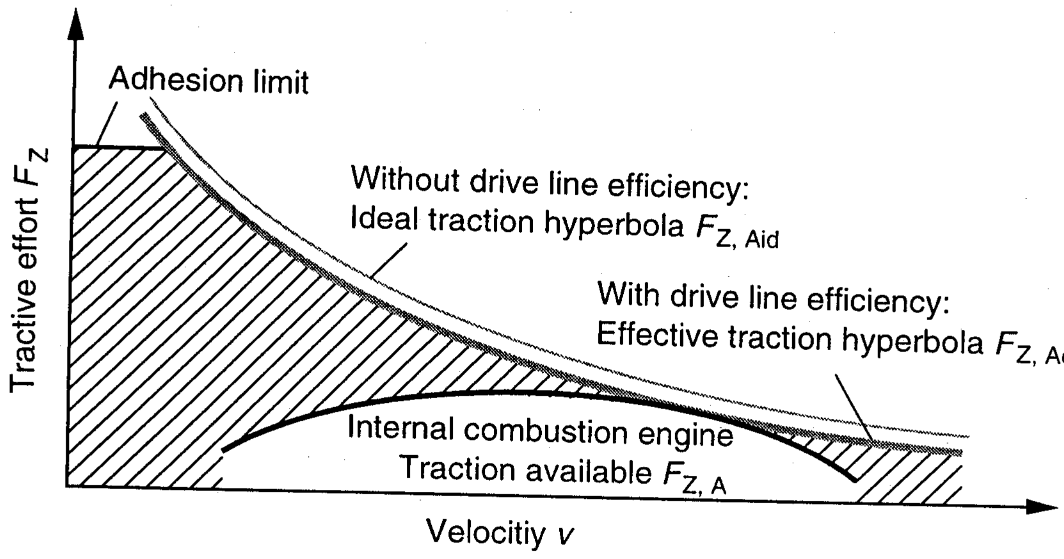


Figure 2.21. Secondary map of an internal combustion engine without a gearbox

The problem with the internal combustion engine as a prime mover is now clear. The whole shaded area in Figure 2.21 cannot be used without an additional output converter. The output converter must convert the characteristic of the combustion engine in such a way that it approximates as closely as possible to the ideal of the traction hyperbola (Figure 2.22).

Output converter:
- ❍ *speed converter*
  ≡ mechanical or hydrodynamic clutch,
- ❍ *speed/torque converter*
  ≡ geared transmission or continuously variable transmission.

The proportion of shaded area, i.e. the proportion of impossible driving states, is significantly smaller when an output converter is used, and the power potential of the engine can be better applied. Figure 2.22 shows how increasing the number of speeds gives a correspondingly better approximation to the traction hyperbola. With a continuously variable transmission the traction hyperbola can correspond to the traction characteristic curve over the range of ratios.

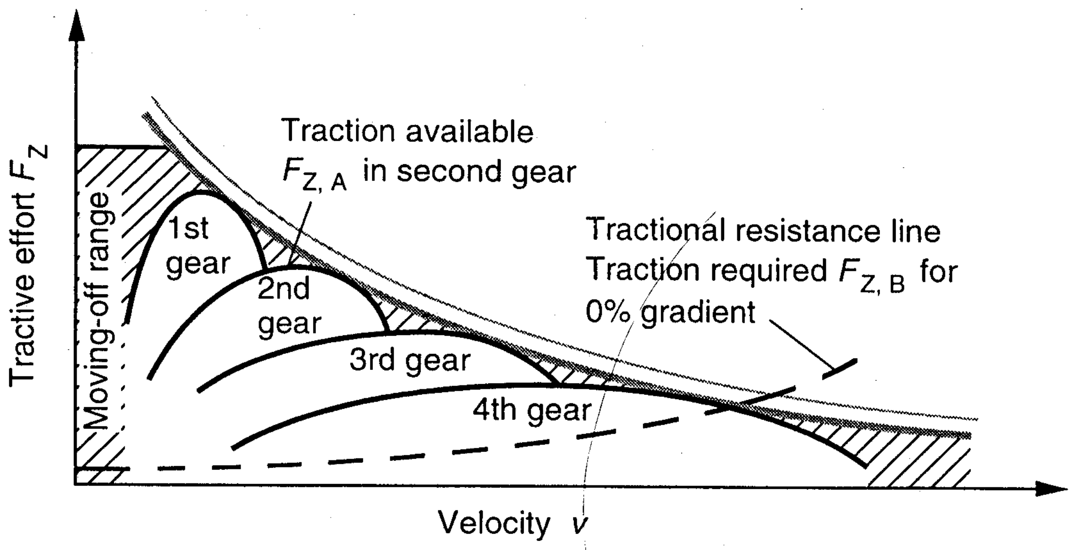


Figure 2.22. Secondary map of an internal combustion engine with rear-mounted 4-speed gearbox: traction diagram

The second fundamental disadvantage of the internal combustion engine, that is does not deliver torque from rest, is overcome by means of moving-off element (force locking clutch).

The master clutch is generally mounted between the engine and the transmission in the power train. See Section 4.2 "Speed Converter for Moving Off".

The role of the transmission in defining engine operating points that are favourable in terms of efficiency and performance is dealt with in detail in Chapter 5 "Matching Engine and Transmission".

2.3.3 Main and Auxiliary Functions of Vehicle Transmissions, Requirements Profile

For transmissions to function adequately as the link between the engine and the drive wheels, it is advisable for the design engineer to consider the "vehicle transmission" as a functional whole, including gearbox and master clutch, i.e. the whole system of adapting speed and torque, including changing gear and moving off.

The four main functions of a vehicle transmission are to:

- *Enable the vehicle to move off from rest.*
- *Adapt power flow.*
  Convert output torque T_2 and output speed n_2.
  Enable reverse motion.
- *Enable permanent power transmission.*
  Positive or force locking engine power transmission with minimal loss.
- *Control power matching.*

In addition to these main requirements there are some ancillary requirements of a transmission, also known as operational requirements, which substantially affect its competitiveness. The result of a survey of manufacturers and users of commercial vehicle transmissions is shown in Table 2.8.

The importance of individual ancillary requirements is displayed on a scale from $0 \equiv$ unimportant to $10 \equiv$ very important. Such a rating list of ancillary requirements is referred to as a *requirements profile.* The ancillary requirements of vehicle transmissions can be broken down as follows:

- operational reliability,
- gearbox costs,
- ease of repair,
- ease of operation,
- power matching,
- installation dimensions and weight,
- customisability,
- emissions (noise, oil, etc.).

Starting from the requirements profile derived from statistical surveys or empirical data, goal conflicts due to design or economic constraints can be recognised, and a suitable compromise sought on the basis of the weighting allocated.

These compromises are translated into specific criteria for the development engineer, contained in the *specification,* which is examined in Chapter 14.

Table 2.8. Commercial vehicle transmission requirements based on a customer survey [2.16]. Score ≡ average assessment of importance on a scale of 0–10

Requirements	Score
Long service life	9.00
Low repair costs	4.89
Low production costs	4.59
Range of ratios	4.02
Gear stage	3.47
Early failures unusual	3.13
Length	2.92
Long maintenance interval	2.63
Operating travel/force	2.59
Low weight	2.55
Traction constantly available	2.47
Vibration resistance	2.35
Small number of seals	2.33
Danger of operator error	2.20
Low maintenance costs	2.19
Overload capability	2.08
Overdrive available	2.06
Fitting wearing parts	1.92
Type of shift	1.86
Owner repairable	1.72

Requirements	Score
Number of parts	1.58
Power take-offs	1.55
Assembly tools	1.55
Time to remove and replace	1.50
Shift connections	1.49
Temperature resistance	1.48
Crawler gear available	1.46
Accessibility	1.42
Type of unit construction	1.40
Ratio variants	1.39
Method of assembly	1.34
Moving off	1.32
Spare parts procurement	1.26
Low power loss	1.22
Drive take-up	1.16
Low development cost	1.16
Standard connections	1.11
Height above main shaft	1.07
Clear gearshift pattern	1.06
Good service network	1.00

2.3.4 Interrelations: Direction of Rotation, Transmission Ratio, Torque

The key factors in a gearbox are the direction of rotation, the transmission ratio and the torque. In order to be able to compare and assess various transmission designs and variants, we therefore need definitions to use as a standard for all considerations [2.17].

Definition of Direction of Rotation

The direction of rotation in a power train is defined as positive when the direction of rotation is clockwise in a right-handed Cartesian system of co-ordinates. This is as viewed against the forward direction of movement related to the vehicle, as shown in the left-hand diagram in Figure 2.23.

In the case of complicated gear plans, especially in the case of planetary gears, it is advisable to represent the speeds of rotation of the individual transmission elements with their sign and relative to each other.

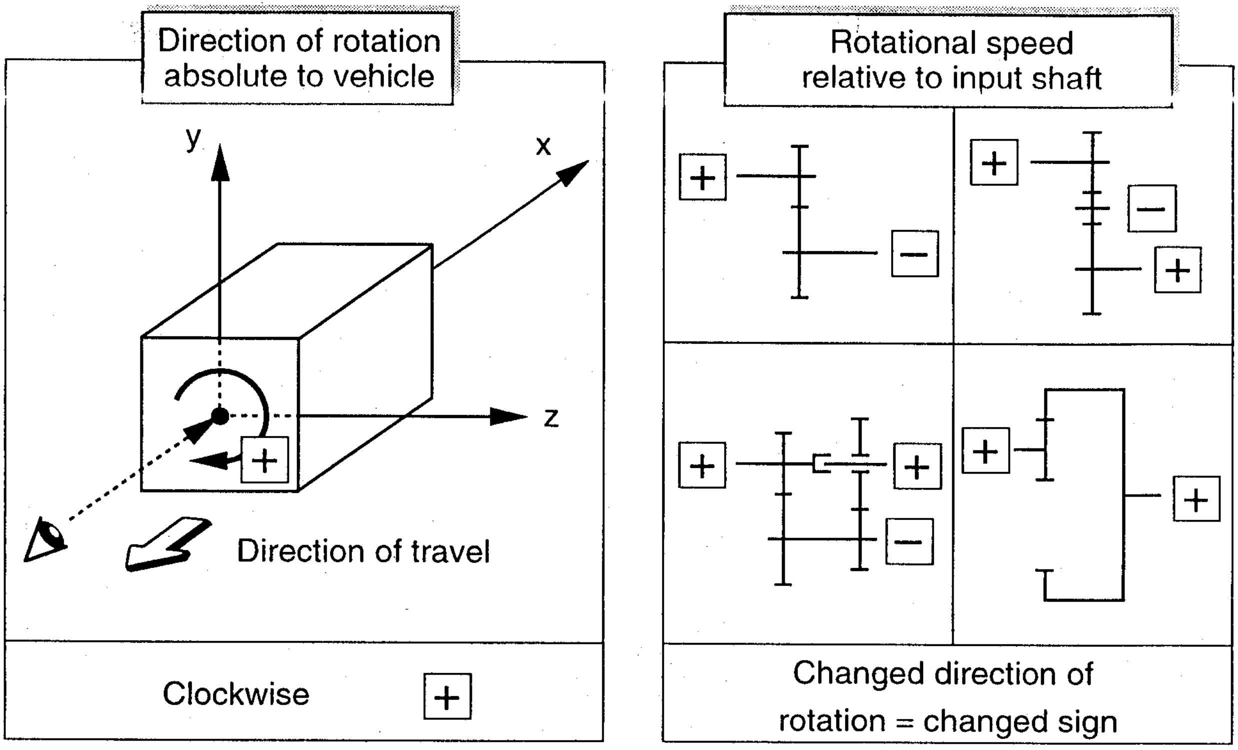


Figure 2.23. Definition of direction of rotation in a power train

It is in principle of no importance which of the two possible directions of rotation is defined as positive, but normally the direction of rotation of the transmission input shaft is taken as positive (right-hand diagram in Figure 2.23).

Definition of Transmission Ratio

The transmission ratio i_G is the relationship between the angular velocity ω_1 of the input shaft of a gearbox to ω_2 of the output shaft.

$$i_G = \frac{\omega_1}{\omega_2} = \frac{n_1}{n_2} \; . \tag{2.2}$$

The relationship between the output speed n_2 and the input speed n_1 in a power-train component is called the speed conversion ν (Equation 4.2). The torque conversion μ gives the relationship between the output torque T_2 and the input torque T_1 of a power-train component (Equation 4.3).

With Equation 2.2 and the sign rules derived above, the following characteristics result for the transmission ratio:

$i_G > 0$	transmission input and output shaft rotate in the same direction,
$i_G < 0$	change of direction of rotation in the transmission,
$\lvert i_G \rvert > 1$	speed reducing ratio,
$\lvert i_G \rvert < 1$	speed increasing ratio.

In the case of continuously variable transmissions and with transmission combinations:

$i_G = \infty$	stationary output with rotating input,
$i_G = 0$	stationary input with rotating output.

The ratios inside a gearbox are designated by the gear ratio u. The gear ratio u of a gear pair is the ratio of the number of teeth z_2 of the larger wheel to the number of teeth z_1 of the smaller wheel (pinion).

$$u = \frac{z_2}{z_1} \quad \text{with} \quad z_2 \geq z_1 \ . \tag{2.3}$$

German standard DIN 3990 specifies that in the case of spur gears the number of teeth of a wheel with external gearing is positive, and that the number of teeth of a wheel with internal toothing (ring gear) is to be taken as negative.

Definition of Torque

Further important factors affecting a gearbox are the torque values acting on its shafts. Their directions of action must be defined by showing their sign. Here again it is in principle of no consequence which torque direction is taken as positive.

The torque direction of the transmission input shaft is normally also defined as positive. By separating the transmission components and establishing torque equilibrium, it can be shown that the torque direction is always reversed along a free connecting shaft.

As shown in Figure 2.24, the torque direction changes along a transmission component, but the direction of rotation does not. The sign of the power P absorbed (positive) or delivered (negative) at a particular point can be determined from the speed of rotation and the torque at that point in the transmission by means of Equation 2.4.

$$P = \omega T = 2 \pi n T \ . \tag{2.4}$$

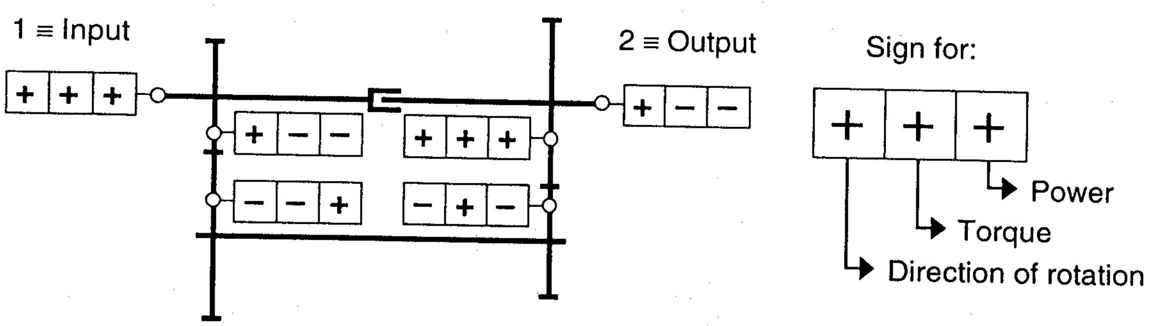
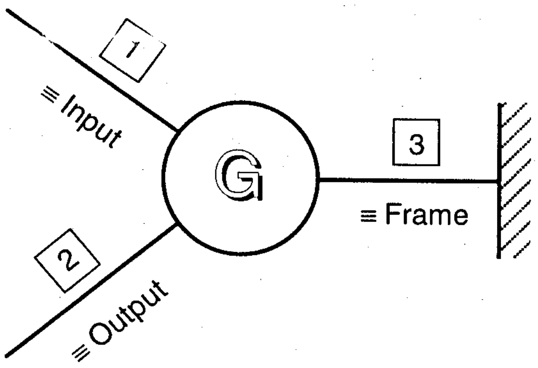


Figure 2.24. Sign rules for rotational speed, torque and power

MÜLLER [2.18] proposes four important rules for speeds of rotation, torque values and power values in a transmission:

- *All parallel shafts in a transmission rotating in the same direction will have speeds with the same sign.*
- *In an "input shaft" the signs for speed of rotation and torque are the same; in an "output shaft" they are opposite to each other.*
- *"Input power" is always positive; "output power" is always negative.*
- *The two equal connecting torque values of a free connecting shaft have opposite signs at the connecting ends.*

A transmission consists of at least three sections, one of which must be the "frame". This important condition is necessary to provide a reaction for the difference in force or torque between the input and output side resulting from the conversion of movement. In vehicle transmissions the gearbox housing is the frame.



$$T_1 + T_2 + T_3 = 0 \,, \tag{2.5}$$

$$P_1 + P_2 + P_3 = 0 \,, \tag{2.6}$$

$$T_1\,\omega_1 + T_2\,\omega_2 + T_3\,\omega_3 = 0 \,, \tag{2.7}$$

$$\omega_3 = \omega_{\text{frame}} = 0 \,. \tag{2.8}$$

Figure 2.25. WOLF transmission symbols

The symbolic representation proposed by WOLF [2.19], as shown in Figures 2.25 and 2.26, clearly illustrates these relations. From Equations 2.2, 2.7 and 2.8 it follows that

$$T_2 = -T_1 \frac{\omega_1}{\omega_2} = -T_1\, i_G \,. \tag{2.9}$$

With Equation 2.5 the reaction torque T_3 can be calculated as

$$T_3 = -T_1 - T_2 = T_1 \left(i_G - 1\right) . \tag{2.10}$$

Two fundamental characteristics of transmissions concerning reaction torque emerge from Equation 2.10:

- For a transmission ratio of $i_G = 1$, i.e. direct drive, the transmission takes on the function of a clutch, i.e. its frame does not have to provide any reaction torque.
- The reaction torque of the frame changes its sign, i.e. its direction, when shifting from a speed reducing gear to a speed increasing gear.

The third "frame" section can also be in the form of a second input or second output member. In these cases the term "differential drives" is used.

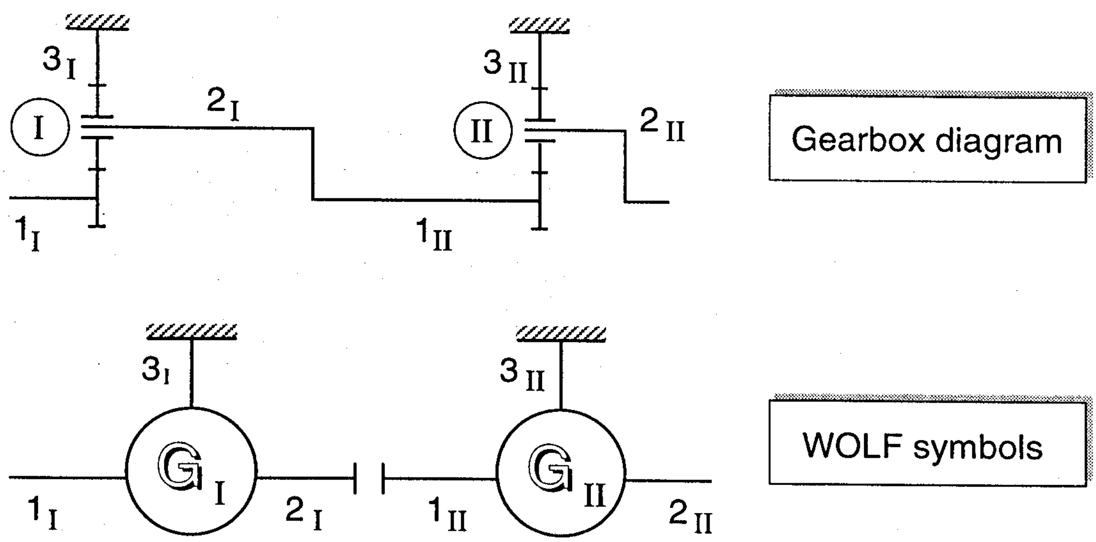


Figure 2.26. Planetary coupling gear represented by WOLF symbols

2.3.5 Road Profiles, Load Profiles, Typical Vehicle Use and Driver Types

In addition to the "internal" factors affecting the transmission (i.e. the design data of the individual vehicle sub-assemblies), the designer should be fully informed about "external" factors such as driving style, vehicle use and road type (see also Table 2.6).

This information can be acquired through field trials and customer surveys, focusing on the following criteria:

- *Road types*
  Proportion of total mileage on various types of roads such as motorway, main road, urban traffic or mountain road (see Table 2.9).
- *Loading*
  Percentage distribution of journeys with numbers of passengers, cargo and trailer weight.
- *Driving style*
  Shifting frequency, gearshift engine speed, acceleration habits in town (moving off from traffic lights), on main roads (when leaving built up areas) and on the motorway (overtaking).

In addition to the above-mentioned criteria it is useful to classify drivers by type (Table 2.9).

Table 2.9. Breakdown of driver type in terms of power-train loading in a passenger car [2.20]

	Proportion of kilometres covered in %			
Driver type	Motorway	Main road	Urban traffic	Mountain road
Motorway driver	70	14	13	3
Trunk road driver	30	56	11	3
Urban driver	30	23	45	2
Mountain driver	40	30	20	10

The load patterns determined by road tests and computer simulation can be translated into load cycles using suitable classification methods. The transmission service life is then generally estimated using the MINER and HAIBACH damage accumulation hypothesis, based on the duty cycles and the corresponding Wöhler curves.

Information on load cycles and service life is given in Section 7.4 "Operational Integrity and Service Life". The development of transmissions using computer simulation road tests is described in Chapter 15.

2.4 Fundamental Performance Features of Vehicle Transmissions

The development process is successful only if it results in a product that sells

Any specification for a product contains the following requirements: it must be suited to its intended purpose, affordable (economic) and acceptable (environment friendly, easy to operate, and functional). These superordinate development goals define the fundamental performance features of a transmission. Vehicle transmissions have to be designed to provide torque conversion suited to operating conditions, adapted to the consumers' requirements and at a competitive price.

This means specifically service life appropriate to the intended use (serviceability), low noise level, low weight, high efficiency and ease of operation (Figure 2.27). Such performance features are often enough for a rapid rough comparison of one's own designs and those of others.

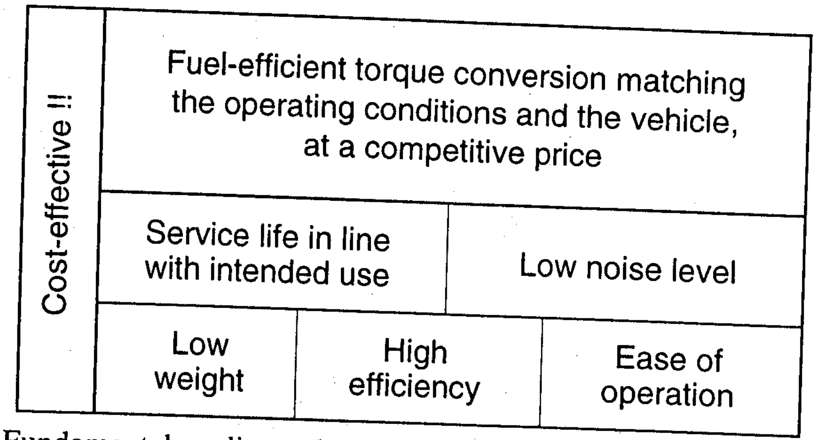


Figure 2.27. Fundamental quality and performance features of vehicle transmissions

2.4.1 Service Life and Reliability of Transmissions

Premature failure and subsequent premature wear impair the availability and economic efficiency of a passenger car or a truck. It is often overlooked that a slight over-rating of the transmission relative to the engine can achieve an enormous increase in service life. Over-rating by 15% doubles the service life, whereas the price of the transmission only increases by 15% at a first approximation (see Figure 2.30).

The standard values listed in Table 2.10 apply to service life, the most important performance characteristic of transmissions. B_{10} service life refers to the service life within which 10% of the transmissions of a production batch of a given transmission type fail. With passenger car transmissions, the required B_{10} service life is 150 000 km. The market requirements of commercial vehicle transmission service life are now very high. They vary from case to case (Table 2.10).

Table 2.10. B_{10} service life of passenger car and truck transmissions under different operating conditions

Operating conditions		B_{10} service life in km
Passenger cars		≥ 150 000
Trucks		
On building sites	(Off/on-road)	≥ 300 000
Urban traffic	(Stop and go)	≥ 400 000
Long distance		≥ 800 000

The applicable load profile for the individual speeds (Section 7.4) is particularly important. It depends on the route profile, vehicle loading, and driving style.

Transmission reliability requires thorough calculation and analysis at the design stage. See also Section 7.4 "Operational Integrity and Service Life" and Chapter 16 "Reliability and Testing of Vehicle Transmissions".

It must be borne in mind that a transmission is a system with components that are more or less reliability-critical. Components such as gearwheels, shafts or bearings can now be readily calculated. Other critical elements such as seals are still more difficult to calculate.

2.4.2 Centre Distance Characteristic Value

Characteristic values can be determined for transmissions and other products by using standard procedures to establish values for key basic variables of the future design quickly and without complex calculations. This method of "design using characteristic values" is ideal for use with computer-aided design. When the draft design has been transferred to the computer, the graphic and quantitative aspects need to be refined. Such characteristic values relate for example to size, mass or cost.

The centre distance a of a transmission is its most important parameter. The smaller the centre distance can be with a given output torque T_2, the smaller the overall dimensions of the transmission. The centre distance is determined by the gear with the greatest torque multiplication $i_{G,\,max}$ (first gear). It is possible to gain a good impression, of the order of size of a competitive centre distance before carrying out any calculation, by relating the centre distances of one's own transmissions and those developed by competitors to their output torque T_2. This type of analysis encapsulates a great variety of production and operating experience with gearbox designs proven in practice.

Figure 2.28 is an example of such a centre distance analysis, showing the trend of the centre distance with coaxial, two-stage countershaft gearboxes as a function of the output torque T_2 at the output shaft. Centre distances for passenger car gearboxes are accordingly 65–90 mm. Gearboxes for medium weight commercial vehicles have a centre distance of 100–130 mm, whereas units for heavy trucks are in the range 130–160 mm. The scatter is explained by differing design methods, by different applications with different load profiles, and by technical production factors. As a result of the existence of transfer lines for centre distance drill holes in the gearbox housing, it is often more economical to achieve the required service life by adapting the face widths rather than optimising the centre distance.

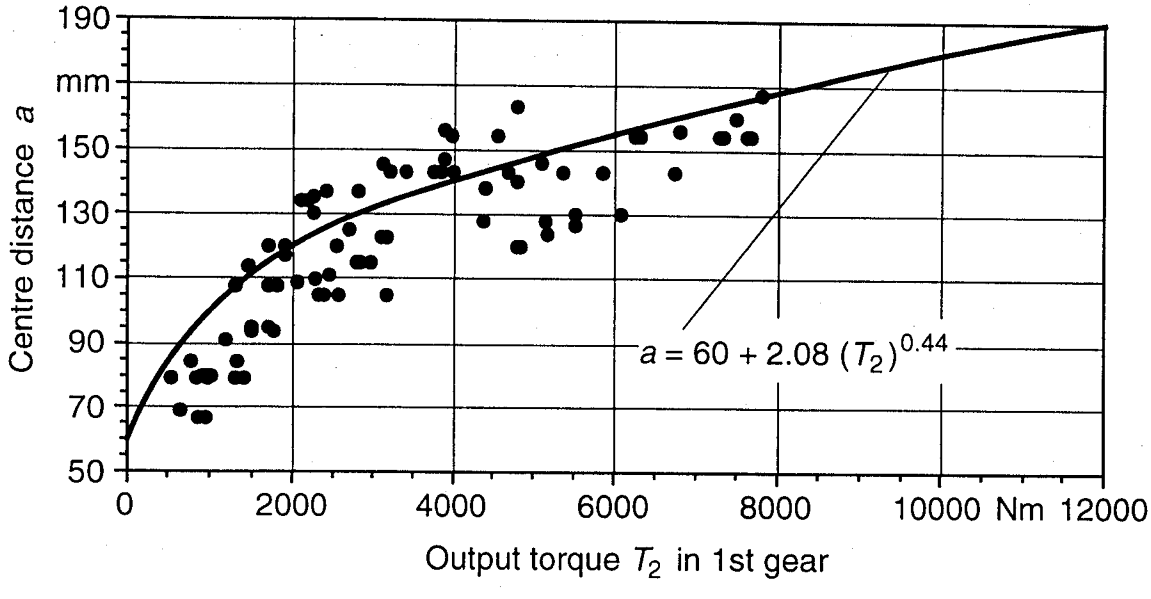


Figure 2.28. Trend of centre distance a with coaxial two-stage countershaft gearboxes as a function of the output torque T_2 at the output shaft

Another reason is in-company standardisation of centre distance intervals. By reference to Figure 2.28 and the relationship derived from it between centre distance a and output torque T_2 in first gear, we derive

$$a = 60 + 2.08\, T_2^{\,0.44} \tag{2.11}$$

or

$$a = 60 + 2.08 \left(i_{G,\,max}\; T_1\right)^{0.44}. \tag{2.12}$$

This enables centre distances in millimetres to be estimated for coaxial, two-stage countershaft gearboxes of single-range construction, leading to an economical gearbox size. For this purpose only the maximum transmission input torque T_1 in Nm and the required maximum ratio of the transmission $i_{G,\,max}$ need to be known. Of course such a prognosis cannot replace precise transmission design and centre distance calculation. See Chapter 7 "Design of Gearwheel Transmissions for Vehicles".

2.4.3 Gearbox Mass Characteristic Value

Another key performance characteristic of the transmission is its mass m_G. The cost of a gearbox is proportional to its weight at the first approximation. The mass of the transmission can be related to the input torque T_1 the maximum ratio $i_{G,\,max}$ and the number of gears. Figure 2.29 shows an analysis of a large number of transmissions that have been designed and produced.

The gearbox mass m_G in kg is plotted against the parameter $T_2\, z^{\,0.5}$ or $T_1\, i_{G,\,max}\, z^{\,0.5}$. The points on this diagram can be approximated by a parabola. Their equation is

$$m_G = 0.49\, T_2^{\,0.58}\, z^{0.29} \tag{2.13}$$

or

$$m_G = 0.49 \left(i_{G,\,max}\; T_1\right)^{0.58} z^{0.29}. \tag{2.14}$$

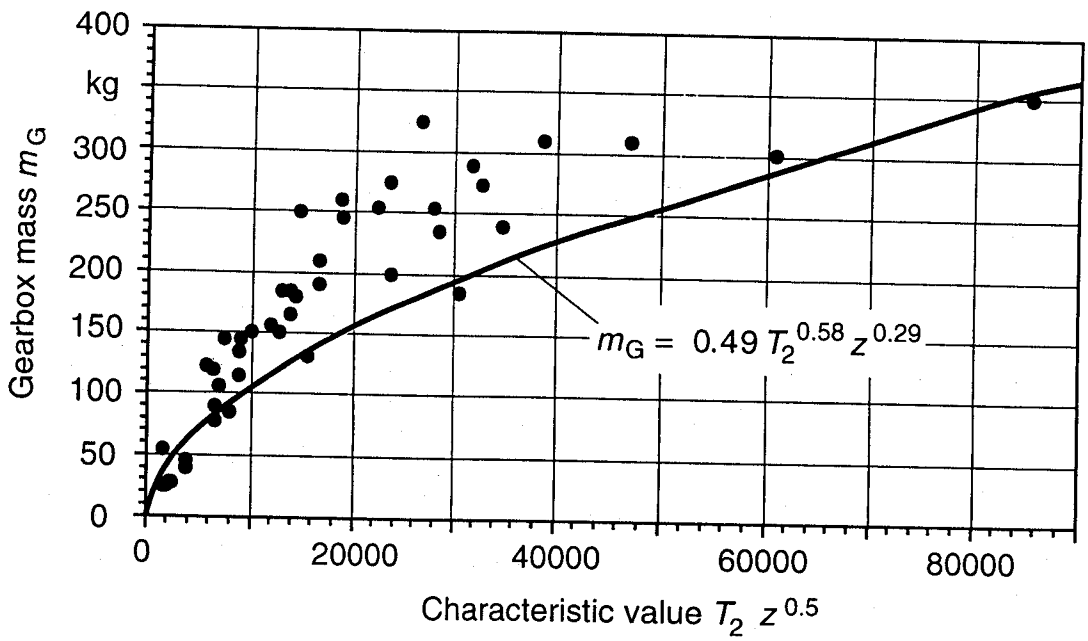


Figure 2.29. Trend of gearbox mass m_G for vehicle transmissions with a cast iron housing (density $\rho = 7.3$ kg/l) as a function of the output torque T_2 at the output shaft and the number of speeds z

The diagram in Figure 2.29 can be used in two ways:

- assessing the anticipated weight of a transmission, whose total ratio, input torque and number of speeds are known from the specification,
- checking whether a developed transmission is competitive in relation to its mass.

2.4.4 Gearbox Cost Characteristic Value

As with transmission mass, parameters can be established for transmission costs or selling price. Figure 2.30 shows an analysis of transmissions that have been built. The relative selling price *RSP* is plotted against the parameter $T_2 z^{0.5}$ or $T_1 i_{G,\max} z^{0.5}$. (Reference: passenger car gearbox with input torque T_1 = 200 Nm, z = 5 speeds, and $i_{G,\max}$ = 3.85: RSP = 1.) The points in this diagram can be approximated by a parabola with the equation:

$$RSP = 0.035\, T_2^{\,0.45}\, z^{0.225} \tag{2.15}$$

or

$$RSP = 0.035 \left(i_{G,\max}\; T_1\right)^{0.45} z^{0.225}\ . \tag{2.16}$$

The selling price (or production cost) trend shown in the graph enables the transmission designer to estimate whether the efficiency of his transmission is worthwhile for the customer in terms of the price/performance relationship of other transmissions. It is also possible to forecast the achievable market price at the product planning stage.

The comparative cost of different types of vehicle gearboxes is also of interest. The specific design and production features and different operating conditions and safety requirements of the different types of gearboxes result in significantly different relative weight costs. The relative weight costs (Euro/kg) are shown in Figure 2.31 for seven different types of transmissions used for different purposes. Commercial vehicle synchromesh transmissions have the lowest weight costs (12 Euro/kg at 1997 prices). Then come construction vehicle transmissions. Conventional passenger car automatic transmissions are not significantly more expensive, because of the large production volume.

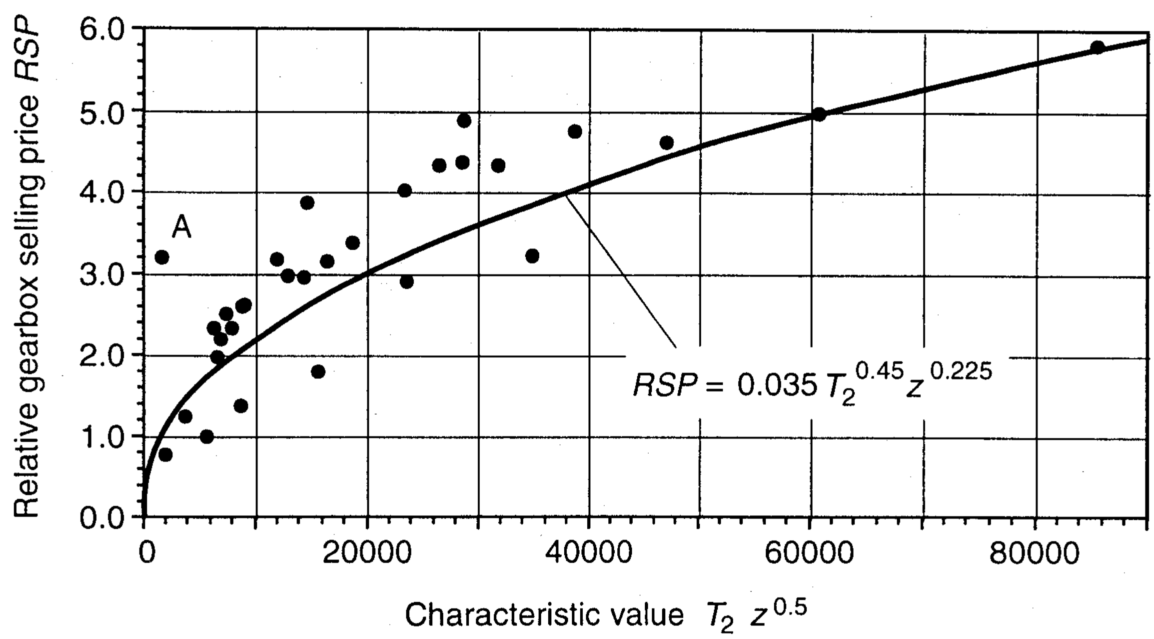


Figure 2.30. Trend of gearbox selling price *RSP* as a function of the output torque T_2 at the output shaft and the number of speeds z. Gearbox *A* is a "rogue value"; the level of demand meant that an increased marketing price could be achieved

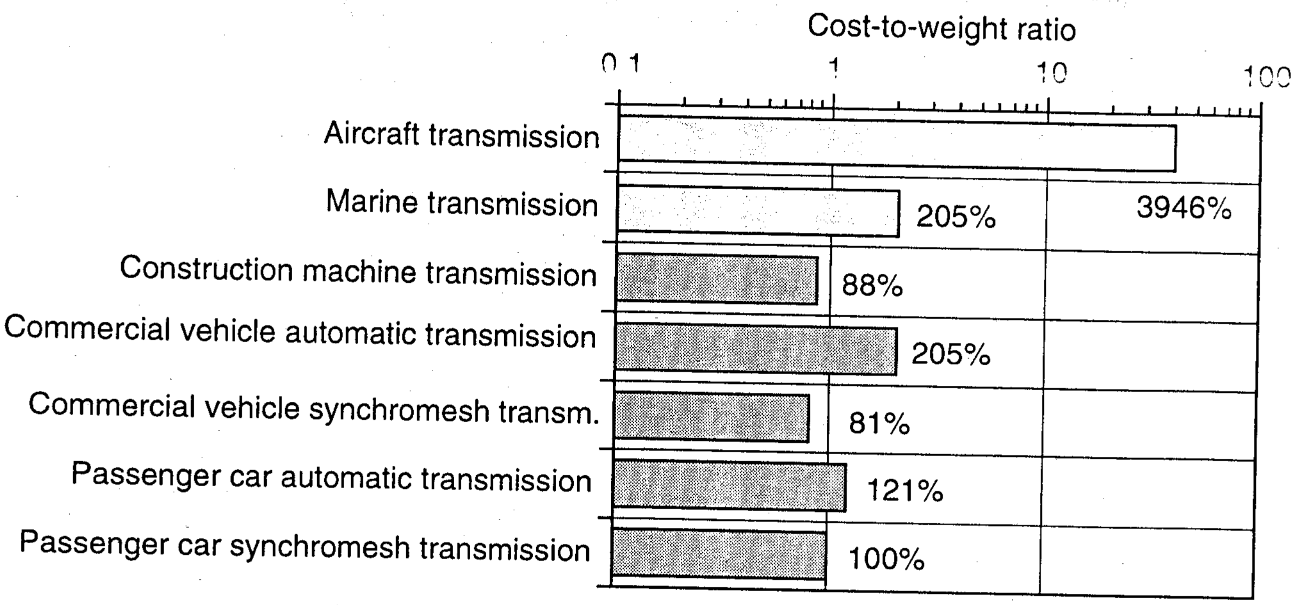


Figure 2.31. Relative weight costs for various types of vehicle transmission. 1 ≡ 100% correspond to 15 Euro/kg at 1997 prices

Commercial vehicle automatic transmissions are a different case. The reliability requirements with difficult load profiles and low quantities lead to high weight costs. Aircraft transmissions have by far the highest weight costs because of their complex weight-saving design and 100% quality monitored production.

2.4.5 Transmission Noise

The transmission often generates a high proportion of noise, being the main link in the engine/power-train/road-wheel chain. In addition to the general broadband noise, the meshing frequencies of gearwheel transmissions are very disturbing because of their discrete sounds. Primary noise reduction is therefore necessary at the design stage, the gear specification stage, and during production, combined with quality assurance (see Section 7.5 "Developing Low-Noise Transmissions").

Because of its importance, transmission noise is subject to strict regulations (see Section 7.3), which are likely to become more stringent.

2.4.6 Gearbox Losses and Efficiency

As part of the effort to save energy, the question of efficiency, and hence of gearbox friction losses, has received more attention. No mechanism converts torque as effectively as a gear pair in terms of production cost, converter ratio and efficiency.

Nevertheless the requirement is to determine and if possible improve the efficiency of vehicle transmissions as a function of torque, speed and other characteristic values. The overall power loss of a transmission is made up of various load-dependent and load-independent elements (Figure 2.32. See also Section 3.1.7 "Efficiency Map", especially Figure 3.5).

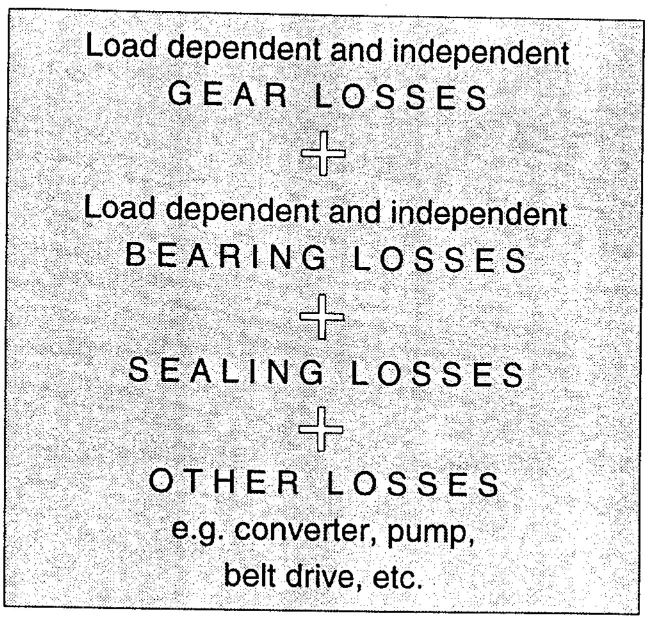


Figure 2.32.
Composition of losses in vehicle gearboxes

Gearwheel-based vehicle transmissions are the most efficient of all torque/speed converters (Table 2.11), and have the best power/weight ratio. This is also why single- and multi-speed geared transmissions, of both single- and multi-range design, have been successful for vehicles. Figure 2.33 shows the high efficiencies achieved with this transmission principle using the example of a mechanical vehicle transmission of double-range design. The use of hydrostatic transmissions, torque converters or continuously variable transmissions based on the pulley or frictional wheel principle generally leads to lower efficiencies.

Table 2.11. Reference values for the efficiency ranges of gearwheels and vehicle gearboxes

Type of gearbox		η **in %**
Gear pair	Spur gear	99.0–99.8
	Bevel gear	90–93
Mechanical transmission with splash lubrication	Passenger car	92–97
	Commercial vehicle	90–97
Conventional automatic transmission with various gear ratios, with torque converter and lock-up clutch		90–95
Mechanical continuously variable transmission	Application force not controlled by power requirement	70–80
	Application force controlled by power requirement	80–86
Hydrostatic continuously variable transmission without power split and mechanical component		80–86

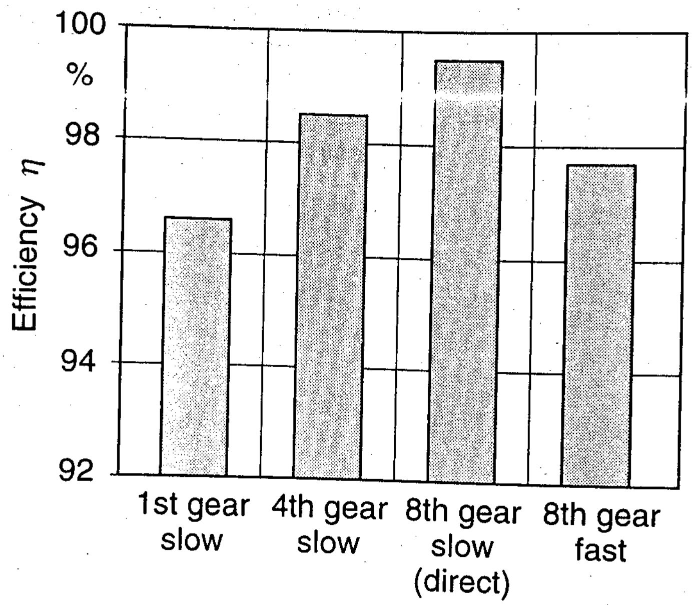


Figure 2.33. Efficiency of a mechanical commercial vehicle gearbox for various speeds where n_1 = 1500 1/min and T_1 = 500–2000 Nm; calculation as described in [2.21]

2.5 Transmission Design Trends

The range of transmission designs has become much more diverse since around 1975, especially in the case of passenger car transmissions. 4-speed synchromesh gearboxes were standard, as were conventional 3-speed automatic transmissions with a small market share of approximately 15% in Europe. The Figures 2.34 and 2.35 show the variety of emergent designs.

The 5-speed synchromesh gearbox is now standard. 4-speed and more recently 5-speed automatic transmissions will increase their market shares somewhat. But they will encounter increasing competition from continuously variable transmissions, which are to be available from 1995 in mid-size vehicles. But this is conditional on continuously variable transmissions really delivering the fuel economy they promise.

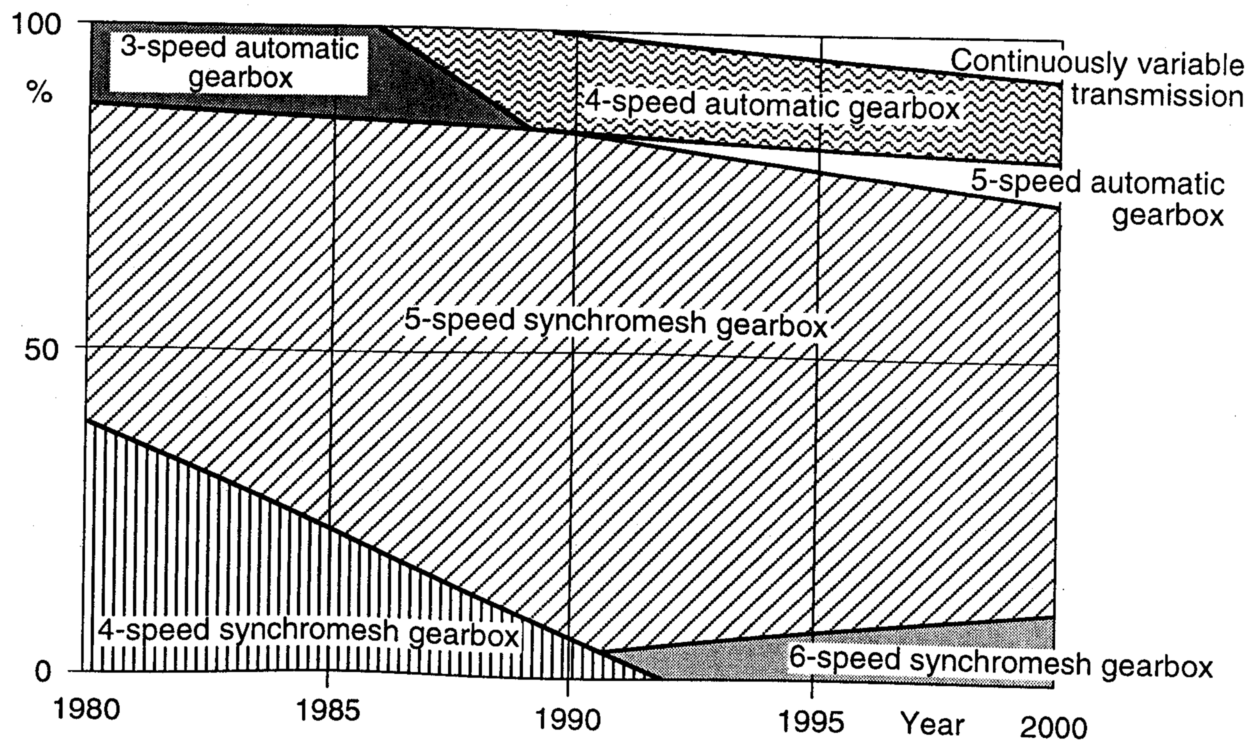


Figure 2.34. Trend in the use of passenger car transmissions in Europe [2.8]

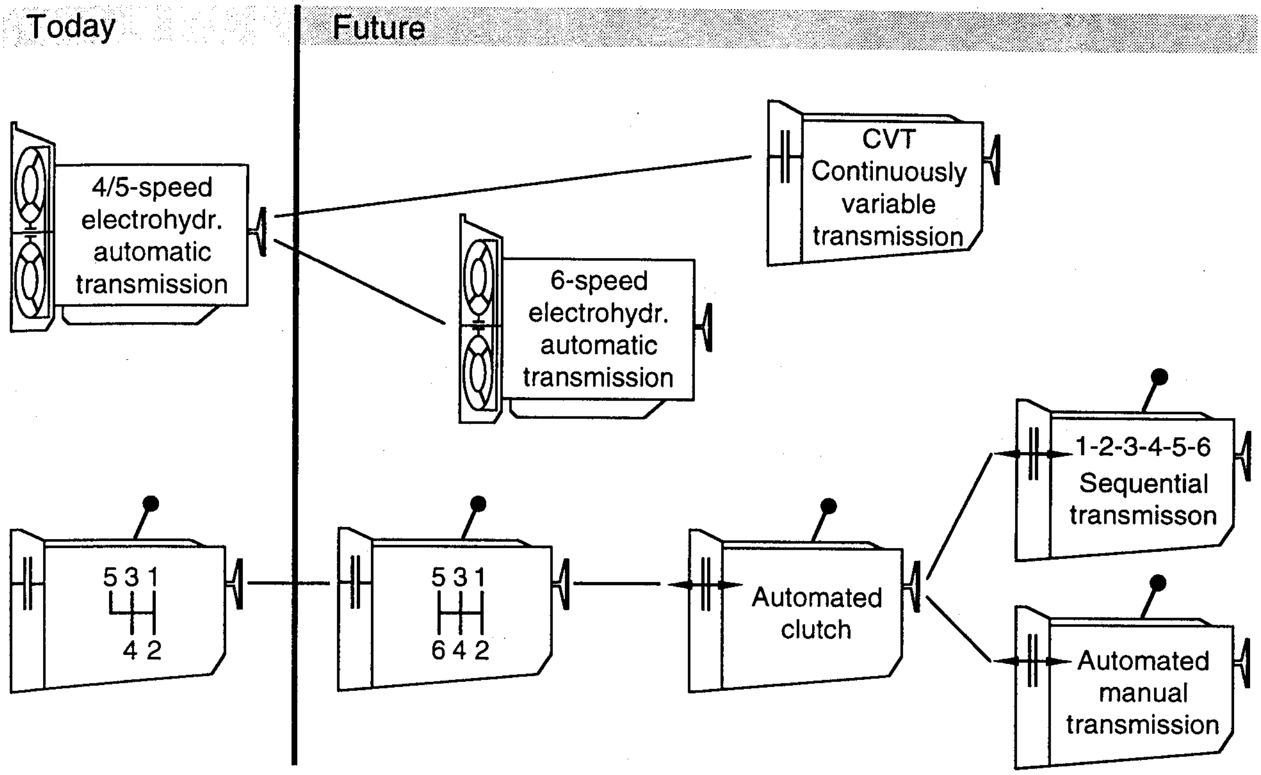


Figure 2.35. Passenger car transmissions today and in future

The problem with automatic and continuously variable transmission remains that of broad acceptance by customers. Most drivers still prefer personal control, being actively involved in the process. 6-speed synchromesh gearboxes will only occupy a niche market for sports cars and high powered Diesel cars. As average speeds on highways fall, such manual transmissions are not very appropriate, especially since the frequency of gear changing will increase.

Electronic clutch controls for dry clutches are currently about to enter mass production. They can be combined with conventional selector gearboxes. Automated multi-speed gearboxes (8-speed transmissions of double range construction) may provide an alternative to continuously variable transmissions for passenger cars.

Mechanical gearboxes with five to sixteen gears of single- or multi-range design will remain the standard for commercial vehicles, and will increasingly be available in semi-automatic or fully automatic form to reduce driver stress. The shifting process is already semi-automatic. Automating moving off with automated dry clutches instead of torque converters is difficult to achieve with commercial vehicles, but the problem has in principle been solved. Conventional automatic transmissions with torque converter, lock-up clutch and planetary gear-sets have only established themselves in buses, which will remain their domain. The success of hydrostatic automatic transmissions for commercial vehicles and electrical drives for buses remains to be seen.

3 Mediating the Power Flow

Reciprocity: Supply and Demand

Vehicle transmissions mediate between the engine and the drive wheels. The transmission adapts the power output to the power requirement by converting torque and rotational speed. The power requirement at the drive wheels is determined by the driving resistance [3.1, 3.2].

3.1 Power Requirement

The anticipated driving resistance is an important variable when designing vehicle transmissions. Driving resistance is made up of

- wheel resistance F_R,
- air resistance F_L,
- gradient resistance F_{St}, and
- acceleration resistance F_a.

3.1.1 Wheel Resistance

Wheel resistance comprises the resisting forces acting on the rolling wheel. It is made up of rolling resistance, road surface resistance and slip resistance.

Figure 3.1 shows the forces and torques acting on the wheel. The integral of the pressure distribution over the tyre contact area gives the reaction force R. It is the same as the wheel load G_R. Because of the asymmetrical pressure distribution in the wheel contact area of the rolling wheel, the point of application of the reaction force R is located in front of the wheel axis by the amount of eccentricity e.

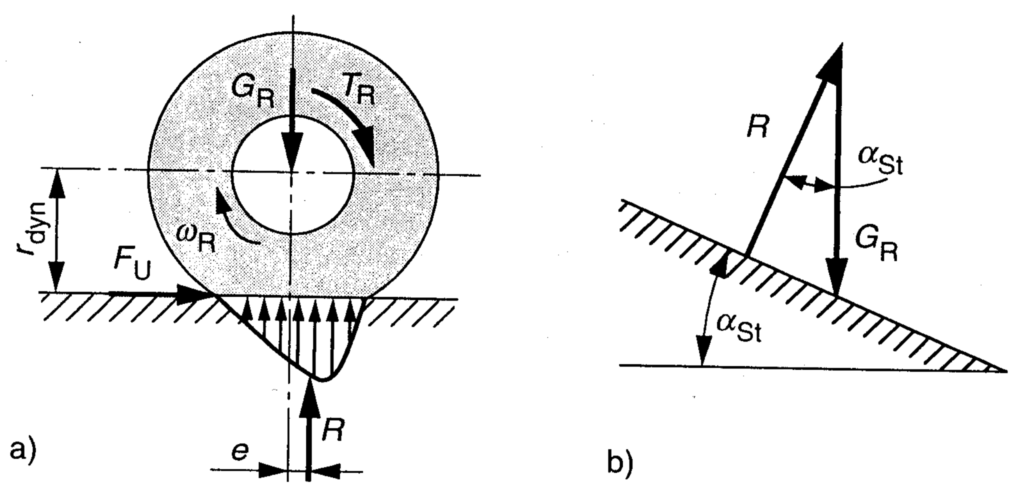


Figure 3.1. Forces and torques at the wheel. **a)** on the level; **b)** on uphill/downhill stretches

If the wheel is unaccelerated and driven by T_R, then

$$T_R = F_U\, r_{dyn} + R\, e \ . \tag{3.1}$$

For a wheel rolling without drive torque and braking torque ($T_R = 0$)

$$-F_U = \frac{e}{r_{dyn}} R \ . \tag{3.2}$$

The circumferential force $-F_U$ is equal to the rolling resistance force $F_{R,\,Roll}$ given these assumptions. On a level surface $R = G_R$, and therefore

$$F_{R,\,Roll} = \frac{e}{r_{dyn}} G_R \ . \tag{3.3}$$

Trials have revealed an almost linear correlation between the rolling resistance force $F_{R,\,Roll}$ and the wheel load G_R. The relationship is defined by the formula

$$F_{R,\,Roll} = f_R\, G_R \ . \tag{3.4}$$

The dimensionless proportionality factor f_R is designated the rolling resistance coefficient. From (3.3) and (3.4) it is given as

$$f_R = \frac{e}{r_{dyn}} \ . \tag{3.5}$$

Table 3.1 shows standard values for rolling resistance coefficients both on and off-road. Rolling resistance is chiefly a function of ground speed, wheel load, tyre pressure and tyre type. The influence of speed can be described according to [3.1] by the following polynomial: $f_R = A_0 + A_1\, v + A_2\, v^4$.

Since driving resistance calculations normally assume straight running on a dry surface, and rolling resistance is anyway the dominant wheel resistance, wheel resistance is normally assumed to be equal to rolling resistance. The following formula then applies:

$$F_R = F_{R,\,Roll} \ . \tag{3.6}$$

Table 3.1. Reference values for the rolling resistance coefficient f_R. For road speeds below 60 km/h, f_R can be assumed to be constant. (See also Table 5.1)

Road surface	Rolling resistance coefficient f_R
Firm road surface	
Smooth tarmac road	0.010
Smooth concrete road	0.011
Rough, good concrete surface	0.014
Good stone paving	0.020
Bad, worn road surface	0.035
Unmade road surface	
Very good earth tracks	0.045
Bad earth tracks	0.160
Loose sand	0.150–0.300

When travelling up gradients/down slopes at an angle of α_{St} (Figure 3.1b), then

$$R = G_R \cos \alpha_{St} \, . \tag{3.7}$$

For the whole vehicle with a mass m_F, the wheel resistance F_R, which is considered equal to the rolling resistance, is thus given by

$$F_R = f_R \; m_F \; g \cos \alpha_{St} \; . \tag{3.8}$$

In the lower speed range, the rolling resistance coefficient can be regarded as a constant at the first approximation. The angle of inclination α_{St} can be ignored on normal journeys with gradients/downhill slopes of less than 10%. With a gradient of 10% $\alpha_{St} \approx 5.7°$ and thus $\cos \alpha_{St} \approx 1$.

3.1.2 Adhesion, Dynamic Wheel Radius and Slip

There is a frictional connection between the tyres and the road surface. The transmittable circumferential force F_U, (Figure 3.1a), is proportional to the wheel load reaction force R, with a maximum value

$$F_{U,\,max} = F_{Z,\,max} = \mu_H \; R \; . \tag{3.9}$$

The maximum traction F_Z between the tyres and the road surface is constrained by the adhesion limit (Figure 2.22). See also Section 6.1.4 in relation to circumferential force, lateral force and Kamm circle. Table 3.2 gives static friction figures μ_H of pneumatic tyres on road surfaces.

Table 3.2. Static coefficient of friction μ_H of new pneumatic tyres on road surfaces [3.3]

Road speed	**Static coefficient of friction μ_H**	
km/h	Dry road surface	Wet road surface
50	0.85	0.65
90	0.80	0.60
130	0.75	0.55

For many vehicle dynamics calculations, the radius of the tyres is needed (Table 3.3). A distinction is made between

- *static wheel radius* r_{stat}
  The distance from the centre of the wheel to the datum plane with the wheel at rest,
- *dynamic wheel radius* r_{dyn}
  Calculated from the distance travelled per revolution of the wheel, rolling without slip. The dynamic wheel radius is calculated from a distance travelled at 60 km/h [3.4]. The increasing tyre slip at higher speeds roughly offsets the increase in r_{dyn}.

Table 3.3. Dynamic wheel radius of common tyre sizes [3.5]

Size	Rolling circumference (m)	r_{dyn} (m)	Size	Rolling circumference (m)	r_{dyn} (m)
Passenger cars			Passenger cars		
135 R 13	1.670	0.266	205/65 R15	1.975	0.314
145 R 13	1.725	0.275	195/60 R15	1.875	0.298
155 R 13	1.765	0.281	205/60 R15	1.910	0.304
145/70 R13	1.640	0.261	Light commercial vehicles (vans)		
155/70 R13	1.680	0.267	185 R 14	1.985	0.316
165/70 R13	1.730	0.275	215 R 14	2.100	0.334
175/70 R13	1.770	0.282	205 R 14	2.037	0.324
175 R 14	1.935	0.308	195/75 R16	2.152	0.343
185 R 14	1.985	0.316	205/75 R16	2.200	0.350
195/70 R14	1.940	0.309	Trucks/buses		
185/65 R14	1.820	0.290	12 R 22.5	3.302	0.526
185/60 R14	1.765	0.281	315/80 R 22.5	3.295	0.524
195/60 R14	1.800	0.286	295/80 R 22.5	3.215	0.512
195/70 R15	2.000	0.318	215/75 R 17.5	2.376	0.378
185/65 R15	1.895	0.302	275/70 R 22.5	2.950	0.470
195/65 R15	1.935	0.308	305/70 R 19.5	2.805	0.446

The slip between the tyres and the surface can be described as

drive slip
$$S_T = \frac{\omega_R \, r_{dyn} - v_F}{\omega_R \, r_{dyn}}, \tag{3.10}$$

brake slip
$$S_B = \frac{v_F - \omega_R \, r_{dyn}}{v_F}, \tag{3.11}$$

where v_F is the actual vehicle speed.

3.1.3 Air Resistance

Air flow around the moving vehicle, and through it for purposes of cooling and ventilation. The air resistance is made up of the pressure drag including induced drag (turbulence induced by differences in pressure), surface resistance and internal (through-flow) resistance.

The air resistance is a quadratic function of the flow rate. The flow rate v is derived from the sum of the vehicle speed v_F and the wind speed component v_W in the direction of the vehicle longitudinal axis. If the wind speed direction is the same as the direction of travel of the vehicle (following wind), then the wind speed is deducted from the vehicle speed to calculate the flow rate. Driving resistance calculations normally assume calm, in which case: $v = v_F$.

Air resistance is calculated from the product of dynamic pressure $1/2\, \rho_L\, v^2$ and the maximum vehicle cross-section A multiplied by the dimensionless drag coefficient c_W. At an air pressure of 1.013 bar, a relative air humidity of 60% and a temperature of 20 °C the air density $\rho_L = 1.199$ kg/m³.

Table 3.4. Reference values for drag coefficient c_W. In the case of goods trucks, the c_W coefficient and the maximum vehicle cross-section are very much dependent on the particular design

Vehicle	c_W	A (m²)	$c_W A$ (m²)
Motorcycle with rider	*0.5–0.7*	*0.7–0.9*	*0.4–0.6*
BMW R100S	0.59	0.79	0.47
Open convertible	*0.5–0.7*	*1.7–2.0*	*0.85–1.4*
Limousine	*0.22–0.4*	*1.7–2.3*	*0.4–0.9*
VW Golf, '92 model	0.30	2.0	0.60
Compact car "2000"	0.22	2.0	0.44
Coach	*0.4–0.8*	*6.0–10.0*	*2.4–8.0*
Kässbohrer Setra S315HD	0.45	7.4	3.33
Truck (solo)	*0.45–0.8*	*6.0–10.0*	*2.7–8.0*
Truck with trailer	*0.55–1.0*	*6.0–10.0*	*3.3–10.0*
Articulated vehicle	*0.5–0.9*	*6.0–10.0*	*3.0–9.0*

The drag coefficient c_W represents the special case of straight flow, i.e. the wind direction is in line with the longitudinal axis of the vehicle. Table 3.4 gives the c_W-values and the maximum vehicle cross-sections (projected frontal area) of some vehicles.

Drag is calculated by

$$F_L = \frac{1}{2} \rho_L \, c_W \, A \, v^2 \, . \tag{3.12}$$

The aerodynamics of vehicles with high-drag flow-impeding bodywork, such as commercial vehicles, can be greatly improved by the use of air dams.

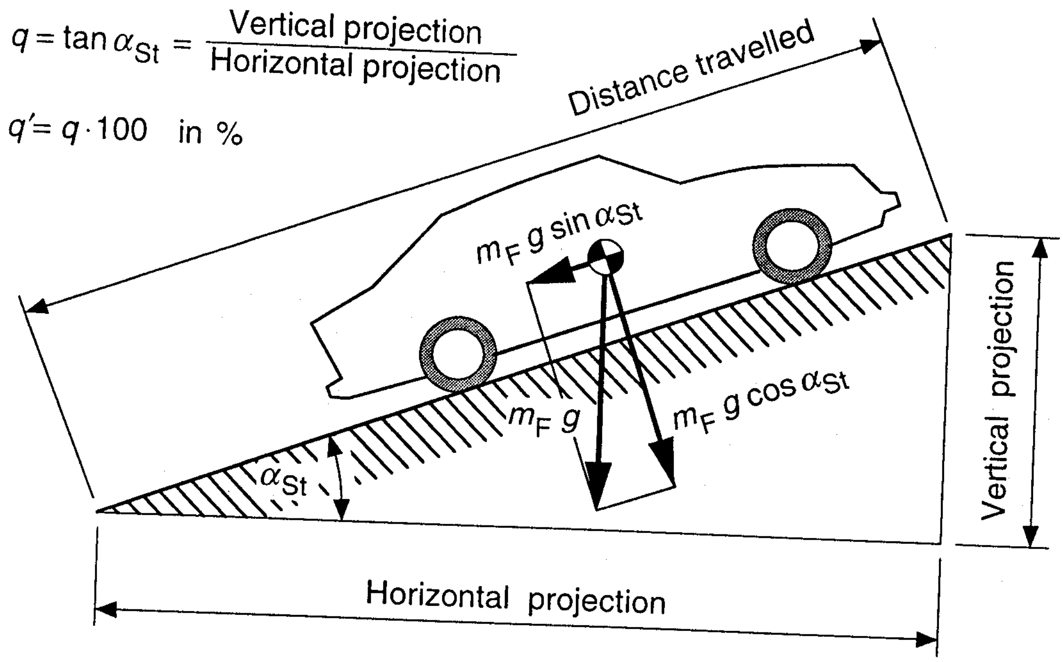


Figure 3.2. Forces acting on the vehicle travelling uphill

3.1.4 Gradient resistance

The gradient resistance or downhill force relates to the slope descending force (Figure 3.2) and is calculated from the weight acting at the centre of gravity:

$$F_{St} = m_F \, g \sin \alpha_{St} \,. \tag{3.13}$$

The road gradient q is defined as the quotient of the vertical and horizontal projections of the roadway (Figure 3.2).

When designing roads, gradients of more than 7% are normally avoided. Except in extreme cases, the following approximation is valid

$$\sin \alpha_{St} \approx \tan \alpha_{St} = \frac{q'}{100} \,. \tag{3.14}$$

Table 3.5 shows the maximum gradients (q'_{max}) of some Alpine passes.

Table 3.5. Maximum gradients of some passes in Europe

Pass	q'_{max}	Pass	q'_{max}
Germany:		*Austria:*	
Achen pass	14%	Großglockner	12%
Oberjoch	9%	Timmelsjoch	13%
France:		Turracher Höhe	26%
Col de Braus	15%	Wurzen pass	18%
Iseran	12%	*Switzerland:*	
Italy:		Simplon	10%
Brenner highway	12%	St. Bernardino	12%
Stilfser-Joch	15%	St. Gotthard	10%

3.1.5 Acceleration Resistance

In addition to the driving resistance occurring in steady state motion (v = const), inertial forces also occur during acceleration and braking. The total mass of the vehicle m_F (translatory component) and the inertial mass of those rotating parts of the drive accelerated or braked (rotary component) are the factors influencing the resistance to acceleration:

$$F_a = m_{red,i} \, a \,, \quad \text{with} \tag{3.15}$$

$$m_{red,i} = m_F + \frac{\sum J_{red,i}}{r_{dyn}^2} \,. \tag{3.16}$$

The rotational component is a function of the gear ratio. The moment of inertia of the rotating drive elements of engine, clutch, gearbox, drive shaft, etc., including all the road wheels (even the non driven road wheels), are reduced to the driving axle.

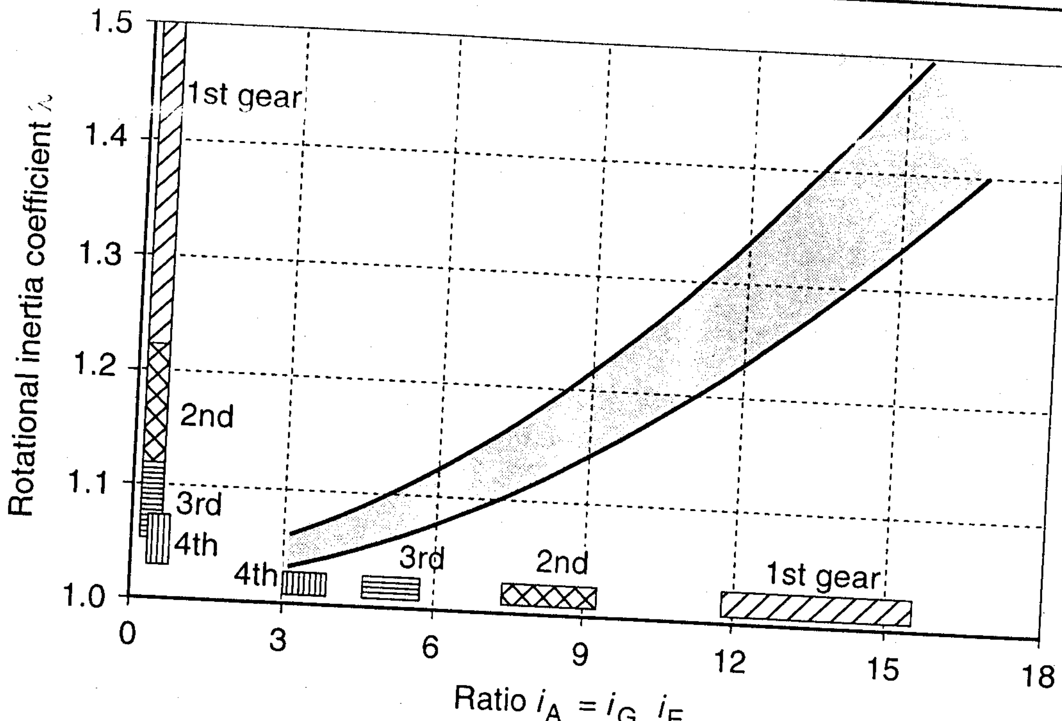


Figure 3.3. Reference values for rotational inertia coefficients of passenger cars [3.1, p. 36]

This reduces to $J_{\text{red, i}}$. The acceleration resistance is frequently represented in simplified form as

$$F_a = \lambda\ m_F\ a\ , \tag{3.17}$$

where λ is a rotational inertia coefficient, which expresses the proportion of the total mass that is rotary. Reference values for the rotational inertia coefficient for passenger cars are shown in Figure 3.3.

Since the gear ratio is a quadratic factor in determining the reduced moment of inertia, the rotational inertia coefficients for vehicles with high-ratio gears are widely spread. (Reference values for trucks as per [3.2] are: crawler gear: $\lambda \approx 10$, 1st gear: $\lambda \approx 3$, direct gear: $\lambda \approx 1.1$).

3.1.6 Total Driving Resistance

The traction $F_{Z,\,B}$ required at the drive wheels is made up of the driving resistance forces described above, and is defined as

$$F_{Z,\,B} = F_R + F_{St} + F_L + F_a\ . \tag{3.18}$$

Together with Equations 3.8, 3.12, 3.13 and 3.17, this may be expanded to

$$F_{Z,\,B} = m_F\ g\left(f_R\ \cos\alpha_{St} + \sin\alpha_{St}\right) + \frac{1}{2}\rho_L\ c_W\ A\ v^2 + m_F\ \lambda\ a\ . \tag{3.19}$$

With steady state motion ($a = 0$) and the approximations mentioned ($\cos\alpha_{St} \approx 1$ and $\sin\alpha_{St} \approx \tan\alpha_{St}$) this simplifies to

$$F_{Z,\,B} = m_F\ g\left(f_R + \tan\alpha_{St}\right) + \frac{1}{2}\rho_L\ c_W\ A\ v^2\ . \tag{3.20}$$

This may be used to calculate the power requirement $P_{Z,B}$

$$P_{Z,B} = F_{Z,B}\, v \ . \tag{3.21}$$

Figure 3.4 shows the individual components of driving resistance of a mid-size passenger car, and the resultant traction requirement.

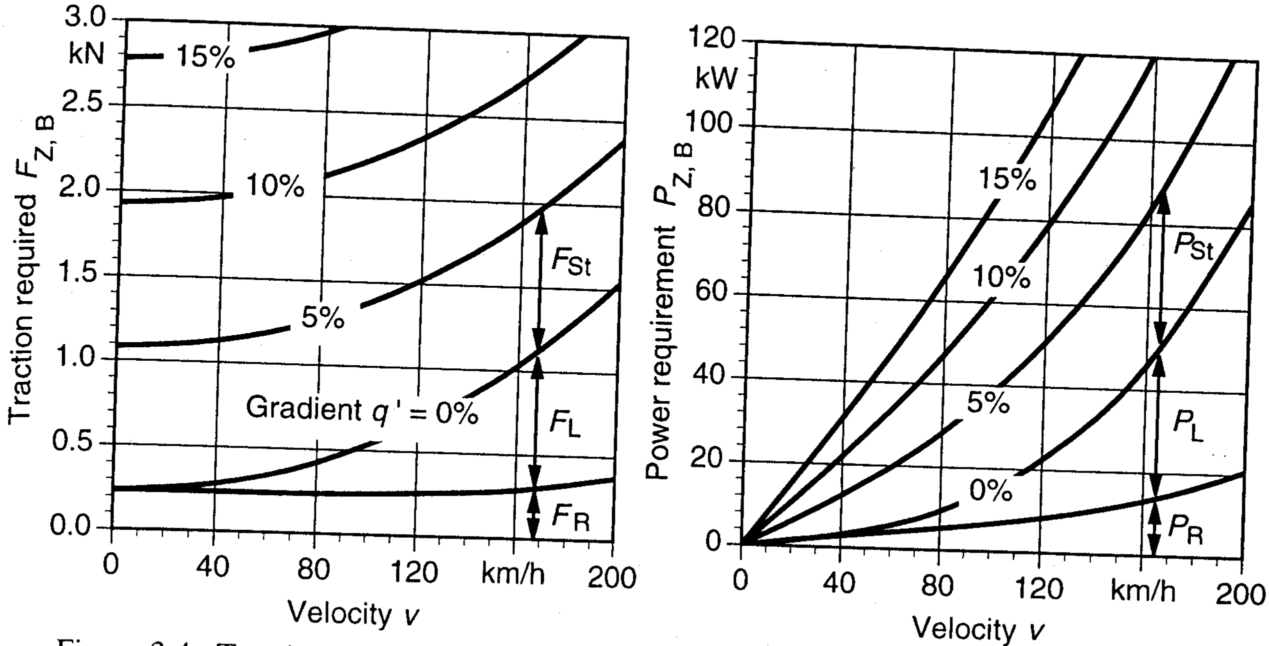


Figure 3.4. Traction required and the resultant power required for a mid-size passenger car. For vehicle data see Figure 5.3

Taking into account the power-train ratio i_A and the overall power-train efficiency η_{tot}, the traction $F_{Z,A}$ available at the wheels may be calculated from the engine characteristic curve as follows

$$F_{Z,A} = \frac{P(n_M)}{v}\,\eta_{tot} = \frac{T(n_M)\, i_A}{r_{dyn}}\,\eta_{tot} \ . \tag{3.22}$$

The traction required and the traction available for a vehicle are shown in a "traction diagram". The traction diagram is discussed in detail in Section 5.1 "Traction Diagram".

3.1.7 Efficiency Map

The efficiency of the engine and power train affect fuel consumption, emissions and driving performance. The engine efficiency is represented by the specific fuel-consumption curves. See also Section 3.3.3 "Consumption Map". Reference values for the amount of loss with different designs of vehicle transmission are given in Section 2.4.6 "Gearbox Losses and Efficiency".

To calculate the traction available or the point at which to operate an engine, it is necessary to know the power-train efficiency η_{tot} from the engine output shaft through to the driving wheels. In a sense the power-train efficiency represents another driving resistance. It is made up of the efficiencies

$$\eta = \frac{P_2}{P_1} = 1 - \frac{P_V}{P_1} \ , \tag{3.23}$$

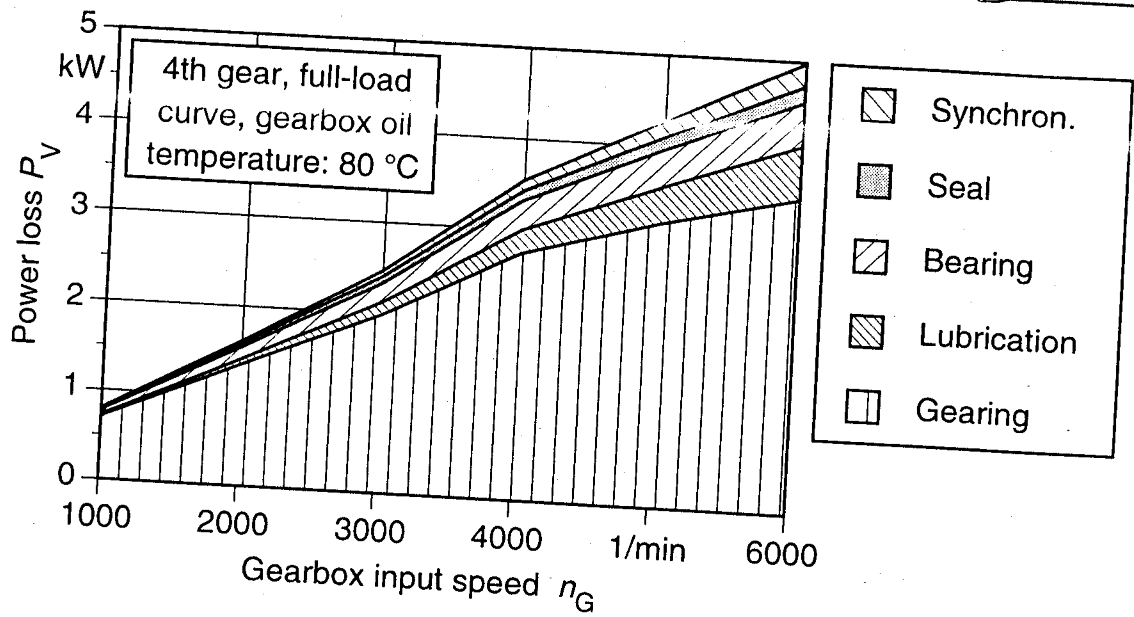


Figure 3.5. Distribution and size of power losses of a coaxial 5-speed manual gearbox in 4th gear (direct drive) at maximum engine torque [3.6]

or equivalently the power losses P_V of the individual components of the power train (Figure 3.5):

- *gearing losses:*
  - friction losses, load-dependent,
  - churning and compressing losses attributable to splash lubrication, unrelated to load,
- *bearing losses:*
  - friction losses, load-dependent,
  - lubrication losses, unrelated to load,
- *sealing losses:*
  - friction losses caused by radial shaft seals at shaft exits,
  - friction losses caused by piston rings used to keep oil under pressure at the shift elements,
- *synchronising losses:*
  - fluid friction between synchroniser ring and friction taper,
- *clutch losses:*
  - fluid friction with wet running, multi-disc clutches and brakes in automatic gearboxes and automated manual gearboxes,
- *torque converter losses:*
  - losses in the torque converter,
- *auxiliary units:*
  - power to drive auxiliary units.

The main losses involved relate to the following components of the power train:

- moving-off element, e.g. torque converter,
- selector gearbox, e.g. gearwheel transmission, pulley transmission,
- final drive,
- auxiliary units, e.g. steering pump, oil pump in automatic gearboxes, air conditioning system, variable displacement pump in continuously variable transmissions.

A further distinction is made between losses that are

- ❍ dependent on the input speed and the input torque,
- ❍ dependent only on the engine speed, including in particular pumps directly driven by the engine,
- ❍ almost independent of rotational speed and torque. For example the efficiency of the final drive is normally assumed to be constant.

Figure 3.6 shows the efficiency of the power train from the engine output shaft through to the drive wheels in fourth gear of a 5-speed manual gearbox. Provision is made for a steering pump as an ancillary unit. The level of efficiency only declines rapidly in the low load area.

In the case of manual gearboxes, a constant level of efficiency can often be assumed with sufficient accuracy. In the case of continuously variable transmissions, the part-load efficiency is significantly worse, and the drag torque is significantly higher (Figure 3.6).

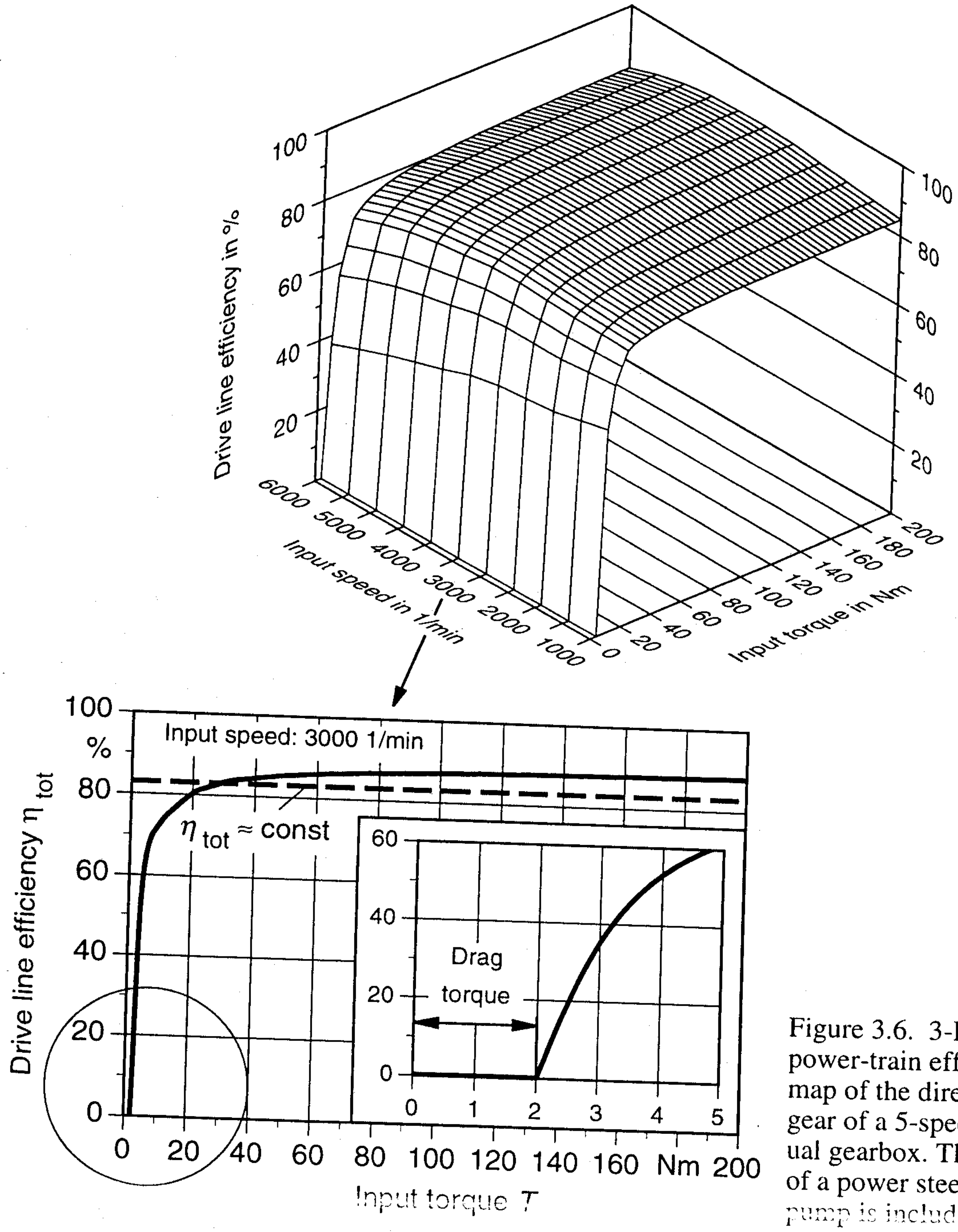


Figure 3.6. 3-D total power-train efficiency map of the direct 4th gear of a 5-speed manual gearbox. The effect of a power steering pump is included

3.2 Diversity of Prime Movers

The driving resistance described in Section 3.1 has to be overcome by the prime mover in co-operation with the other components of the power train.

The energy supply, which has to be carried in the vehicle, means "dead" weight and "dead" volume. Energy supplies with as high an energy density as possible are desirable. Figure 3.7 shows the working capacity at the drive wheels from various types of energy supply.

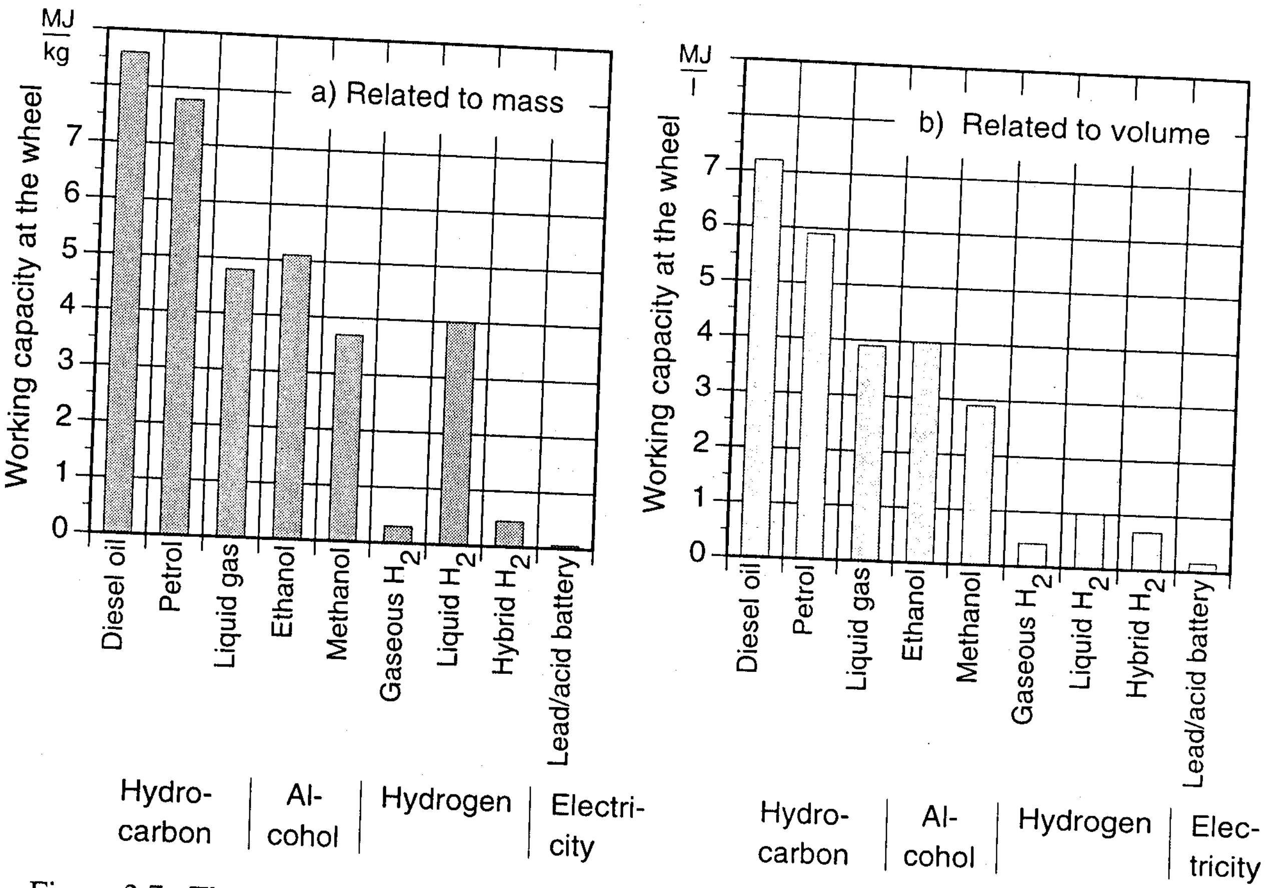


Figure 3.7. The energy available from various forms of automobile energy storing systems. Mechanical energy available at the drive wheels related to: **a)** mass and **b)** the volume, of energy supply + energy accumulator (container). Various levels of engine efficiency with energy conversion are taken into account [3.7]

The weight of the energy accumulator is factored in, as is the transmission efficiency figure (energy at the wheel/energy of the fuel). Further important criteria in selecting transportable power storage are rapid recharging of the energy accumulator and the necessary infrastructure. Diesel oil and petrol fare best in these respects. The space required to stow batteries is about 30 times greater than that required for diesel fuel or petrol with the same capacity. Fuel cells are however now approaching the capacity of diesel fuel and petrol where H_2 is extracted from methanol on the move. Source: VDI-Nachrichten, No.: 22, 30. Mai. 1997.

3.2.1 Overview

The drive system of a vehicle can be made up of a variety of combinations of components for storing energy, converting energy and converting output. The prime mover used is a crucial factor determining the assemblies and design of the associated power train.

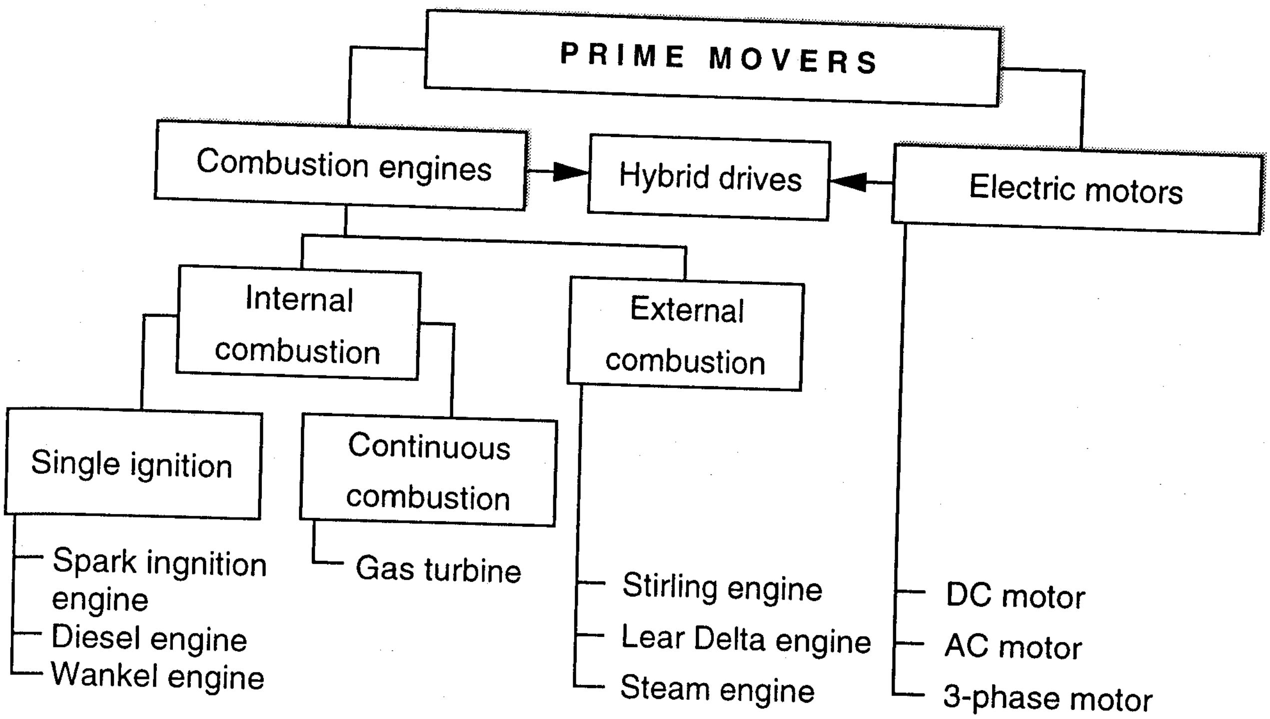


Figure 3.8. Overview of prime movers for motor vehicles; AC: alternating current; DC: direct current

Various prime movers could be used in a vehicle (Figure 3.8). They can be broken down into combustion engines and electric motors. In selecting a suitable power unit, the following factors must be considered:

- *operating performance*
  drive characteristics, ease of control, startability, energy accumulator, etc.,
- *economy*
  specific energy consumption, specific manufacturing cost, etc., and
- *environment friendliness*
  pollutant emissions, noise, vibration, etc.

In the case of the electric motor it should be noted that there is as yet no satisfactory solution to the problem of energy transport in the vehicle. The fuel cell is currently the most promising technology in prospect. The engine characteristic is a decisive technical consideration in selecting the prime mover, i.e. the power available at full load across the engine speed range.

The operation of the various prime movers is not discussed here. References are made to the literature as appropriate. In the discussion that follows, the term internal combustion engine refers to a spark ignition or diesel engine.

3.2.2 Electric Drive

The direct current (DC) motor has the ideal motor characteristic where P = const, which corresponds to the ideal traction hyperbola. It can be operated from rest, i.e. motor speed zero. 3-phase motors (synchronous and induction type) offer the advantage of being smaller and lighter than DC motors, making the preferred choice for automotive applications. Figure 3.9 shows the components of an electric drive. It is not essential to have a speed/torque converter.

The use of passenger cars with exclusively electric drive will remain restricted to cities and their environs because of their limited range and performance, and the battery charging time required.

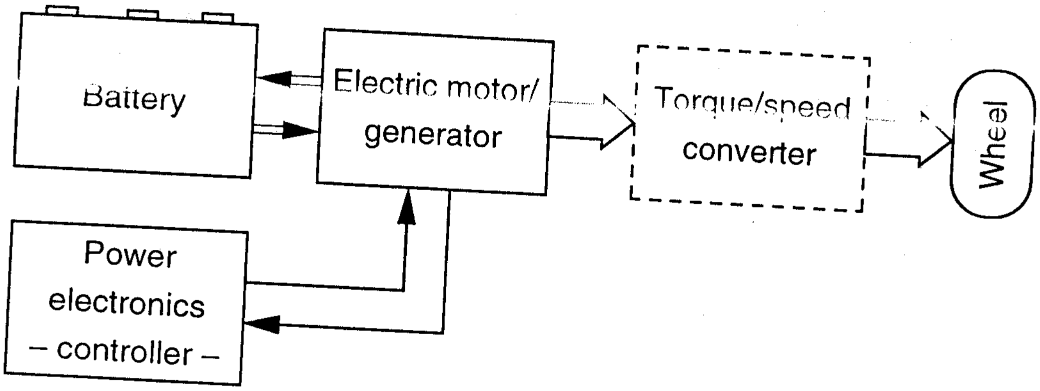


Figure 3.9. Components of an electric drive

Electric vehicles are therefore likely merely to complement conventional vehicles, and therefore account for at the most 10% of the total number of passenger cars [3.8].

3.2.3 Hybrid Drive

Hybrid drives are drives that have at least two different prime movers and energy accumulators [3.2]. Possible energy accumulators are

- chemical energy accumulator: – conventional fuel tank,
- electrical energy accumulator: – battery,
- mechanical energy accumulator: – flywheel,
  – hydraulic accumulator.

Mechanical energy accumulators have up to now been used principally to assist moving off. They are much used in vehicles with a high proportion of stop and go driving, for example city buses, where recuperated braking energy is used. But there are designs under development, in which flywheels running in a vacuum on magnetic bearings act as the energy accumulator for emission-free operation [3.9]. The mechanical energy of the flywheel or gyro store is converted into electrical energy and feeds the electric motor.

In hybrid drives, numerous drive combinations are possible. But they all have the problem of additional weight.

Internal Combustion Engine + Electric Drive

The combination of combustion engine plus electric drive gives better range and availability than a vehicle with electric drive only. In conurbations, driving can be emission-free. Vehicles with hybrid drive are less energy-efficient than vehicles with just one type of drive, because of their additional weight.

Regenerative braking is used to offset the increased energy consumption resulting from higher driving resistance.

In order to keep the vehicle weight within bounds, "savings" have to be made with both types of drive; compromises are necessary. It has to be decided whether there should be a preferred type of drive, and which it should be. With hybrid drives having both a combustion engine and an electric motor, a distinction is made between

- *serial hybrid power train (Figure 3.10):*
  - no mechanical coupling of combustion engine and wheels,
  - no mechanical transmission required,
  - two electric units are needed (generator + generator/electric motor),

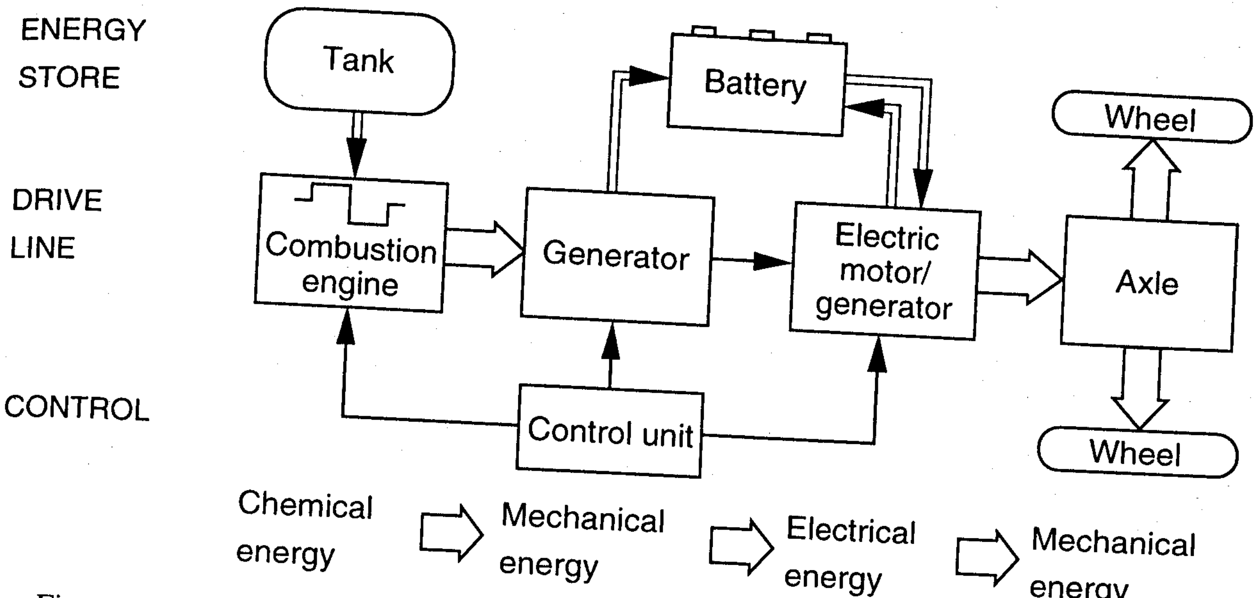


Figure 3.10. Serial hybrid power train. Source: HÖHN

❍ *parallel hybrid power train (Figure 3.11):*
- both drive sources can be combined,
- mechanical transmission is necessary,
- only one electric machine required,
- the battery does not need to be recharged from the mains. When the internal combustion engine is being used, it can run at optimum efficiency. If the specific power output of the engine is higher than that required to overcome the driving resistance, the excess power can be used to charge the battery. But note that the conversion losses during charging have to be taken into account.

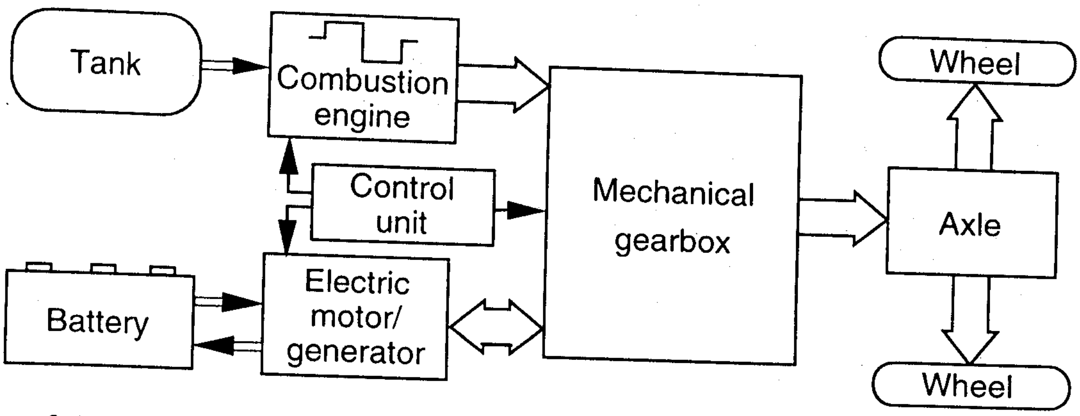


Figure 3.11. Parallel hybrid power train. Source: HÖHN

3.3 Power Output, Combustion Engine Characteristic

Internal combustion engines based on the spark ignition and diesel principle will retain their dominant position in automotive engineering for the foreseeable future. Spark ignition engines are usually used in passenger cars. The key features are the high power/weight ratio, good performance and low combustion noise. The disadvantages are the quality of fuel required and the high part-load consumption.

The economy of the diesel engine is based on its low consumption, especially in the part load range, its low maintenance requirement (no ignition system), the low fuel quality required and its good gaseous emissions ratings. The disadvantages are the level of particulate emissions, noise, irregular running, the lower engine speed spread (n_{max}/n_{min}), the low power output per litre, and the resultant greater weight and higher price. The higher capital cost means that it only becomes economical in vehicles which do a high mileage. Almost all commercial vehicles use diesel engines.

3.3.1 Torque/Engine Speed Characteristic

There are two typical characteristic curves to describe the engine characteristic of combustion engines. One is the torque/engine speed curve at full load (100% accelerator pedal position) and the other is the corresponding full-load power curve (engine characteristic). Figure 3.12 shows the map of an internal combustion engine and the characteristic points of the full load characteristic curve. The maximum braking torque (0% accelerator pedal position) increases almost linearly with engine speed to a maximum of approx. 30% of the nominal torque T_n.

Various measures are used to facilitate comparison of different engines. Key variables are torque increase (torque elasticity)

$$\tau = \frac{T_{max}}{T_n} \tag{3.24}$$

and the engine speed ratio (engine speed elasticity)

$$\nu = \frac{n_n}{n(T_{max})} . \tag{3.25}$$

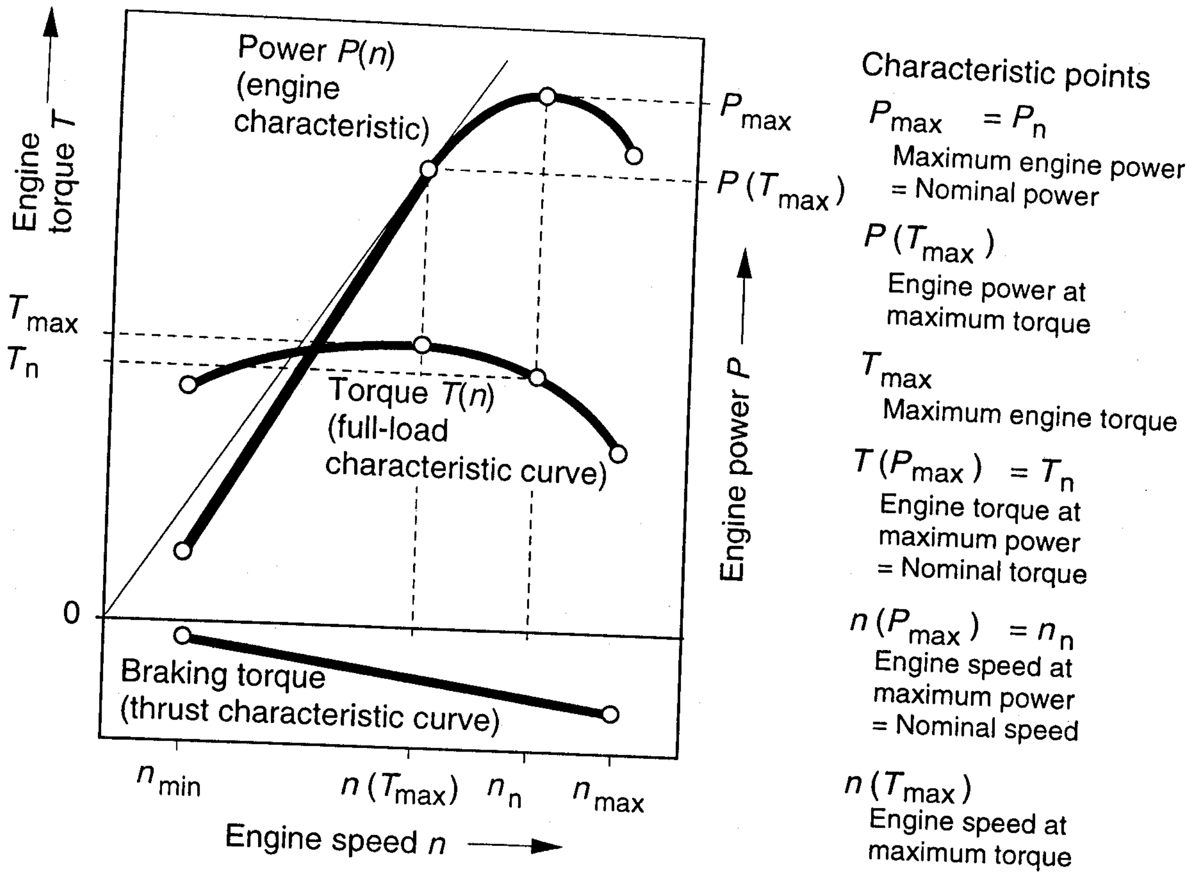


Figure 3.12. Characteristic curves of an internal combustion engine

An engine is considered to have greater elasticity the greater the product $\tau\,\nu$ is. This is apparent in the form of better engine power at low and medium engine speeds, which in turn means less frequent gear changing.

Figure 3.13 gives examples of the characteristic curves for various passenger car and commercial vehicle engines. Different engine characteristics can be achieved by varying the engine design. A distinction is made in principle between three typical characteristics (Figure 3.14).

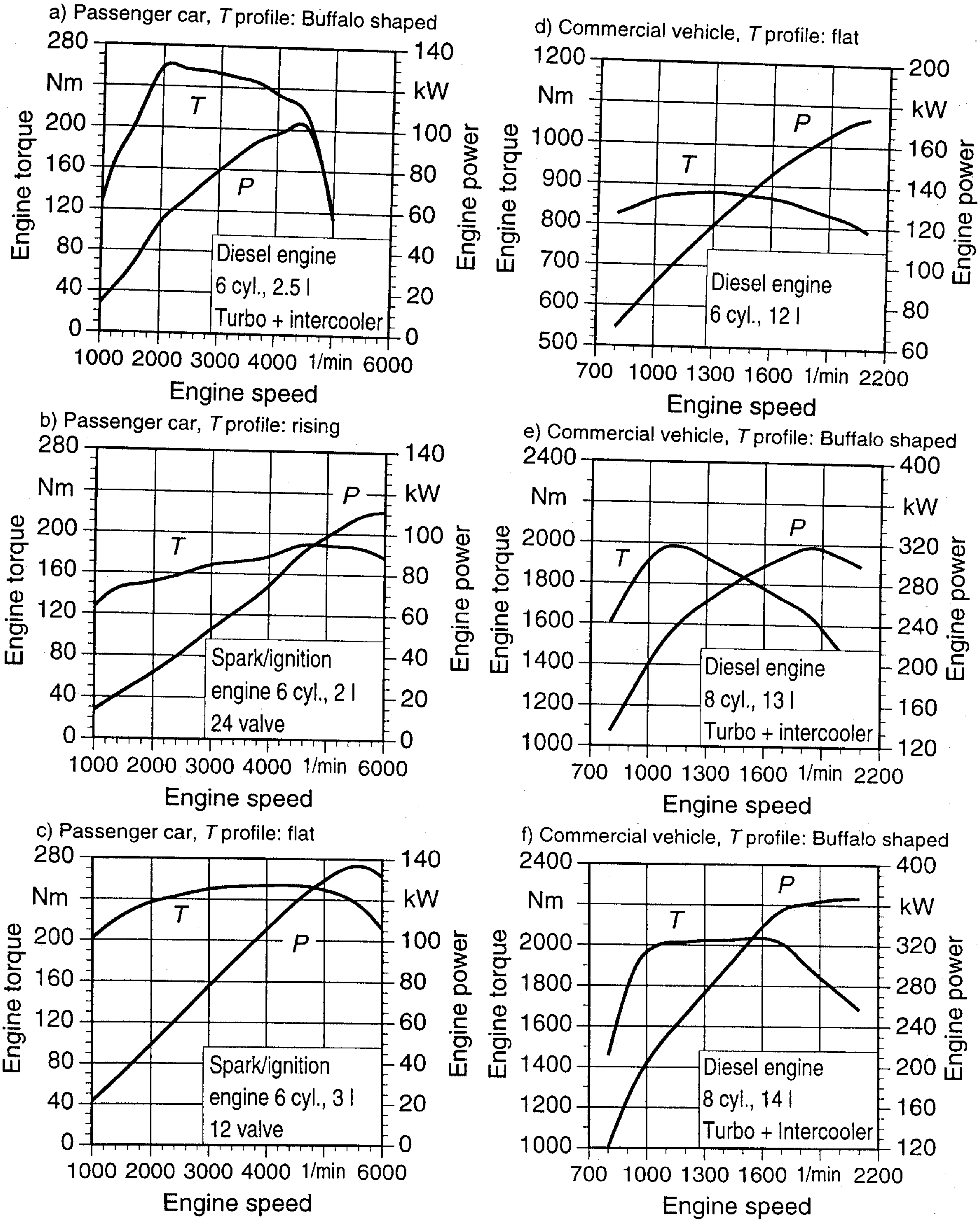


Figure 3.13. Engine diagrams of various passenger cars **a)–c)**, and commercial vehicles **d)–f)**

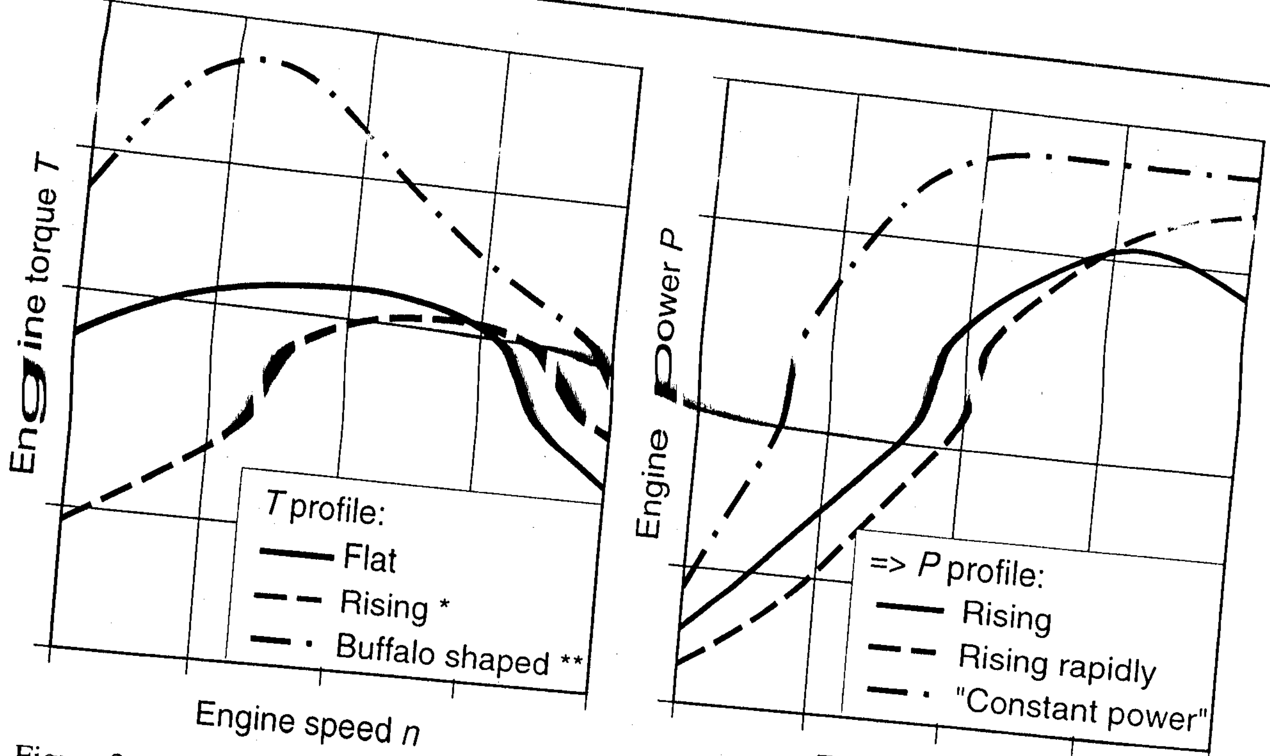


Figure 3.14. Typical engine profiles.
*) Typical of 4-valve engines. **) Typical of commercial vehicle turbo diesel engines

3.3.2 Engine Spread, Throttle Map

The spread of the engine is an important variable influencing the interaction of the internal combustion engine with the transmission (Chapter 5). In its function as a speed and torque converter, the transmission has a range of ratios: the overall gear ratio (Section 4.3.1). It is defined as the quotient of the maximum and minimum transmission ratio.

The engine spread refers to an engine's speed and torque range. Vehicles with powerful engines accordingly have a large torque spread. Diesel engines have a lower maximum speed than spark ignition engines and accordingly have a small engine speed spread. Figure 3.15 shows the characteristic diagram of two passenger car engines. The engine shown on the left is a 2.0 litre spark ignition engine with a power rating of 111 kW; the one shown on the right is a 2.5 litre turbo diesel engine with intercooler, rated at 105 kW.

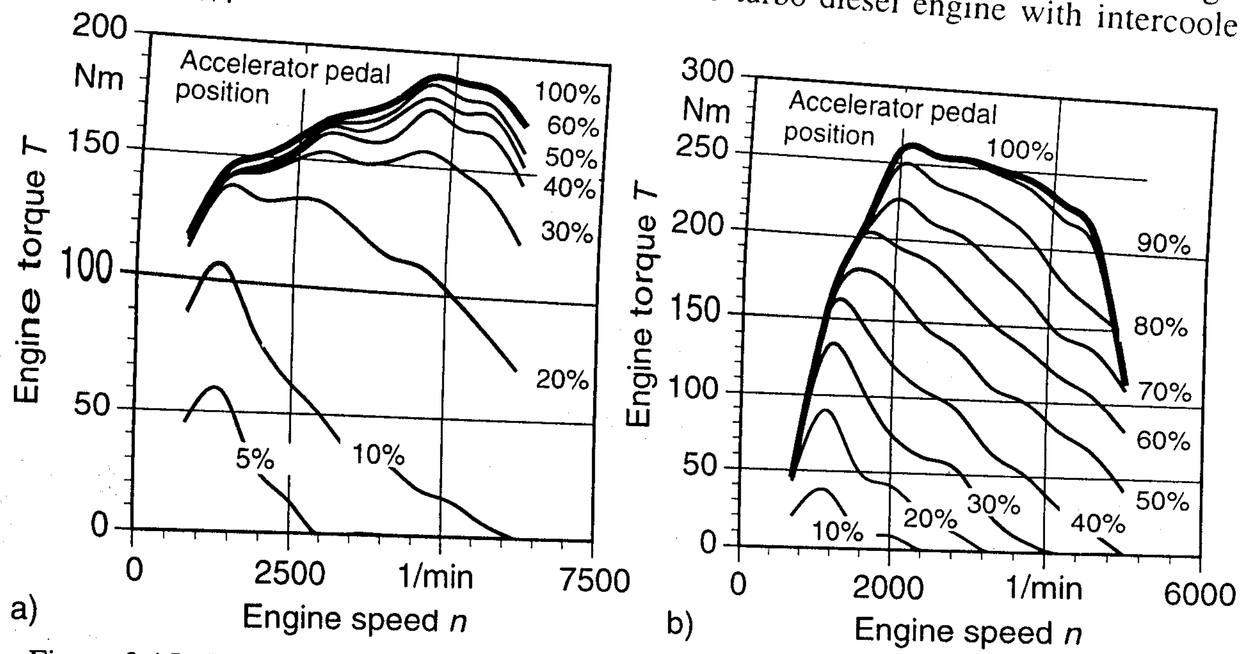


Figure 3.15. Passenger car engine performance maps, accelerator pedal position (throttle map). **a)** 2.0 litre spark-ignition engine with 111 kW, 6 cylinder, 24 valves; **b)** 2.5 litre turbo-diesel engine with intercooler and 105 kW, 6 cylinder

The diesel engine has a smaller speed spread than the spark ignition engine, but a greater torque spread. The transmission ratios have to be selected to accommodate this. The spread of the engine and the overall gear ratio (in combination with the graduations between gear ratios) are the main factors determining the functional characteristics of the vehicle.

In the displayed engine maps the driver uses the accelerator pedal to indicate the power desired from the engine. When the accelerator pedal is fully depressed (100%) this corresponds to the engine full load curve, and when the accelerator pedal is not depressed (0%), to the thrust characteristic curve. Figure 3.15 shows the lines for the same accelerator pedal position for the two engines. The almost equidistant pattern is typical of diesel engines. The term "throttle valve angle" is often used instead of "relative accelerator pedal position". A throttle valve angle of 90° then corresponds to the engine full load line. Diesel engines do not have a throttle valve for preparing the mixture, so the term "relative accelerator pedal position" or "control rack travel" is used.

3.3.3 Consumption Map

The fuel consumption of a stationary internal combustion engine can be represented as a function of engine speed and torque. A consumption characteristic diagram of this type is shown in Figure 3.16; the absolute consumption b_{abs} is shown in g/h. It increases rapidly with the engine output. If the specific consumption b_e is shown in g/kWh, the term "onion diagram" is used (Figure 3.17). There is a minimum consumption line $b_{e,\,min}$ just below the full-load characteristic curve in the lower engine speed range. The precise position depends on the engine. In the case of spark-ignition passenger car engines the minimum consumption is around 250 g/kWh, and with commercial vehicle diesel engines it is around 190 g/kWh.

In the engine map the effective average pressure p_{me} in the cylinder is often plotted instead of the engine torque. The following relationship applies

$$p_{me} = \frac{T_M \, 2\pi}{V_H \, i} \quad \text{with} \quad i = \frac{2}{\text{number of strokes}}, \tag{3.26}$$

(1 bar = 10^5 N/m^2), where V_H is the total swept volume in m^3. In four-stroke engines $i = 0.5$.

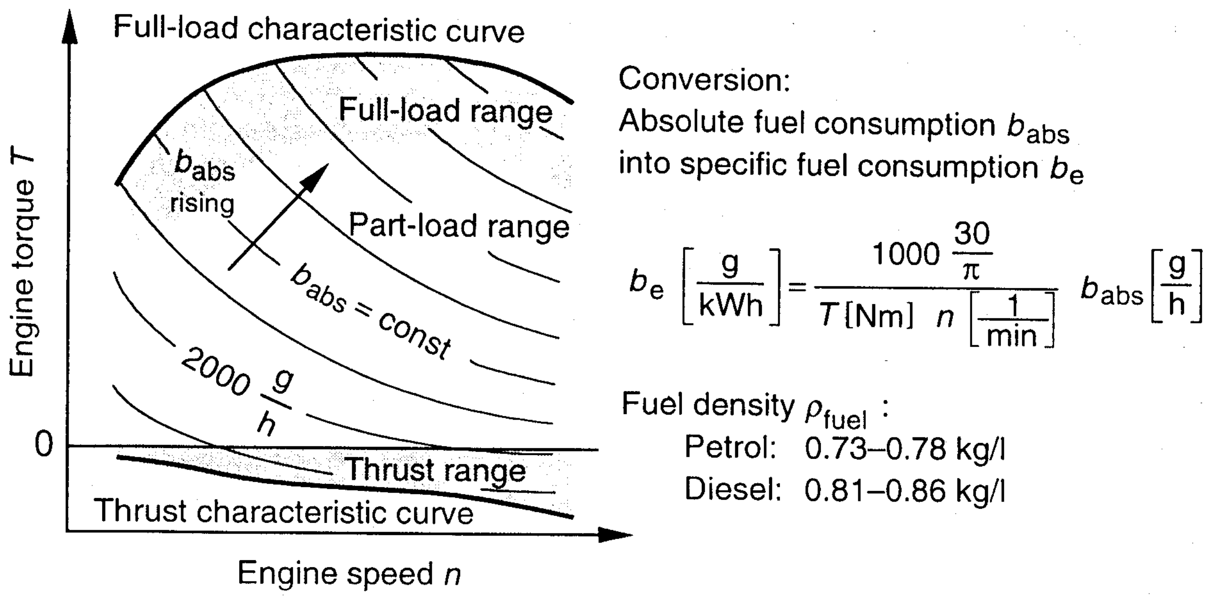


Figure 3.16. Consumption map of a spark ignition engine. Absolute consumption b_{abs} in g/h

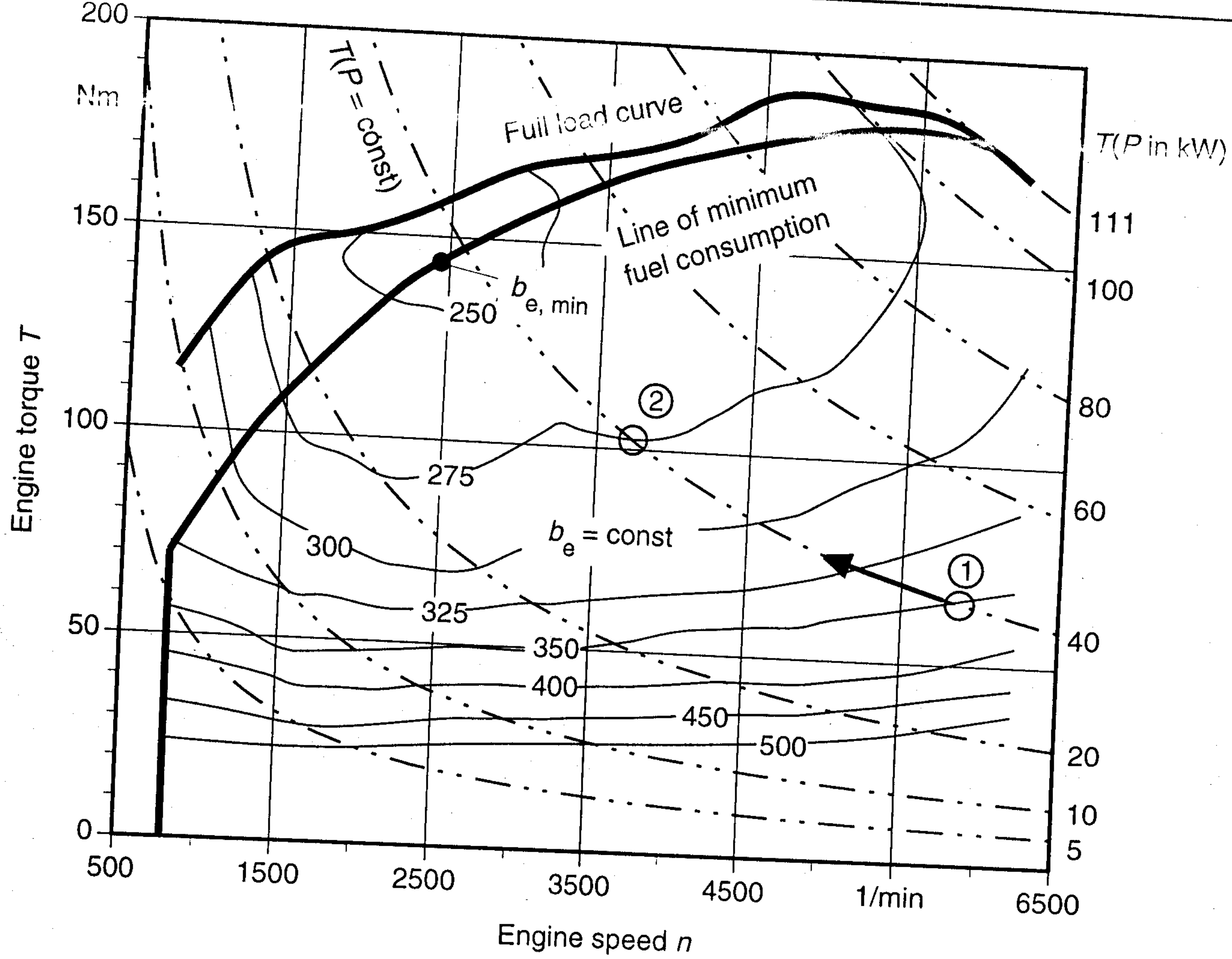


Figure 3.17. Engine map ("onion diagram"), specific fuel consumption b_e in g/kWh. Consumption map of a 2.0 litre spark ignition engine with 111 kW, 6 cylinders, 24 valves

Like the engine characteristic the consumption map is an important basis for matching engine, transmission and vehicle. The transmission exploits the fuel-efficient areas of the engine performance map. Figure 3.17 shows the contour lines of constant specific fuel consumption b_e (onion curves) as well as the torque/engine speed curves at a constant engine output (demand power hyperbola) $T(P = \text{const})$. In this way the same engine output can be achieved at different torque/engine speed values – points *1* and *2* in the engine map – and thus also with different levels of fuel consumption. A minimum fuel consumption point can be found on any power hyperbola. The curve passing through these points is the minimum fuel consumption curve.

4 Power Conversion: Selecting the Ratios

The transmission puts the engine to work

Chapter 3 dealt with the relationship between the power available from the engine and the power requirement arising from driving resistance. The torque/speed profile of the internal combustion engine is not suited to use in motor vehicles (see also Section 2.3.2 "Why do Vehicles Need Gearboxes? "). Output converters are needed for the final output to approximate as closely as possible to the ideal engine characteristic with P_{max} = const over the entire engine speed range. Clutches serve to adapt engine speed, transmissions serve to adapt both speed and torque. The conversion ratio is determined by theoretical and practical engineering constraints, which depend very much on the application.

The basic design of the transmission involves first determining the maximum and minimum ratio, i.e. the "overall gear ratio" of the transmission, and then selecting the intermediate ratios. Chapters 4 and 5 deal with the selection of these key features. They are the basis for the calculation, engineering and design of components (Figure 4.1).

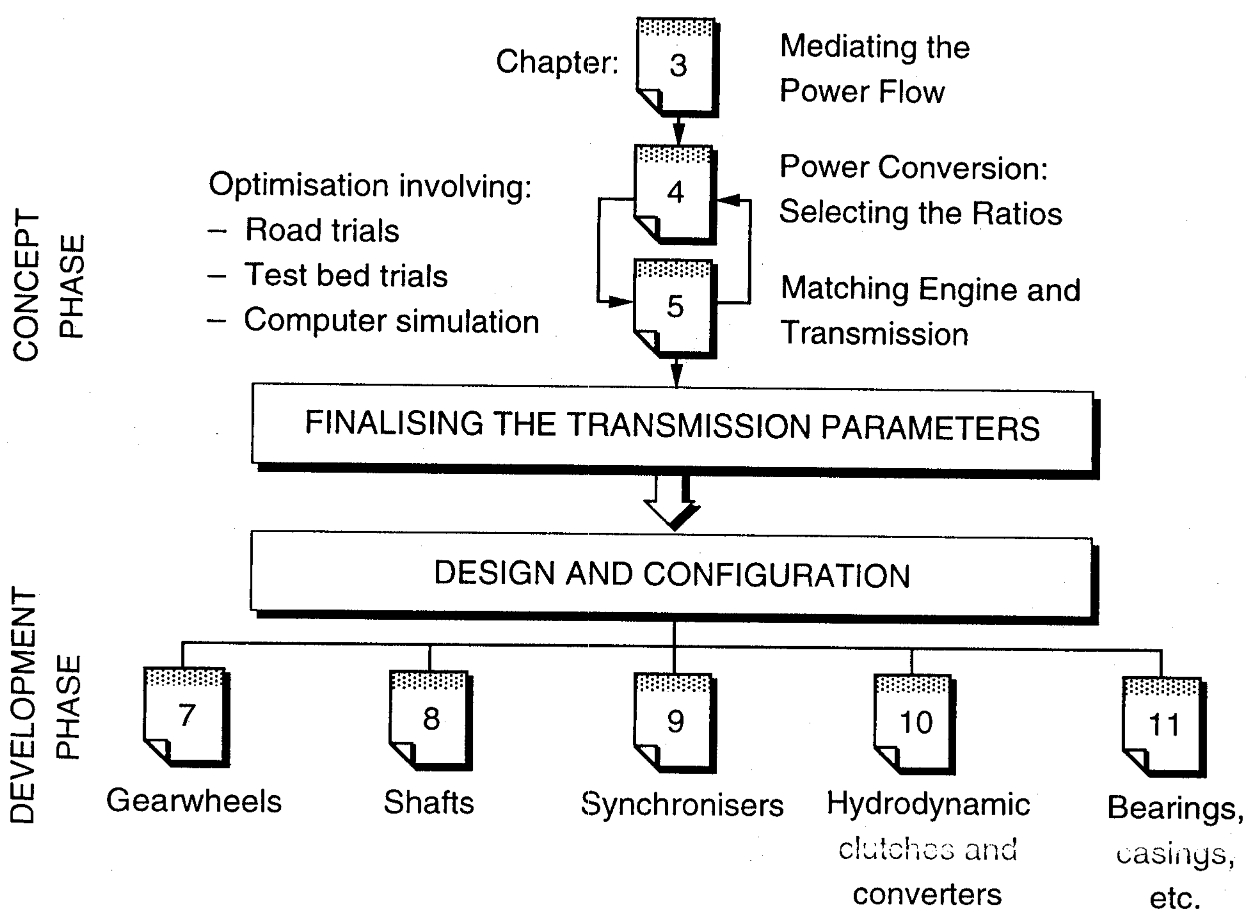


Figure 4.1. The ratios selected are the key features of the transmission, and thus form the basis for subsequent development work

4.1 Power Train

In vehicles with internal combustion engines, the output conversion between the engine and the drive wheels is achieved by the combined action of the assemblies of the power train. Figure 4.2 shows the hierarchical structure of the various ratios in the power train, starting from the total power-train ratio i_A. The total ratio of the power train is derived from the ratio i_S of the moving-off element, the ratio i_G of the transmission and the final ratio i_E.

$$i_A = i_S \, i_G \, i_E \; . \tag{4.1}$$

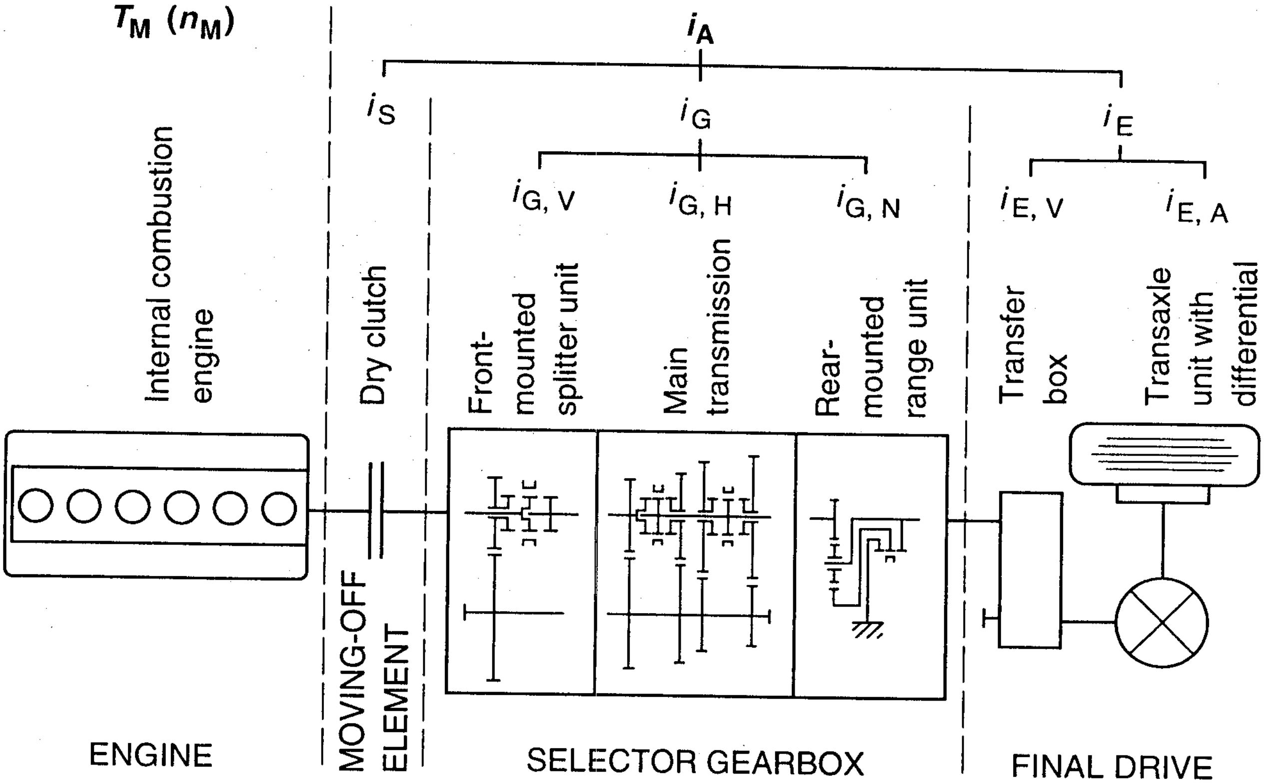


Figure 4.2. Hierarchical structure of the power-train ratio i_A using the example of a commercial vehicle with standard drive, i.e. front-mounted engine with rear-wheel drive

The ratio of output speed n_2 to input speed n_1 of a power-train component is defined as speed conversion ν,

$$\nu = \frac{n_2}{n_1} \; . \tag{4.2}$$

The torque conversion μ represents the ratio between the output torque T_2 and the input torque T_1 of a power-train component,

$$\mu = \frac{T_2}{T_1} \; . \tag{4.3}$$

A ratio of $i \neq 1.0$ should only arise when there is both speed and torque conversion. In this case

$$i = \frac{n_1}{n_2}, \quad \text{if} \quad \mu > 1.0 . \tag{4.4}$$

Master clutches convert only rotational speed, i.e. $i_S = 1.0$. Torque converters convert both rotational speed and torque, $i_S \geq 1.0$. Torque converters are discussed in Chapter 10. The discussion below is based on a dry clutch as the standard moving-off element.

The transmission ratio i_G constantly adapts the traction available from the engine in steps – or rather in an infinite number of steps – to the traction hyperbola for P_{max} = constant (see also Figure 2.22). Range transmissions are fitted on the input (i_V) or output (i_N) side to increase the number of speeds where the vehicle needs a broad overall gear ratio, such as commercial vehicles and off-road vehicles.

The balance between performance and fuel consumption is achieved by adjusting the final ratio i_E, especially important for commercial vehicles.

4.2 Speed Converter for Moving Off

Internal combustion engines have a minimum rotational speed. The speed difference between the lowest engine operating speed and the stationary transmission input shaft has to be bridged by a speed converter. Frictional engaged clutches are always used as moving-off elements. Figure 4.3 shows a systematic classification of master clutches [4.1]. The main moving-off elements to have established themselves in motor vehicles are:

- the dry clutch where $i_S = 1.0$ is standard for manual transmissions,
- the torque converter where $i_S \geq 1.0$ is standard for conventional fully automatic transmissions.

Less common is the magnetic powder clutch in which a magnetisable powder transmits the power by frictional engagement. Magnetic powder clutches, and also wet multi-disc clutches, are used as automatic clutches in continuously variable transmissions, for example.

In dry clutches the pressure force is produced by a spring. A distinction is made between coil spring clutches and membrane spring clutches depending on the type of spring used. A distinction is also made between clutches activated by pulling and by pushing.

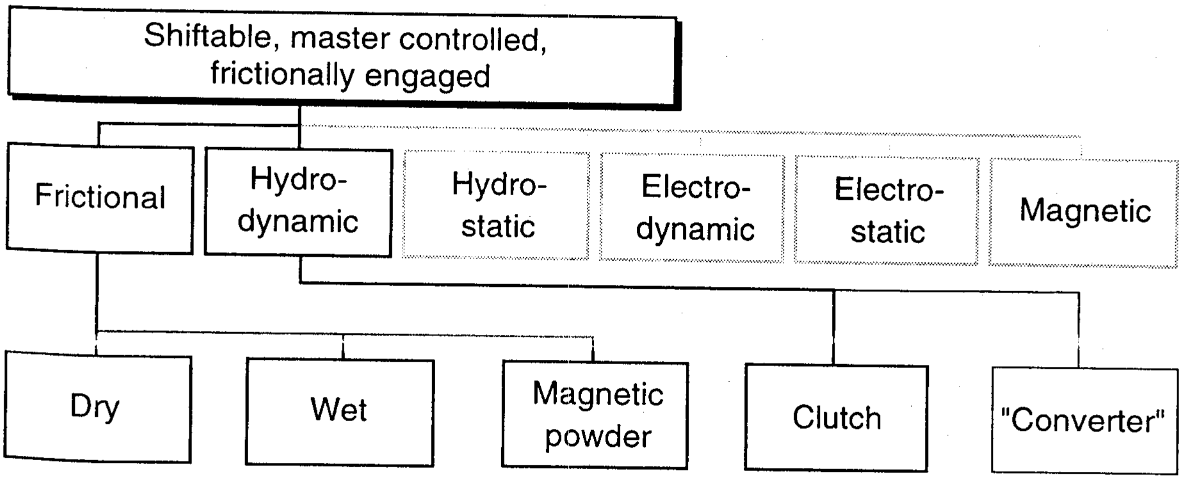


Figure 4.3 Systematic classification of master clutches on the basis of their characteristics

The characteristic features of a speed converter are (Figure 4.4):

- the output torque T_2 is equal to the input torque T_1: $T_2 = T_1$,
- the output speed n_2 is less than or equal to the input speed n_1: $n_2 \le n_1$,
- the input power P_1 is reduced by the power loss P_V: $P_2 = P_1 - P_V$.

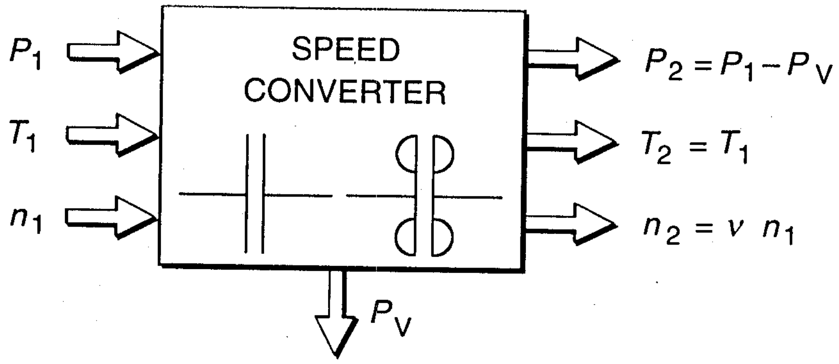


Figure 4.4. Speed converter input and output values

Figure 4.5 shows an idealised clutch operation sequence when moving off. The input and output speeds converge in the course of the clutch operation sequence. Some of the input power is converted into waste heat during the continuous slip phase. The efficiency of the clutch η_C is given by Equations 4.2 and 4.3 and is

$$\eta_C = \frac{P_2}{P_1} = \frac{T_2\ 2\pi\ n_2}{T_1\ 2\pi\ n_1} = \mu_C\ \nu_C\ , \tag{4.5a}$$

if $T_2 = T_1$, i. e. $\mu = 1$, then

$$\eta_C = \frac{n_2}{n_1} = \nu_C\ . \tag{4.5b}$$

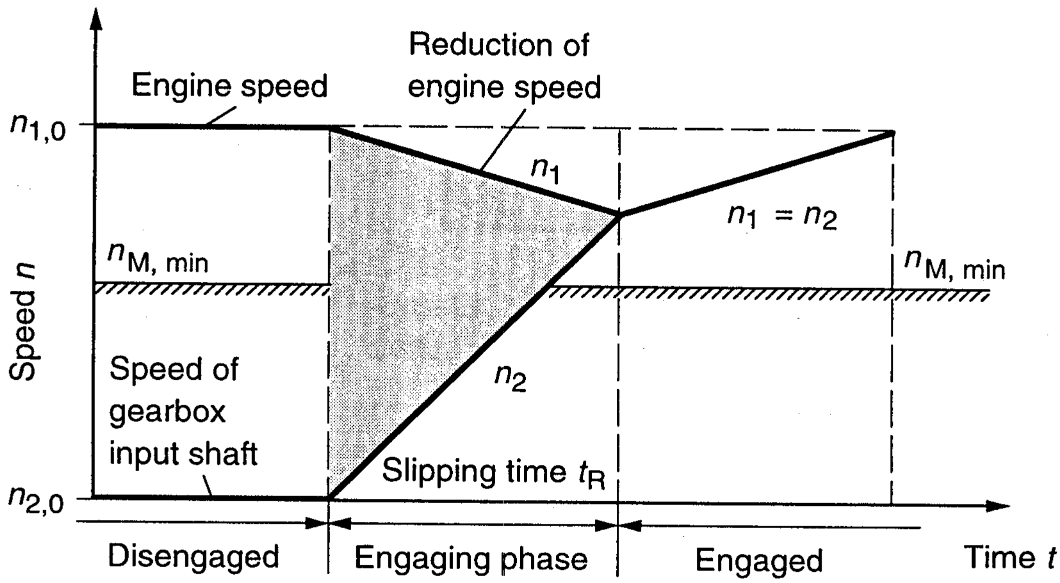


Figure 4.5. Idealised moving-off sequence with a friction clutch

The slip S is defined as the ratio of the difference between the input and output speeds to the input speed

$$S = \frac{n_1 - n_2}{n_1} . \tag{4.6}$$

Equations 4.5 and 4.6 give the following relationship between efficiency, slip and speed ratio

$$S = 1 - \eta_C = 1 - \nu_C . \tag{4.7}$$

The master clutch must be so designed that it both transmits the maximum output torque with sufficient reliability, and tolerates the thermal stress arising in repetitive "stop and go" use [4.2].

4.3 Total Ratio and Overall Gear Ratio

The power train has to offer ratios between engine speed and road wheel speed enabling the vehicle to:

- move off under difficult conditions,
- reach the required maximum speed, and
- operate in the fuel-efficient ranges of the engine performance map.

The maximum ratio required $i_{A,\,max}$ is fixed by the first condition. The second condition gives the maximum road speed ratio $i_A(v_{max,\,th})$. The smallest power-train ratio $i_{A,\,min}$ is given by the third condition. Figure 4.6 shows the speed spread of a transmission in a diagram of velocity against engine speed. The engine speed range (primary side) is "spread" by the transmission to the speed range of the secondary side. The operating range extends between the ratio boundaries.

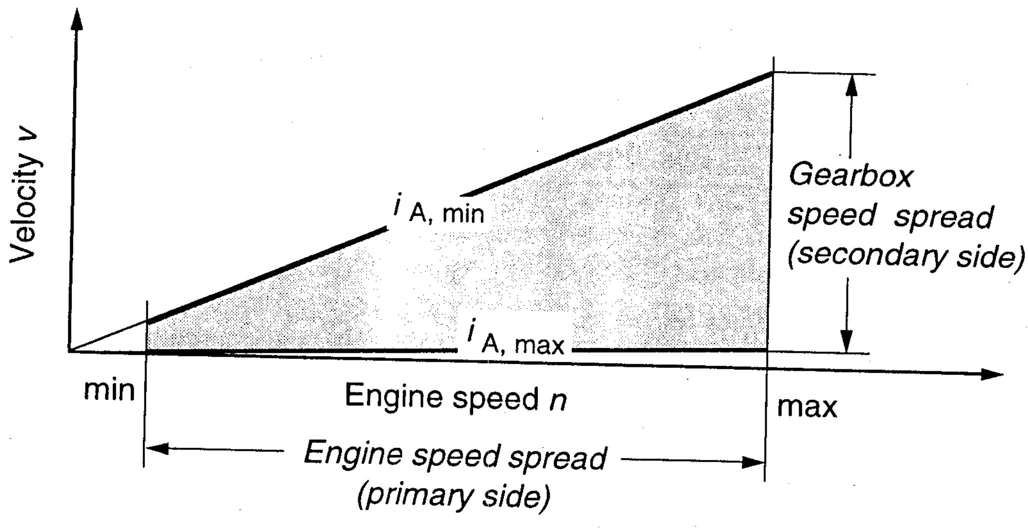


Figure 4.6. Velocity/engine-speed diagram, overall gear ratio

Increasing legal constraints and traffic density are reducing the importance of maximum speeds of passenger cars. By the same token, acceleration performance is gaining in importance.

A wide overall gear ratio is particularly important for heavy passenger cars with powerful engines and a low drag coefficients [4.3]. They need:

- a high stall torque ratio $i_{A,\,max}$ for moving off and accelerating,
- a low minimum ratio $i_{A,\,min}$ for low engine speeds at high road speeds to reduce fuel consumption.

4.3.1 Overall Gear Ratio

The overall gear ratio of the transmission, often referred to as the range of ratios, is the ratio between the largest and smallest ratio

$$i_{G,\,tot} = \frac{i_{G,\,max}}{i_{G,\,min}} = \frac{i_1}{i_z}\,, \quad \text{with the gears } n = 1 \text{ up to } z\,. \tag{4.8}$$

The overall gear ratio depends on:

- the specific power output of the vehicle ($P_{max} / (m_F + m_{payload})$ in kW/t),
- the overall gear ratio of the engine, (see Section 3.3.2), and
- the intended use.

Vehicles with a low specific power output, such as commercial vehicles, need a larger overall gear ratio. The same applies for vehicles with diesel engines, which have a small engine speed spread. Reference values for overall gear ratios of various vehicles are shown in Figure 4.7.

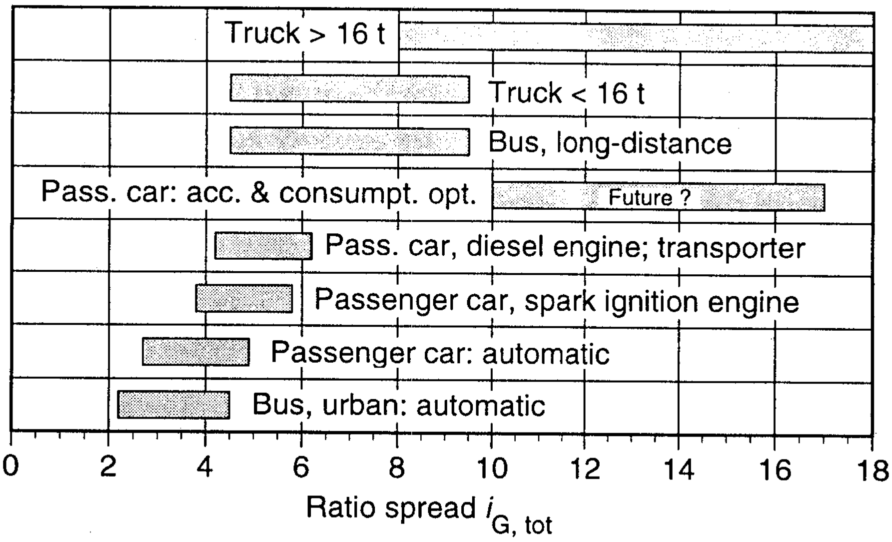


Figure 4.7. Reference values for overall gear ratios for various types of vehicle. In the case of automatic transmissions, the conversion of the torque converter ($\mu_{max} \approx 2$–3) has to be added

For passenger cars in particular it is necessary to consider that:

- However great the overall gear ratio is, the transmission can only move the operating point on the demand power hyperbola (see also Figure 3.17). The most fuel-efficient range cannot be exploited by a passenger car with a powerful engine travelling on the level at moderate speeds since there is "insufficient power required". The engine and all power-train components have to fit together: power-train matching, see Chapter 5.
- Overdrive gears ($i_G < 1.0$) result in reduced gear efficiency.

4.3.2 Selecting the Largest Power-Train Ratio

The greatest traction requirement must be known to determine the ratio of the gear with the largest torque multiplication. The adhesion limit – i.e. the maximum force that can be transmitted between the tyres and the road – is a physical limit and must be taken into account when establishing the traction $F_{Z,\,A}$ at the road wheels (see Equation 3.9)

$$F_{Z,\,A} \leq F_{Z,\,max} = \mu_H \; R \; .$$

Table 3.2 gives static friction coefficients μ_H for certain operational conditions. Air resistance may be ignored at the speeds anticipated in the lowest gear. At the drive wheels a balance must be struck between the maximum requirements of acceleration, gradient, road surface, payload and trailer load:

Maximum traction available $F_{Z,\,A}$ = Maximum traction required $F_{Z,\,B}$

$$T_{M,\,max} \; i_{A,\,max} \; \eta_{tot} \; \frac{1}{r_{dyn}} = m_F \; g \left(f_R \; \cos \alpha_{St} + \sin \alpha_{St} \right) + m_F \; \lambda \; a \; . \tag{4.9}$$

The largest ratio $i_{A,\,max}$, often called the stall torque ratio, depends mainly on the specific power rating (kW/t) of the vehicle. Two extreme conditions may be considered:

- The maximum gradient that can be climbed at an acceleration of $a = 0$ m/s^2.
  Climbing performance, Section 5.2.2,
- The maximum acceleration on the level.
  Acceleration performance, Section 5.2.3.

The stall torque ratio for *passenger cars* and *commercial vehicles* designed for maximum gradability is, from Equation 4.9:

$$i_{A,\,max} = \frac{r_{dyn} \;\; m_F \; g \left(f_R \; \cos \alpha_{St} + \sin \alpha_{St} \right)}{T_{M,\,max} \; \eta_{tot}} \; . \tag{4.10}$$

The dynamic wheel radius r_{dyn} of most common tyre sizes is shown in Table 3.3. Reference values for r_{dyn} are: ≈ 0.3 m for passenger cars and ≈ 0.5 m for commercial vehicles. Reference values for the rolling resistance coefficient f_R are shown in Table 3.1. A climbing performance of q'_{max} greater than 50% is normally required for an unladen passenger car. This ensures that a trailer can be towed and steep ramps mounted with ease.

Acceleration performance depends not only on the stall torque ratio, but also to a significant degree on how closely the gears approximate to the traction hyperbola. The acceleration performance required depends very much on the brand image of the vehicle.

The largest ratio in *commercial vehicles* is often dictated by the vehicle's intended use. For example, building site vehicles and road sweepers have gears for extremely slow movement (v_{crawl}). Using the kinematic relationship

$$v = \omega_W \; r_{dyn} \; , \tag{4.11}$$

the crawler gear in a commercial vehicle is given by

$$i_{A,\,max} = \frac{3.6\,\frac{\pi}{30}\,n_{M,\,min}\left[\frac{1}{min}\right] r_{dyn}\,[m]}{v_{crawl}\left[\frac{km}{h}\right]}, \tag{4.12}$$

where $n_{M,\,min}$ is in 1/min, r_{dyn} is in m and v_{crawl} in km/h. These very high-ratio gears are known as crawler gears.

4.3.3 Selecting the Smallest Power-Train Ratio

Assuming there is no slip in the power transmission from wheel to road and that the (nominal) maximum speed is reached at maximum engine speed, then the smallest power-train ratio is given by

$$i_{A,\,min} = \frac{3.6\,\frac{\pi}{30}\,n_{M,\,max}\left[\frac{1}{min}\right] r_{dyn}\,[m]}{v_{max}\left[\frac{km}{h}\right]}, \tag{4.13}$$

where $n_{M,\,max}$ is in 1/min, r_{dyn} in m and v_{max} in km/h.

Commercial Vehicles

The limiting factors of legal speed restrictions and diesel engine cut-off speed mean that the maximum speed will often be a design parameter when developing commercial vehicle power trains. The design ranges for commercial vehicles in Europe arising from the maximum permissible speed v_{max} are shown in Figure 4.8.

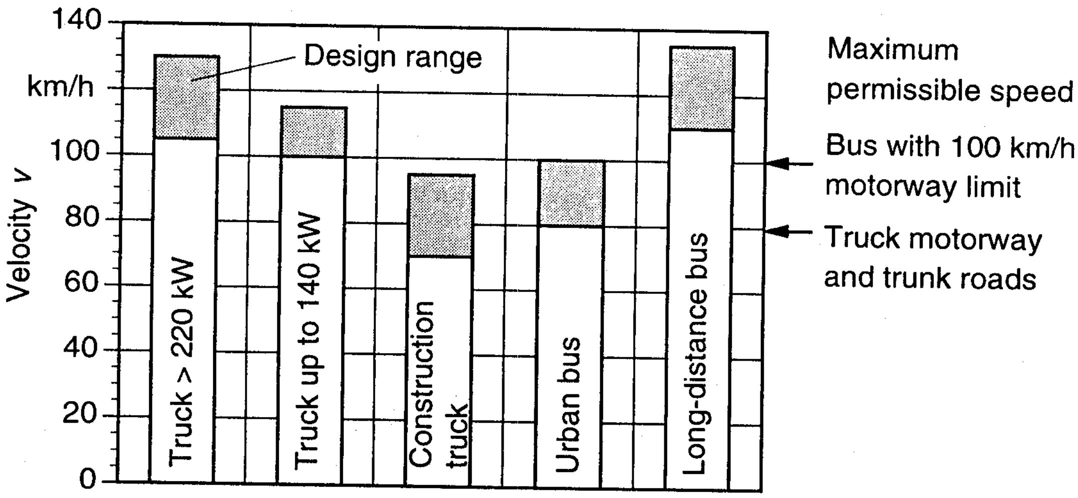


Figure 4.8. Design speeds for determining $i_{A,\,min}$ for commercial vehicle power trains. The maximum speed data relates to Germany

Passenger Cars

There are various factors to be considered in selecting the smallest ratio. One factor is the high proportion of duty time spent in the highest gear, which can be more than 80% in the case of passenger cars. Depending on the type of design selected, a distinction is made between:

1/ v_{max} – *optimum design*: $i_{A,\,min} = i_A(v_{max,\,th})$,
2/ *overrevving design*,
3/ *underrevving design*.

1/ v_{max} – Optimum Design

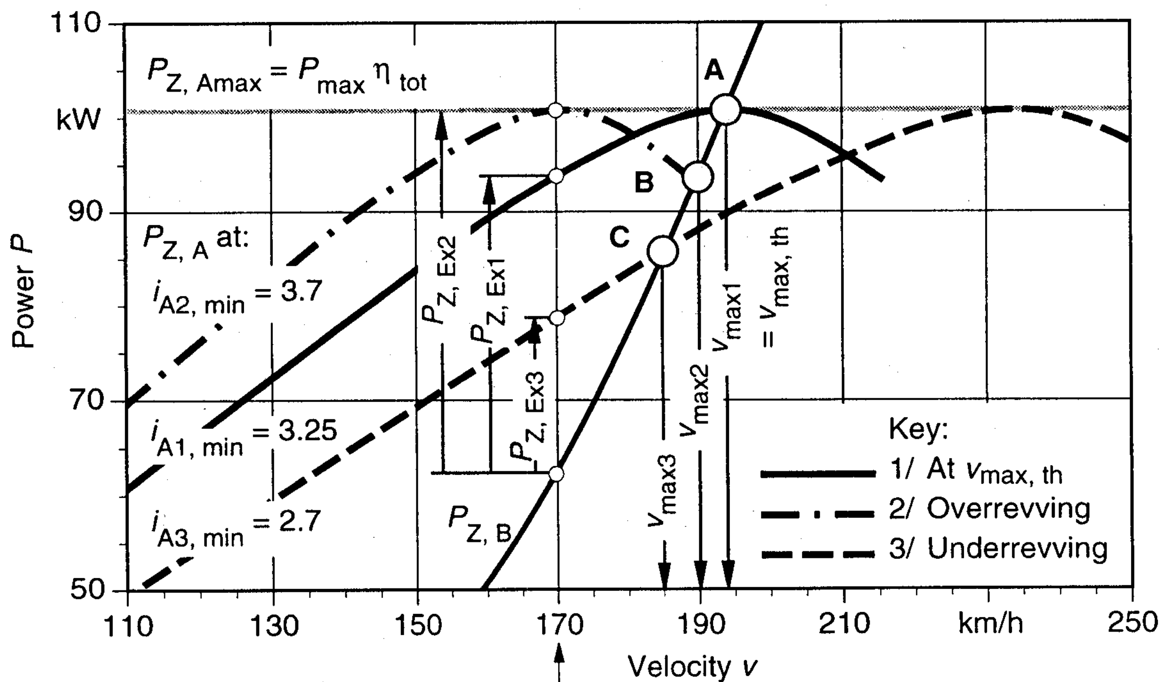
In order to convert the maximum engine power installed in the vehicle into maximum performance, the required power curve $P_{Z,\,B}$ must pass through the point of maximum engine power available $P_{Z,\,Amax}$ (= P_n) [4.4]. This is called the design point *A*, Figure 4.9. It represents the maximum speed $v_{max,\,th}$ theoretically available ($q' = 0\%$; no wind).

The acceleration reserve and fuel consumption in top gear are also important factors in the case of passenger car transmissions. The excess power available $P_{Z,\,Ex}$ is a measure of acceleration reserve, and the engine speed n_M serves as a measure for fuel consumption (Figure 4.9).

2/ Overrevving Design

The power available and the power required intersect in the declining section of the power supply curve $P_{Z,\,A}$ as shown in Figure 4.9, Point *B*. The speed v_{max2} which can be achieved at this point with an overrevving design is less than $v_{max,\,th}$.

The power-train ratio $i_{A2,\,min}$ is greater than $i_{A1,\,min} = i_A(v_{max,\,th})$. This is achieved by increasing the ratio of the highest gear i_z or the final ratio i_E. Since the engine speed is then higher for a given road speed, the operating point moves into the range of higher fuel consumption on the engine map. The high level of excess power $P_{Z,\,Ex2}$ makes this arrangement preferable for sports designs.



n_{M1}	3161	3736	4310	4885	5460	6034	1/min	
n_{M2}	3599	4253	4907	5562	6216			
n_{M3}	2626	3104	3581	4058	4536	5014	5491	5968

Engine speed n_M

Figure 4.9. Selecting the ratio in top gear for passenger cars. The excess and associated engine speeds are designed for a speed of 170 km/h. Design: 1/ at $v_{max,\,th}$; 2/ overrevving; 3/ underrevving

3/ Underrevving Design

The power available and power required intersect on the rising section of the power supply curve, point *C*. In this case the power-train ratio $i_{A3,\,min}$ is less than $i_A(v_{max,\,th})$. The reduction in engine speed is the important feature of this design. The operating point moves into an area of improved fuel consumption. There are various approaches to reducing the power-train ratio in the underrevving design, as follows:

- increase the overall gear ratio with the same number of speeds (Figure 4.10b),
- reduce the final ratio ("long axle design" – Figure 4.10c),
- increase the overall gear ratio by increasing the number of gears – overdrive – (Figure 4.10d).

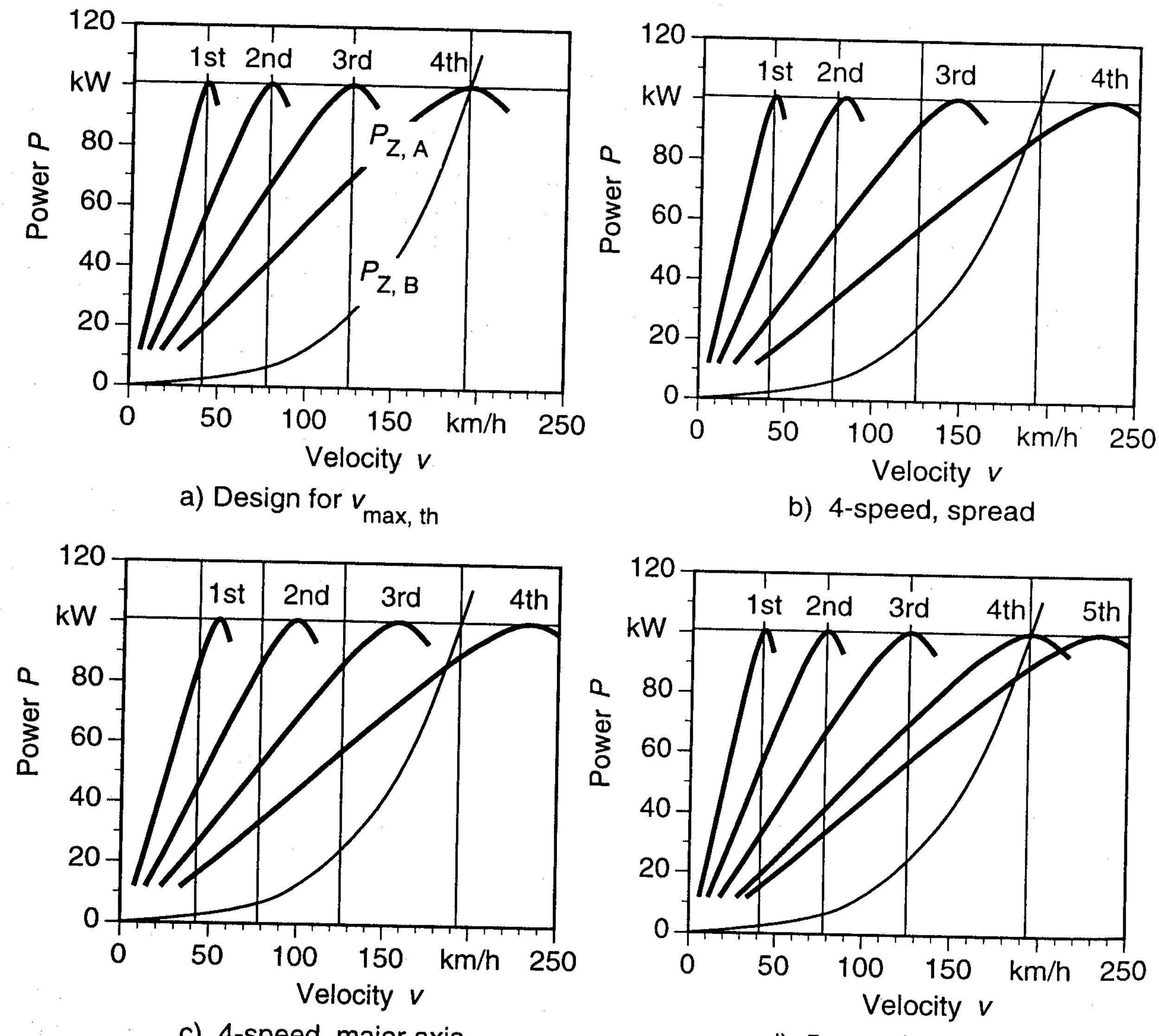


Figure 4.10. Selecting underrevving power-train ratios to improve fuel economy. Starting point: Figure 4.10a

Figure 4.10 illustrates these approaches using the example of a passenger car power train with a 4-speed gearbox. The basic design is a power train optimised for $v_{max,\,th}$ (Figure 4.10a). The effect of increasing the overall gear ratio with the same number of gears is to create relatively large gaps in the power output, thus reducing the vehicle's acceleration performance. Increasing the final ratio ("long axle design") with the same overall gear ratio leads to a smaller stall torque ratio, and thus to reduced climbing performance and increased clutch stress when moving off. For this reason manufacturers now add a fifth gear (overdrive) to the conventional 4-speed manual shift gearbox. Most fifth gear designs reduce engine speed by 10–20% ($i_5 = 0.8–0.9$).

The fifth and sixth speed on manual passenger car gearboxes can be designed as overdrives to reduce engine speed. Alternatively they can provide a close-ratio sports gearbox in which the increased number of gears serves to approximate more closely to the traction hyperbola and thus enhance performance.

4.3.4 Final Ratio

The final ratio i_E selected adapts the handling characteristics and fuel consumption to the intended function, and is particularly important for commercial vehicles. For example trucks and buses operating mostly in flat terrain are fitted with longer axle designs than those operating in hilly regions. The longer axle design ($i_{E,\ long} < i_{E,\ normal}$) reduces the engine speed at normal driving speed, but also the excess traction in all gears (Figure 4.10c).

The final ratio achievable in a single step is $2 \leq i_E \leq 7$. Larger ratios are achieved with an additional ratio stage in the final drive.

The various ways of designing the final ratio are listed systematically in Section 6.9. Section 12.5 describes some examples of final drives currently in use.

4.4 Selecting the Intermediate Gears

The relationship between the ratio of two neighbouring gears, the gear step φ, is given by

$$\varphi = \frac{i_{n-1}}{i_n} \leq \frac{n_{max}}{n\left(T_{max}\right)} \,. \tag{4.14}$$

The transmission stepping should be large enough to enable the next lower gear ($n - 1$) to be engaged when the maximum engine torque is reached in gear n, without exceeding the maximum permissible engine speed n_{max} (Figure 4.11). The following aspects should be considered when selecting the gear ratios:

- The greater the number of gears, the better the engine exploits its efficiency by adhering to the traction hyperbola. But as the number of gears increases, so does the frequency of gear shifting and the weight and size of the gearbox.
- The proportion of distance travelled in the lower gears is low, especially in the case of passenger cars.
- The proportion of distance travelled in each gear depend on the specific power output (kW/t), the route profile, the traffic conditions and driver behaviour.
- The smaller the gear step φ, the easier and more pleasant the gearshift action.
- The thermal load on the synchroniser rings is proportional to the square of the gear step.

In view of these in part contradictory aspects, compromises have to be made in designing the gearbox.

Two formal methods for calculating gear steps have proved effective in practice:

- *geometrical gear steps,*
- *progressive gear steps.*

4.4.1 Saw Profile Diagram

The velocity/engine-speed diagram gives a good overview of appropriate configurations of the transmission ratios. It is often referred to as a gear plan or saw profile diagram, and has the road speed plotted against the engine speed for each gear n, from $n = 1$ to z.

Figure 4.11 shows the velocity/engine-speed diagram for a bus with an 8-speed 2-range transmission. The gearbox is geometrically stepped (see Section 4.4.2).

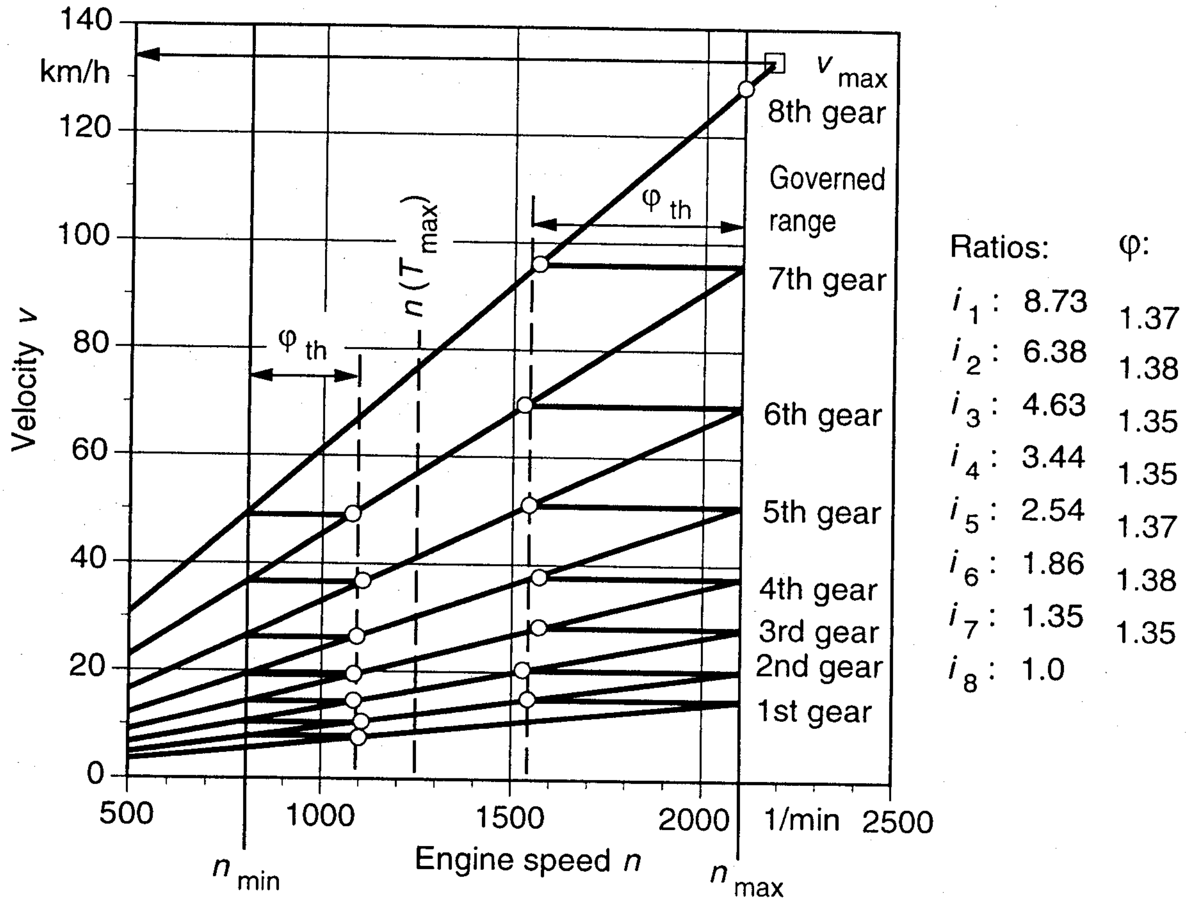


Figure 4.11. Velocity/engine-speed diagram of a bus with 8-speed 2-range transmission. Maximum road speed in the diesel engine governed range.

The maximum speed is reached in 8th gear in the "governed range" of the diesel engine (see also Figure 5.6). The saw profile diagram shows the earliest upshift possible without stalling the engine, and the earliest downshift possible without exceeding the maximum engine speed.

4.4.2 Geometrical Gear Steps

In the geometric design the gear step φ between the individual gears always has the same theoretical value

$$\varphi_{\text{th}} = \sqrt[z-1]{i_{\text{G, tot}}} \ . \tag{4.15}$$

The ratios of the individual gears $n = 1$ to z is then given by

$$i_{\text{n}} = i_{\text{z}}\ \varphi_{\text{th}}^{(\text{z}-\text{n})} \ . \tag{4.16}$$

In practice the gear step will vary slightly from φ_{th} (Figure 4.11). The approximation to the effective traction hyperbola $F_{Z,\,Ae}$ is equally good in all gears (Figure 4.12a). The difference in maximum speed between the gears consequently increases with each shift to a higher gear.

Geometrical gear steps are most common in commercial vehicle gearboxes; the lower specific power output means all the gear steps are of equal significance. Range transmissions (Figure 4.11) have to be geometrically stepped to make all ratio steps the same size, preventing individual gears from overlapping (see also Section 6.7.2 "Multi-Range Transmissions").

4.4.3 Progressive Gear Steps

Progressive gear steps are used for passenger car transmissions. The higher the gear, the smaller the gear step. Figure 4.12b shows the progressive transmission stepping in the traction diagram and the velocity/engine-speed diagram (saw profile diagram).

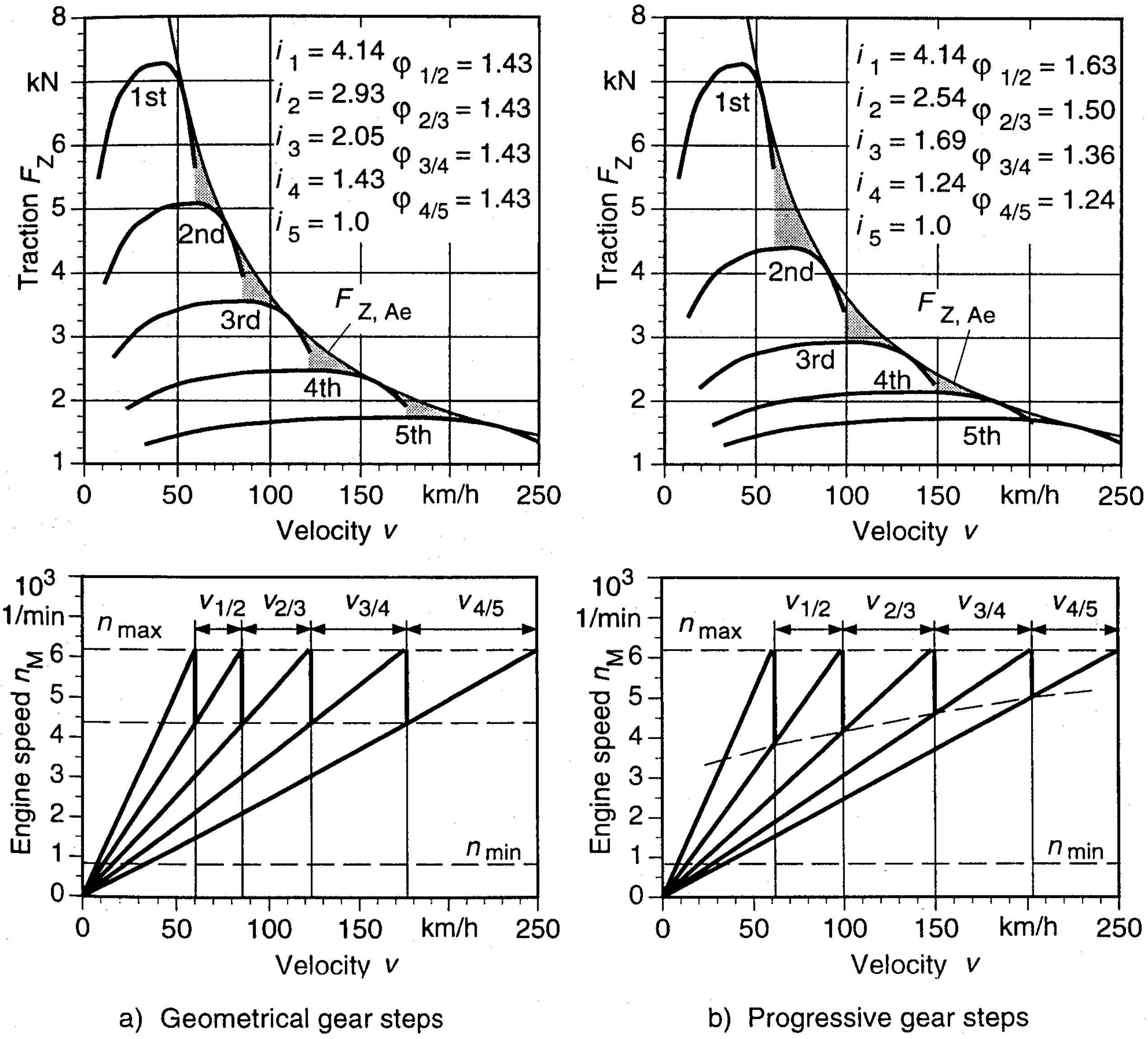


Figure 4.12. Gear steps. Effects in the traction diagram and the velocity/engine-speed diagram (saw profile diagram). Ratios as shown in Table 4.1.
a) Geometrical gear steps; **b)** Progressive gear steps

This shows clearly how the difference between the gear maximum speeds remains roughly constant with progressive gear steps. In the traction diagram the gaps between the effective traction hyperbola and the traction available are reduced in the top gears.

In the speed range relevant to passenger cars, this is reflected in improved gearshift action (smaller φ), and in improved acceleration performance. The high level of excess power available in the lower speed range of passenger cars means that larger gaps in the traction availability are acceptable.

Given the overall gear ratio $i_{G,\,tot}$ and the selected progression factor φ_2, the base ratio change φ_1 can be calculated thus:

$$\varphi_1 = \sqrt[z-1]{\frac{1}{\varphi_2^{0.5(z-1)(z-2)}}\, i_{G,\,tot}} \;. \tag{4.17}$$

The ratios i_n in the gears $n = 1$ to z are found to be

$$i_n = i_z\, \varphi_1^{(z-n)}\, \varphi_2^{0.5(z-n)(z-n-1)} \;. \tag{4.18}$$

Typical values are: $\varphi_1 = 1.1$ to 1.7,
$\varphi_2 = 1.0$ to 1.2.

The above calculation method provides initial values to be used when selecting ratios. The gear ratios need to be adapted to the vehicle by a process of fine adjustment involving bench tests, road tests and computer simulation of field conditions. Other testing and acceptance conditions, such as fuel consumption and exhaust emissions, may also be significant in particular cases. Table 4.1 gives an example of the design of a passenger car transmission. The progression factor is $\varphi_2 = 1.1$ and the base ratio change is $\varphi_1 = 1.24$.

Table 4.1. **a)** Geometrical gear steps. **b)** Transmission ratios of a 5-speed passenger car gearbox derived by calculation and ultimately built after fine tuning

a) Gear: geom.	1	2	3	4	5
$i_{calculated}$	4.14	2.93	2.05	1.43	1.0
b) Gear: progr.	1	2	3	4	5
$i_{calculated}$	4.14	2.54	1.69	1.24	1.0
i_{built}	4.2	2.49	1.66	1.24	1.0

4.5 Continuously Variable Transmissions

Continuously variable transmissions are torque and speed converters whose ratio can be continuously varied without interrupting the traction flow. In combination with an intelligent engine/transmission control, continuously variable transmissions make it possible to exploit the engine performance characteristic curve more fully.

Engine torque and speed may be freely selected with a continuously variable transmission, but they must be located on the current demand power hyperbola and in the operating map defined by the overall gear ratio (see also Figure 5.12 in Section 5.3.4 "Continuously Variable Transmissions"). Here the engine speed and transmission ratio are directly interrelated.

The overall gear ratio of pulley transmissions is normally $i_{G,\,tot} = 5$ to 6. Continuously variable transmissions are normally described in terms of their governed range rather than the overall gear ratio. It is possible to achieve continuously variable pulley transmissions with a larger governed range by suitable engineering design (for example by using several power branches, see Section 6.6.4). The limited torque capacity of the chain currently restricts its use to vehicles with a transmission input torque of approximately 300 Nm.

The speed of adjustment of the transmission, and thus the engine stroke, is a decisive factor in smooth running [4.5]. The adjustment speed is defined as

$$\dot{n} = \frac{\mathrm{d}n_M}{\mathrm{d}t} = n_{\mathrm{output}} \frac{\mathrm{d}i}{\mathrm{d}t} \ . \tag{4.19}$$

If the adjustment speed is too high, smoothness of operation suffers. The balance of energy required to make the adjustment is taken from the kinetic energy of the vehicle. This can reverse the sign of the acceleration resulting in "gearshift jolt", which the driver finds uncomfortable.

If the adjustment speed is too low, the smoothness of operation is enhanced but the responsiveness of the vehicle suffers.

5 Matching Engine and Transmission

Apart from design, the main factors affecting the market success of a vehicle are performance, fuel consumption, emissions and comfort

Chapter 3 dealt with the supply of and demand for power. Chapter 4 then elaborated the principles for selecting overall gear ratios. The purpose of this chapter is now to consider matching the transmission to the engine and the vehicle. This involves problems of vehicle longitudinal dynamics. The power train and its components are optimised by means of computer driving simulation and road and bench tests. The power-train components – engine, moving-off element, selector gearbox, final drive, etc. – must be "harmoniously" combined. This matching process is called "power-train matching". The main optimisation criteria in this process are

- ❍ performance,
- ❍ fuel consumption,
- ❍ emissions, and
- ❍ comfort.

This adaptation process has to be tackled from both sides, matching the engine to the transmission and vice versa. In practice the characteristics of the engine dominate, and the characteristics of the transmission have to be "adapted" to match.

The transmission mediates between the engine and the road surface; it adapts the traction available to the power required, ensuring the desired performance. For this purpose the speed range of the engine is mapped to a wheel speed range or a road speed range. Similarly the torque range of the engine is mapped to a torque or traction range at the wheels. The speed and torque range of the engine should be referred to as the "engine spread" as discussed in Section 3.3.2. Engine spread and the overall gear ratio together form a field of possible traction at the wheels (Figure 5.1). The transmission enables the most fuel-efficient operating regions of the engine to be exploited (Figure 5.2).

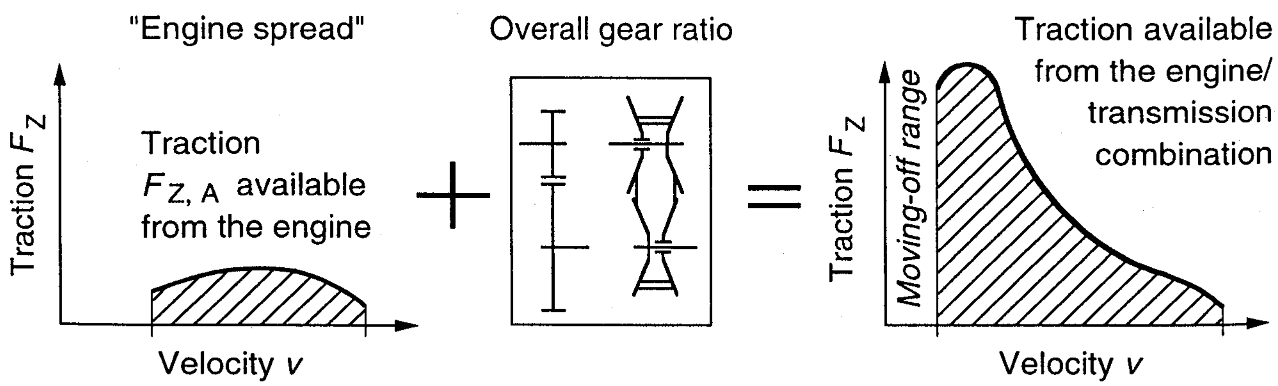


Figure 5.1. Combined action of "engine spread" and overall gear ratio

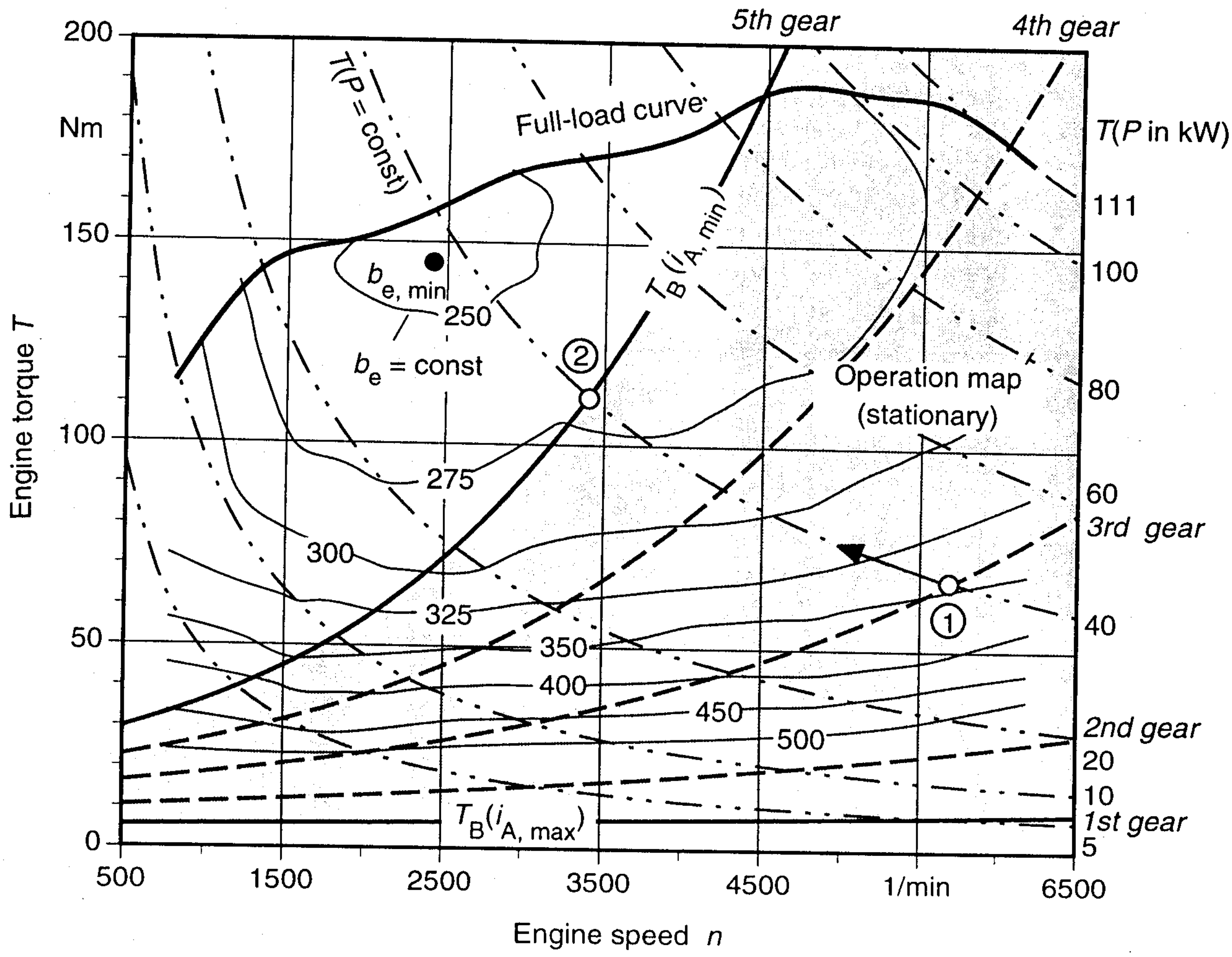


Figure 5.2. Engine performance map with the characteristic onion curves of constant fuel consumption of a 2-litre spark ignition engine. The tractional resistance lines for minimum ($i_{A,\,min}$) and maximum ($i_{A,\,max}$) power-train ratio define an operating map for points representing unaccelerated driving on the level. Vehicle, engine and transmission data as in Figure 5.3

Fuel consumption is influenced greatly by the gear ratio and the final ratio specified, and the gearshift points selected. During unaccelerated driving on a level surface there can be "discrete" operating curves (with geared transmissions), or a whole operating map (with continuously variable transmissions), between the driving resistance lines, T_B, for the minimum and maximum power-train ratios. With geared transmissions the operating points lie at the intersections of the ratio-dependent driving resistance lines and the lines showing engine power available $T(P$ = const). But the operating points can only lie within the specified field. The operating map shifts on uphill and downhill runs and when accelerating, because of the changing driving resistance.

In geared transmissions, the most favourable consumption area of the driving resistance curve is usually only covered by top gear. For example, Figure 5.2 shows how the power 40 kW, $T(P$ = 40 kW), can be produced during unaccelerated movement on a level surface (a = 0 m/s^2 and q' = 0%) fuel-inefficiently in 3rd gear at point *1* is $b_e \approx$ 350 g/kWh, or fuel-efficiently in 5th gear at point *2* is $b_e \approx$ 270 g/kWh. The ratio of an "overdrive" gear has been selected to be fuel-efficient when the driving resistance curve comes as close as possible to Point $b_{e,\,min}$ of lowest specific fuel consumption. See also Section 3.3.3 "Consumption Map".

5.1 Traction Diagram

The acceleration and climbing performance in the various gears of the transmission must be checked. In the traction diagram (Figure 5.3), the traction available in each gear and the traction required at various gradients are plotted as a function of the vehicle speed using Equations 3.19 to 3.22. The traction available is reduced by the power-train efficiency, η_{tot}, which also includes the effect of losses due to accessories such as servo pumps (see Section 3.1.7).

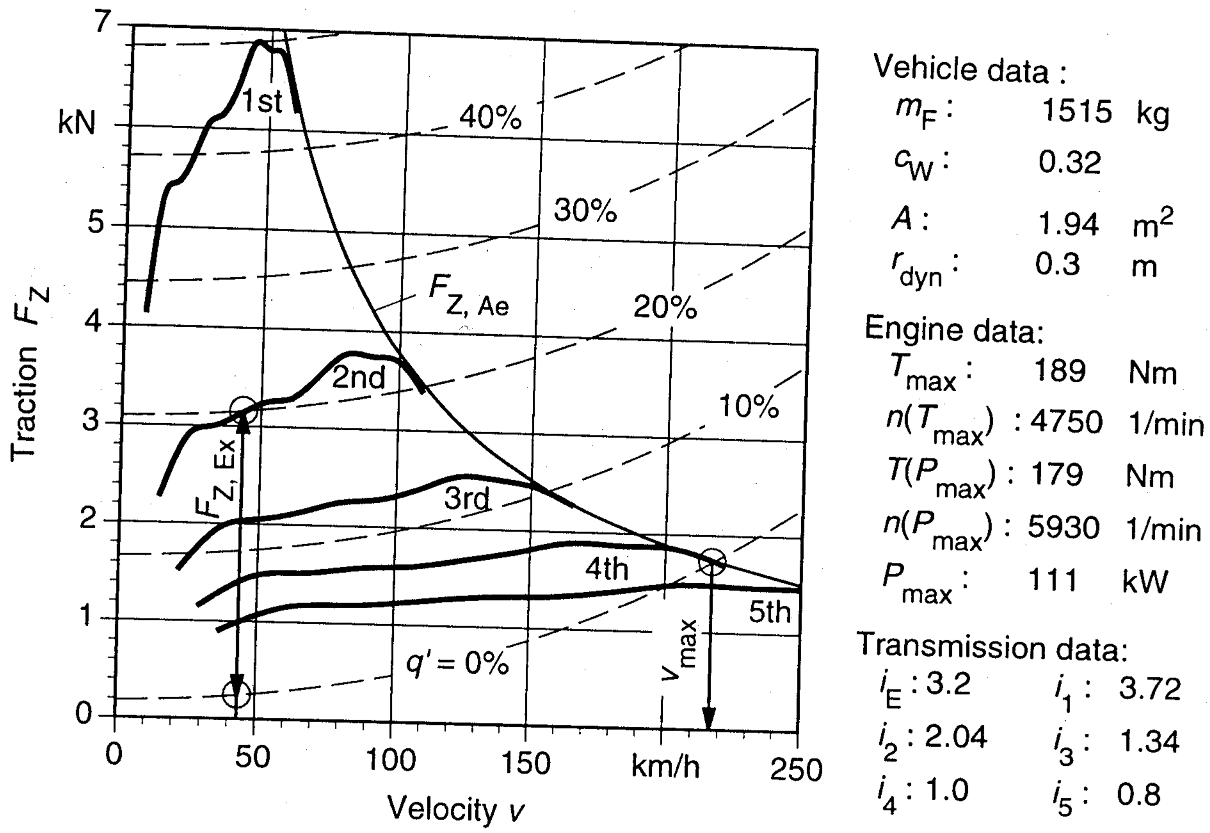


Figure 5.3. Traction diagram with the demand curves for various gradients for a mid-sized passenger car with a 2-litre spark ignition engine as shown in Figure 5.2

The power may be calculated by multiplying the traction by the corresponding speed, and can be shown on the performance diagram (Figure 5.4).

Figures 5.3 and 5.4 are the traction diagram and performance diagram respectively of a mid-size passenger car with a 2-litre spark ignition engine. Figure 5.6 shows the traction diagram of a 16 t truck. The influence of an additional splitter unit in reducing the step between overall gear ratios can be seen there. The splitter unit enables a better approximation to the traction hyperbola $F_{Z, Ae}$.

The maximum speed, the maximum climbing performance and excess traction in the various gears can be found from the traction diagram. The excess traction $F_{Z, Ex}$ is given by the formula

$$F_{Z, Ex} = F_{Z, A} - F_{Z, B} = F_{Z, A} - F_R - F_{St} - F_L - F_a ,$$

$$= \frac{T(n_M)\, i_A}{r_{dyn}} \eta_{tot} - m_F\, g\, (f_R \cos\alpha_{St} + \sin\alpha_{St}) - \frac{1}{2} \rho_L\, c_W\, A\, v^2 - m_F\, \lambda\, a . \tag{5.1}$$

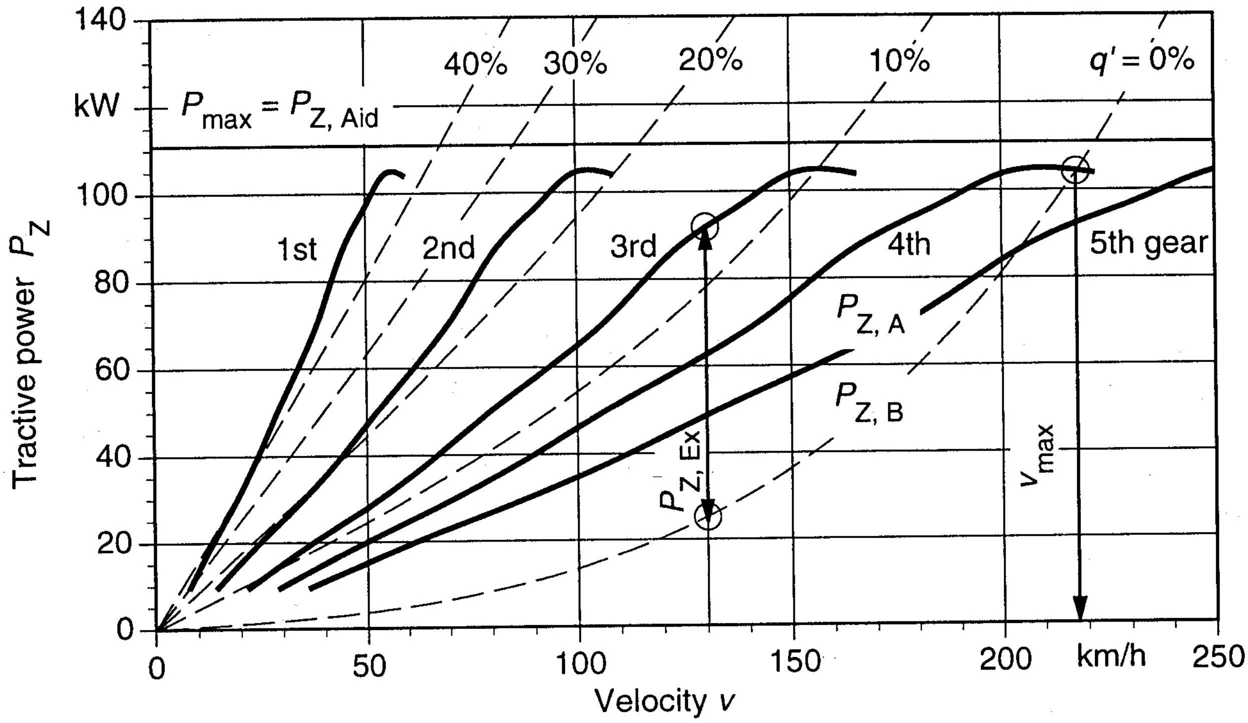


Figure 5.4. Performance diagram (derived from the traction diagram, Figure 5.3)

The traction diagram shows unaccelerated movement, i.e. when $a = 0$ m/s^2. To interpret the climbing and acceleration performance of a vehicle, the excess power available at the operating point, $F_{Z,\,Ex}$, can be written in two ways. To give climbing performance during unaccelerated movement,

$$F_{Z,\,Ex} = F_{Z,\,A} - F_R - F_L = m_F\, g \sin \alpha_{St} \tag{5.2}$$

and to give acceleration performance during movement on the level

$$F_{Z,\,Ex} = F_{Z,\,A} - F_R - F_L = m_F\, \lambda\, a\ . \tag{5.3}$$

5.1.1 Deriving a Traction Diagram (Example)

Construction of the traction diagram in Figure 5.3 is explored below as an example. The procedure can be split into the following steps:

A Determining the Traction Available

1 *Specifying the initial dynamic operating parameters*
The following calculations are based on the engine and transmission data given in Figure 5.3.

2 *Selecting some characteristic points on the full load curve*
The full load curve of the sample engine is shown in Figure 5.2. The values at maximum torque and maximum power provide the first entries in Table 5.1.

Table 5.1. Design table of the traction diagram (continued)

n_M (1/min)	800	2000	3000	4000	4750	5930	6200
T_M (Nm)	115	150	170	175	189	179	170

3 Calculating the associated gear-dependent speeds and tractions

This is to be carried out as an example for 1st gear with $i_1 = 3.72$. From Equation 4.11

$$v\left[\frac{\text{km}}{\text{h}}\right] = \frac{3.6\,\frac{\pi}{30}\,n_M\left[\frac{1}{\text{min}}\right] r_{dyn}}{i_1\, i_E}$$

$v_{1st\ G}$ (km/h)	7.6	19.0	28.5	38.0	45.1	56.3	58.9

The power-train efficiency in 1st gear is assumed to be constant. $\eta_{tot} = 0.92$. Using Equation 3.22

$$F_{Z,A}\,[\text{kN}] = \frac{T\,(n_M)\,i_E\,i_1}{1000\,r_{dyn}}\,\eta_{tot}$$

$F_{Z,A\ 1st\ G}$ (kN)	4.2	5.5	6.2	6.4	6.9	6.5	6.2

4 Entering the traction available/speed values on a diagram

See Figure 5.3 $F_{Z,A}$ curve for 1st gear.

B Determining the Driving Resistance Lines

1 Determining the initial values

The initial values for calculating the road resistance lines are the vehicle data given in Figure 5.3. The density of the air $\rho_L = 1.199$ kg/m³.

2 Calculating the traction required at several speeds and gradients

Using Equation 3.20 for unaccelerated movement

$$F_{Z,B}\,[\text{kN}] = \frac{1}{1000}\left[m_F\,g\,(f_R\,\cos\alpha_{St} + \sin\alpha_{St}) + \frac{1}{2}\,\rho_L\,c_W\,A\,\frac{v^2\left[\frac{\text{km}}{\text{h}}\right]}{3.6^2}\right].$$

For gradients greater than 10%, the approximations $\cos\alpha_{St} \approx 1$ and $\sin\alpha_{St} \approx \tan\alpha_{St}$ are no longer acceptable. Entering the speed-dependent rolling resistance coefficient f_R gives:

v (km/h)	0	50	100	150	200	250
f_R	0.0124	0.0124	0.0131	0.0145	0.020	0.033
$F_{Z,B\ 0\%}$ (kN)	0.18	0.26	0.48	0.86	1.45	2.29

3 Entering the traction required/speed values on the diagram

See Figure 5.3, $F_{Z,B}$ curve for movement on the level, $q' = 0\%$.

C Reading of Relevant Data

1 Maximum speed

The maximum speed of the vehicle on the level is achieved in 4th gear, and is approximately 218 km/h. It is found at the intersection of the traction available line and the road resistance line for $q' = 0\%$.

2 Other performance data

See Tables 5.2 and 5.3.

5.1.2 Engine Braking Force

"Good braking means faster driving". This applies in particular to trucks with their large vehicle weight. Fast downhill speeds are required for trucks to achieve high average speeds and hence economic transport. Attainable downhill speeds are those which can be travelled without acceleration and without activating the service brake (friction brake). Depending on the type of braking, a distinction is made [5.1] between

- *steady-state braking*
  preventing unwanted acceleration on downhill runs,
- *deceleration braking*
  reducing speed and stopping if necessary, and
- *braking at rest*
  preventing undesired movement of the vehicle at rest.

In overrun conditions the internal combustion engine delivers braking torque (see Figure 3.12). The braking torque is principally the result of compression work in the cylinders. In the case of commercial vehicles, additional continuous service brakes, for example an exhaust throttle or retarder, can further increase the engine braking effect. See also Section 11.5 "Vehicle Continuous Service Brakes".

The engine braking power available $F_{B,A}$ is incorporated in the traction diagram in a similar way to the traction available $F_{Z,A}$. $F_{B,A}$ is often also referred to as towing drag. Power flow in overrun conditions is from the wheels to the engine. Whereas the traction diagram is calculated from the full load characteristic curve of the engine when power flow is from the engine to the road (Figure 5.5),

$$F_{Z,A} = \frac{T(n_M)\left(\dfrac{n_M}{n_R}\right)}{r_{dyn}} \eta_{tot} = \frac{T(n_M)\, i_A}{r_{dyn}} \eta_{tot} \tag{5.4}$$

in overrun conditions the calculation is from the road to the thrust characteristic curve of the engine

$$T(n_M) = F_{B,A}\, r_{dyn} \left(\frac{n_R}{n_M}\right) \eta_{tot} \,, \tag{5.5}$$

$$F_{B,A} = \frac{T(n_M)}{r_{dyn}\, \eta_{tot} \left(\dfrac{n_R}{n_M}\right)} = \frac{T(n_M)\, i_A}{r_{dyn}\;\; \eta_{tot}} \,. \tag{5.6}$$

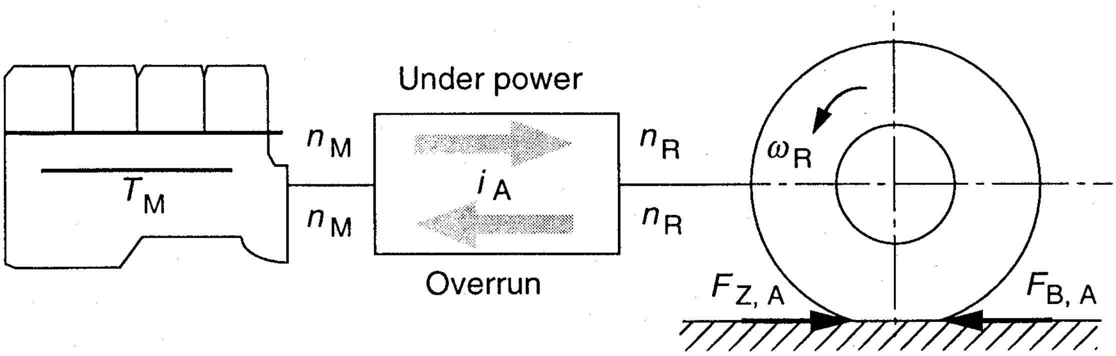


Figure 5.5. Power flow under power and during overrun

If the variation in the power-train efficiency η_{tot} = function (ratio, speed, torque) is taken into account in calculating engine braking force, then it must be remembered that the ratio is defined in the direction of power flow. That means that in overrun conditions "the ratio is reversed".

The equation of motion for braking is derived from the equations for motion under power. See Section 3.1 "Power Requirement". In deceleration braking, i.e. $a < 0$ m/s², inertial forces operate. They correspond to the acceleration resistance F_a. During steady-state braking $a = 0$ m/s²; braking is supported by rolling resistance and air resistance, which are given a negative sign.

On downhill runs the slope descending force F_H corresponds to the gradient resistance F_{St} with negative gradient, $q < 0\%$. Downhill, the resultant braking force requirement $F_{B,B}$ at the wheels is given by

$$F_{B,B} = F_H - F_a - F_R - F_L \,. \tag{5.7}$$

The braking force deficit, $F_{B,D}$, of the engine must be covered by the service brake or, in the case of commercial vehicles, by an additional continuous service brake system.

$$F_{B,D} = F_{B,B} - F_{B,A}(n_M, i_A) \,. \tag{5.8}$$

In the traction diagram for a 16 tonne truck with a 6-speed transmission (Figure 5.6), the curve of maximum engine braking force is shown for each gear. The 173 kW engine has an engine braking power of 57 kW at 2100 1/min. In Figure 5.6 the engine braking curves where an engine brake (exhaust throttle) is in use are represented by continuous grey lines. With an exhaust throttle the engine achieves a braking power of approximately 100 kW at 2100 1/min.

Engine braking force in 5th gear without an exhaust throttle is not adequate to prevent acceleration when travelling down a 5% slope (Point *1*); however this is made possible by using an engine brake in 5th gear (Point *2*). Without an exhaust throttle the vehicle would have to travel down the slope in 3rd gear at a lower speed (Point *3*).

5.1.3 Geared Transmission with Dry Clutch

Figures 5.3 and 5.6 show the interaction of a combustion engine with a geared transmission in the case of a passenger car (Figure 5.3) and a truck (Figure 5.6). In both cases the moving-off element is a conventional dry clutch. In trucks and buses with powerful engines the maximum speed is often reached in the governed range of the diesel engine, i.e. beyond the actual maximum engine speed. The traction diagram (Figure 5.6) shows how the gear sequence may be refined with an optional front-mounted splitter unit. It is plain to see how this achieves an improved adaptation to the effective traction hyperbola.

5.1.4 Geared Transmission with Trilok Converter

The converter test diagram (Figure 5.7) is necessary to determine the traction curve of a geared transmission with a Trilok converter. See also Chapter 10 "Hydrodynamic Clutches and Torque Converters" and Section 4.2 "Speed Converter for Moving Off". In the converter test diagram the pump test torque T_{PV} (T_{P2000}), determined at a pump test speed n_{PV} = 2000 1/min, and the associated torque conversion μ, are shown plotted against the speed conversion ν.

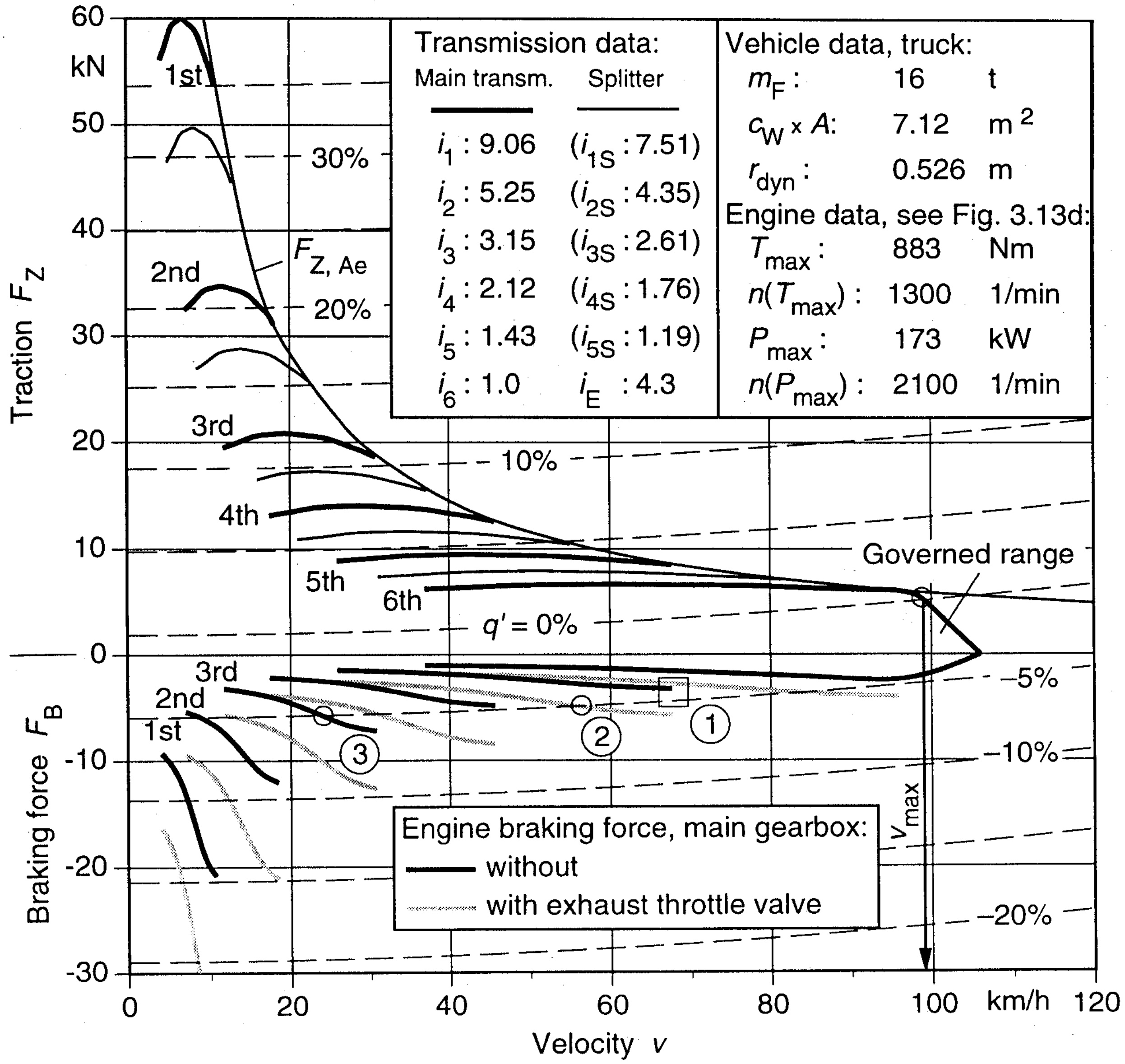


Figure 5.6. Traction diagram of a 16 tonne truck with 6-speed gearbox. Engine braking curves with and without exhaust throttle valve

The pump torque parabolas at constant speed conversion, the so-called converter parabolas, are added to the engine characteristic graph by using the information in the converter test diagram. The converter parabolas span a field of possible engine operating points. With the T_{PV} values from Figure 5.7a and where n_{PV} = 2000 1/min, then

$$k(v) = \frac{T_{PV}}{n_{PV}^2} , \tag{5.9}$$

from this the converter parabolas may be found:

$$T_P = k(v)\, n_P^2 , \tag{5.10}$$

where $T_P = T_M$ and $n_P = n_M$. At the intersections of the converter parabolas with the full load curve, the speed n_P and the torque T_P on the engine (or pump) side are now converted to the speed n_T and the torque T_T on the turbine side of the converter.

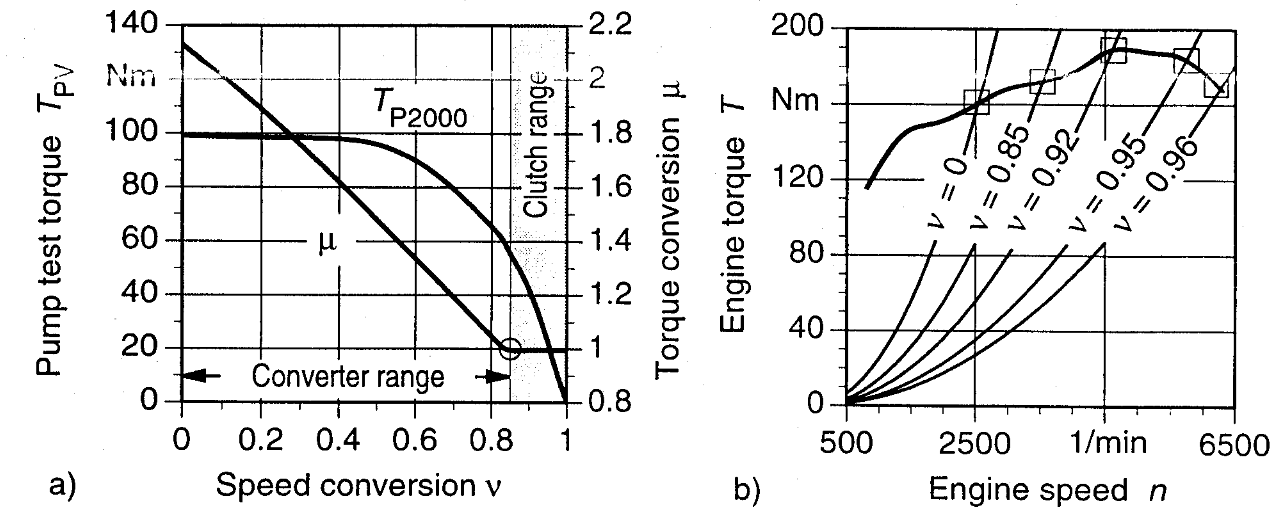


Figure 5.7. **a)** Converter test diagram; **b)** Engine performance map with torque converter parabolas. $n_{PV} = 2000$ 1/min; $\nu_C = 0.85$

where

$$n_T = \nu\, n_P \ . \tag{5.11}$$

For specified ν-values the associated values of μ can now be read off the converter test diagram. Using these values, the turbine torque T_T may be calculated

$$T_T = \mu\, T_P \ . \tag{5.12}$$

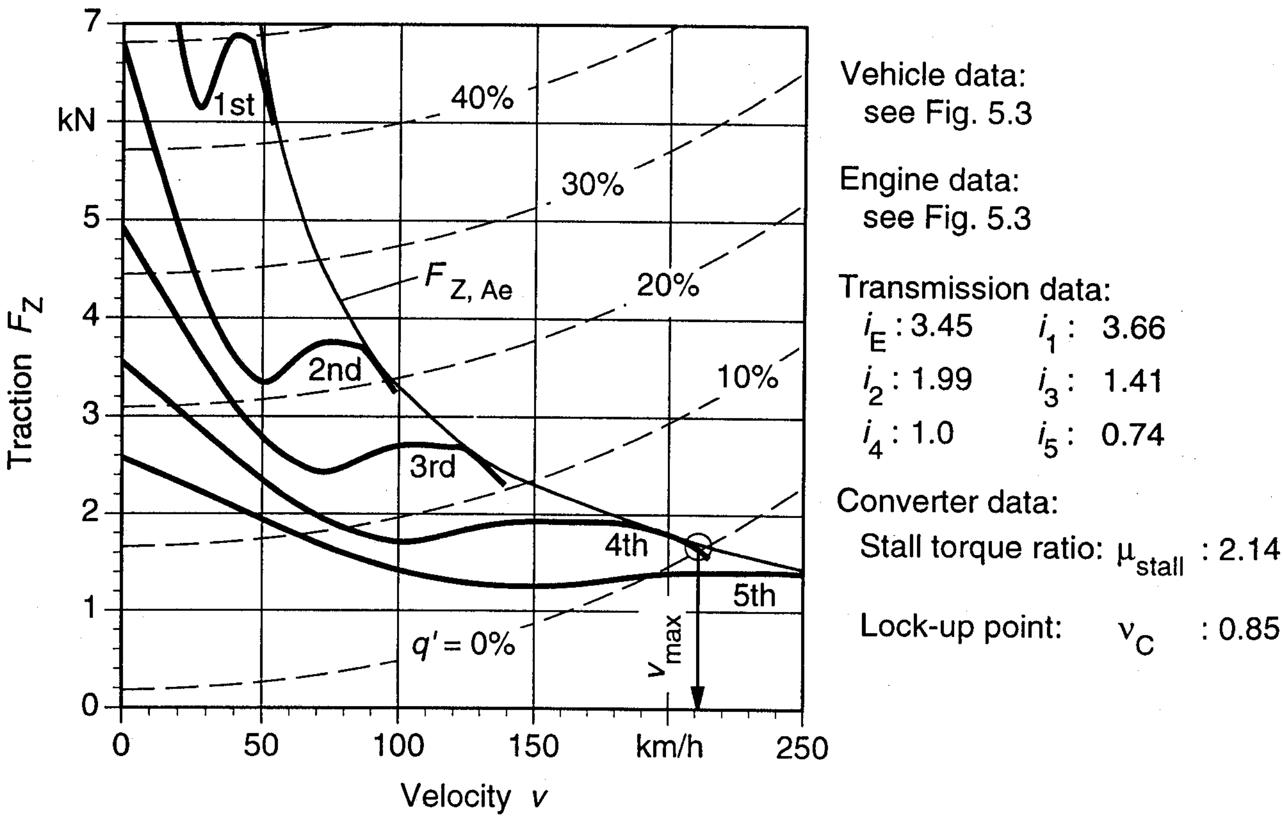


Figure 5.8. Traction diagram of the vehicle as shown in Figure 5.3 with 5-speed automatic transmission and Trilok converter

The traction curves can now be calculated from the transmission input values T_T and n_T, as described in Section 5.1.1. ν_C is the symbol for the speed ratio at the lock-up point. In the converter range of the Trilok converter, i.e. when $\nu < \nu_C$, $\mu > 1$. The combined action of engine and Trilok converter in a conventional automatic transmission is shown in Figure 5.8.

Figure 5.9 shows the turbine (or transmission input) torque curve for full load and part load (lines with the same accelerator pedal position). The lock-up point is displaced according to the pump parabola for ν_C. If the converter has a torque converter lock-up clutch *CC*, this is normally only engaged in the clutch range to enhance smoothness of operation, but it may also be engaged earlier. It is usually only disengaged in the torque conversion range. Controlled disengaging of the lock-up clutch ensures a smooth transition. The precise engagement and disengagement points of lock-up clutches depend on the engine's hum frequencies, and there is also the question of company philosophy.

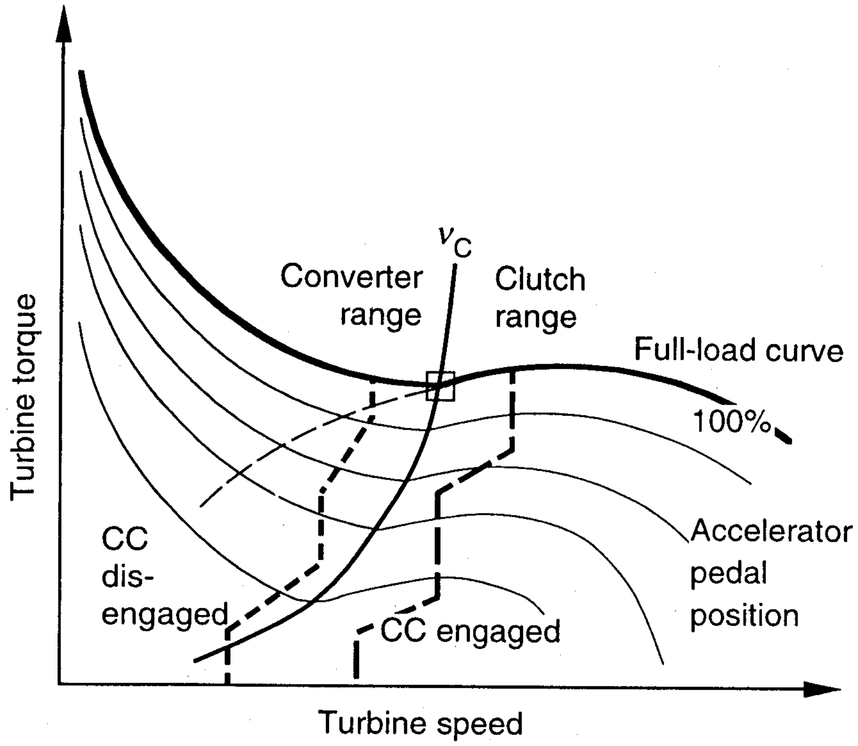


Figure 5.9. Co-operation of engine and Trilok converter at full load and part load

Traditional lock-up clutches are only engaged in the higher gears which are less sensitive to jolts during shifting or changes in load, resulting in improved ride comfort. When shifting in the lower gears it is possible to engage the lock-up clutch, or allow it to stay engaged, by using intelligent profile-controlled lock-up clutches [5.2]. If the lock-up clutch is kept engaged down to low turbine speeds and there is a spontaneous demand for traction, then a question arises of whether to disengage the clutch and cover the demand with the converter, or to shift down with the clutch engaged. This has to be decided when the shift control mechanism is being designed.

5.2 Vehicle Performance

The performance of a vehicle is defined by its maximum speed and its climbing and acceleration capability. The performance of a vehicle can be determined by comparing the traction available and the traction required at any point, as shown in Equation 5.1.

The procedure for determining the maximum speed, acceleration and traction of a vehicle is defined in standards (e.g. German standard DIN 70020). The performance data at the point of maximum engine torque, and at the point of maximum engine power are usually given in order to document the performance of a vehicle. Table 5.2 shows this for the vehicle used as an example in Figure 5.3. Table 5.3 shows some further driving performance and consumption data for the vehicle travelling at constant speed.

Table 5.2. Performance data of the vehicle used as an example in Figure 5.3. Velocity v, traction F_Z, excess traction $F_{Z,\,Ex}$, climbing performance q'_{max} and acceleration a_{max} at the point T_{max} = 189 Nm at 4750 1/min and $T(P_{max})$ = 179 Nm at 5930 1/min

Gear	v (km/h) at		$F_{Z,\,A}$ (kN) at		$F_{Z,\,Ex}$ (kN) at		q'_{max} (%) at		a_{max} (m/s²) at	
	T_{max}	T_n	T_{max}	T_n	T_{max}	T_n	T_{max}	T_n	T_{max}	T_n
1	45.8	57.2	6.9	6.5	6.6	6.2	49	46	3.7	3.4
2	83.5	104.3	3.8	3.5	3.3	3.0	23	21	2.0	1.8
3	127.0	158.5	2.5	2.4	1.8	1.4	12	10	1.2	0.9
4	170.1	212.4	1.9	1.8	0.8	0.2	5	1	0.5	0.1
5	213.2	266.1	1.5	1.4	–	–	–	–	–	–

Table 5.3. Driving conditions and consumption data for some constant speeds. Engine speed n_M, fuel consumption b_s and climbing performance q'_{max}

Gear	n_M (1/min) at (km/h)				b_s (l/100km)				q'_{max} (%)			
	30	60	90	120	30	60	90	120	30	60	90	120
1	3112	–	–	–	15.1	–	–	–	33.7	–	–	–
2	1706	3412	5118	–	8.5	9.8	11.5	–	15.7	16.4	18.3	–
3	1122	2244	3367	4489	6.8	6.6	7.9	9.8	8.7	10.3	10.6	11.3
4	837	1675	2512	3350	6.1	5.7	6.8	8.4	4.9	6.4	6.5	6.0
5	–	1336	2005	2673	–	5.1	6.0	7.4	–	4.7	4.4	3.9

5.2.1 Maximum Speed

The German standard DIN 70020 defines maximum speed as the greatest speed that a vehicle can maintain over a measured distance of 1 km. The main test conditions are:

- vehicle loaded with half the difference between the gross weight and the unladen weight,
- level, dry surface with good grip,
- maximum wind speed ± 3 m/s,
- the vehicle must travel along the test track in both directions without interruption.

The maximum speed may be found from the traction diagram at the intersection of the traction required curve with the traction available curve (Figures 5.3, 5.4, 5.6 and 5.8). The maximum speed in the various gears is determined by selecting the gear ratio of the individual gears of the gearbox.

5.2.2 Climbing Performance

Climbing performance is represented by the gradient resistance as defined in Equation 3.13. Uniform speed ($a = 0$ m/s^2) is assumed when determining climbing performance, so that the entire excess traction $F_{Z,\,Ex}$, as defined in Equation 5.2, is available to negotiate the gradient. The maximum climbing performance is given as

$$\sin \alpha_{St,\,max} = \frac{F_{Z,\,Ex}}{m_F\, g} \; . \tag{5.13}$$

It is normal practice to convert the angle of slope α_{St} into road gradient q' in percent, (Equation 3.14). Excess traction as a function of speed can be read from the traction diagram for the various gears. The climbing performance in each gear can thus be calculated from Equation 5.13, and plotted in a diagram as a function of speed (Figure 5.10a).

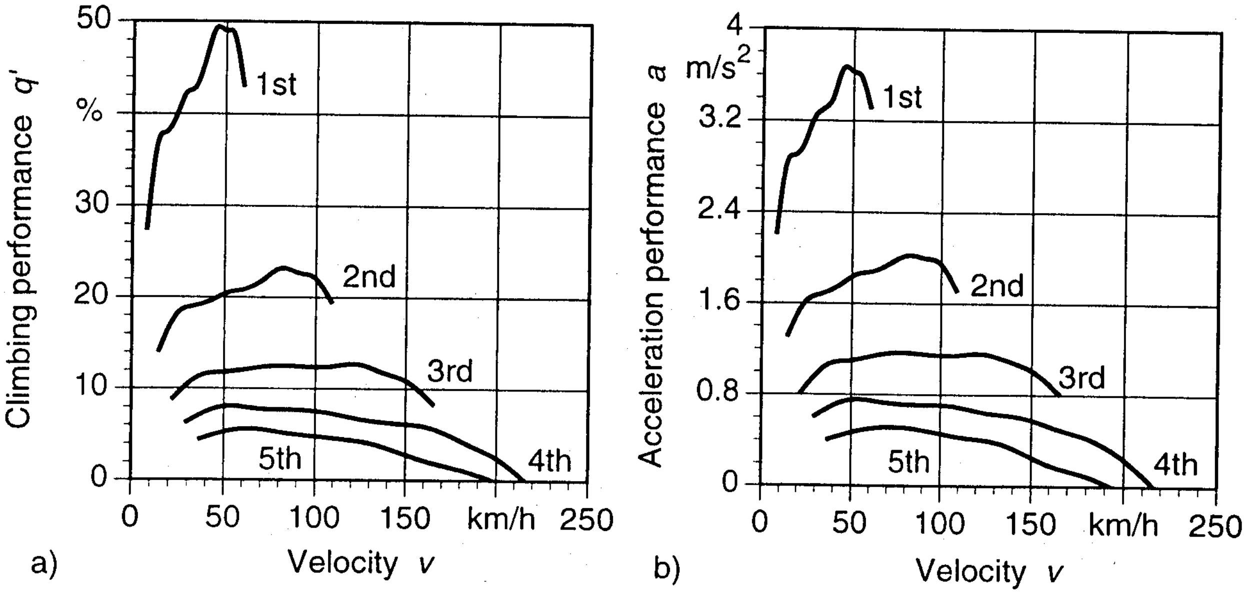


Figure 5.10. **a)** Dependence of climbing performance on gear; **b)** Acceleration of the test vehicle from Figure 5.3 dependent on gear

5.2.3 Acceleration Performance

Maximum acceleration on the level ($\alpha_{St} = 0$) can be derived from Equation 5.3:

$$a_{max} = \frac{F_{Z,\,Ex}}{m_F\, \lambda_n} \; . \tag{5.14}$$

The acceleration performance in each gear may be calculated using Equation 5.14 and the gear-dependent coefficient of rotational inertia λ_n (Figure 5.10b). In commercial vehicles the lowest gear is often given a high ratio to give the vehicle good climbing performance, even when fully loaded; the coefficient of rotational inertia can thus become very large, with the result that acceleration may be better in second gear than in first.

5.3 Fuel Consumption

Fuel consumption is a major factor determining the efficiency of a motor vehicle, but responsible utilisation of resources and pollution reduction are becoming increasingly significant factors. The fuel consumption of a vehicle is expressed either as

- consumption per distance travelled b_s in l/100 km, or
- consumption per unit time b_t in l/h.

It can be determined by calculation or by experiment. There are guidelines setting out the test conditions under which fuel consumption of passenger cars, trucks and buses must be measured [5.3]. As with maximum speed testing, the main test condition is loading the vehicle with half the difference between the gross weight and the unladen weight.

Real-world fuel consumption figures differ from those of the standard test cycles, due to the effects of driving style and road and traffic conditions, environmental factors and the condition and equipment of the vehicle. The driver has a major impact: the speeds and gears he selects determine the operating point of the engine and thus its fuel consumption.

5.3.1 Calculating Fuel Consumption (Example)

The specific fuel consumption b_e at any momentary operating point may be read off the engine performance map with constant specific fuel consumption characteristic curves (Figure 5.2). This requires the engine speed n_M and the associated engine torque $T(n_M)$. The engine speed is calculated from the road speed using Equation 4.11

$$n_M = \frac{v\, i_A}{2\,\pi\, r_{dyn}} \,. \tag{5.15}$$

The engine torque required $T_{Z,\,B}(n_M)$ is calculated, from the traction required at the wheels and the power-train efficiency, using Equation 3.22,

$$T_{Z,\,B}(n_M) = \frac{F_{Z,\,B}\, r_{dyn}}{i_A} \frac{1}{\eta_{tot}} \,. \tag{5.16}$$

If the engine map shows the effective mean pressure p_{me} in the cylinder instead of the engine torque or engine power, then Equation 3.26 may be used to perform the conversion. The required engine power $P_{Z,\,B}(n_M)$ is given by

$$P_{Z,\,B}(n_M) = F_{Z,\,B}\, v \frac{1}{\eta_{tot}} \,. \tag{5.17}$$

The fuel consumption per unit distance can then be calculated using Equations 5.16 and 5.17:

$$b_S = \frac{b_e\, P(n_M)}{\rho_{fuel}\, v} = \frac{b_e\, F_{Z,\,B}}{\rho_{fuel}\, \eta_{tot}} \,. \tag{5.18}$$

In this example the fuel consumption at the operating points marked *1* and *2* is calculated using the engine performance map in Figure 5.2.

Example

A vehicle with the data shown in Figure 5.3 is to travel on the level at a constant 150 km/h. Given that $f_R = 0.0145$ and $\rho_L = 1.199$ kg/m³, the traction requirement at the wheels is 862 N. With a power-train efficiency of $\eta_{tot} = 0.92$ (assumed constant), Equation 5.17 gives a required engine power of approximately 40 kW. If the vehicle travels at this speed in 3rd gear, then operating point *1* in Figure 5.2 shows a specific fuel consumption $b_e \approx 350$ g/kWh. Substituting into Equation 5.18, with a petrol density of $\rho_{fuel} = 755$ g/l, (Figure 3.16):

$$b_S = \frac{350\left[\frac{\text{g}}{\text{kWh}}\right]40\,[\text{kW}]}{755\left[\frac{\text{g}}{\text{l}}\right]150\left[\frac{\text{km}}{\text{h}}\right]} = 0.124\left[\frac{1}{\text{km}}\right] = 12.4\left[\frac{1}{100\,\text{km}}\right].$$

Driving in 5th gear (operating point *2* in Figure 5.2), $b_e \approx 270$ g/kWh. Substituting the traction requirement directly into Equation 5.18:

$$b_S = \frac{270\left[\frac{\text{g}}{\text{kWh}}\right]862\left[\text{N} = \frac{\text{Ws}}{\text{m}} = \frac{\text{kWh}}{\text{km}\;3600}\right]}{755\left[\frac{\text{g}}{\text{l}}\right]0.92} = 0.093\left[\frac{1}{\text{km}}\right] = 9.3\left[\frac{1}{100\,\text{km}}\right].$$

The driver can thus have a decisive effect on fuel consumption by his gear selection and the timing of gear shifts. Figure 5.11 shows the fuel consumption of the sample passenger car in each of the gears. In each gear there is a speed at which fuel consumption is optimal. Because air resistance increases as the square of speed, the power requirement, and thus fuel consumption, increases rapidly at high speed.

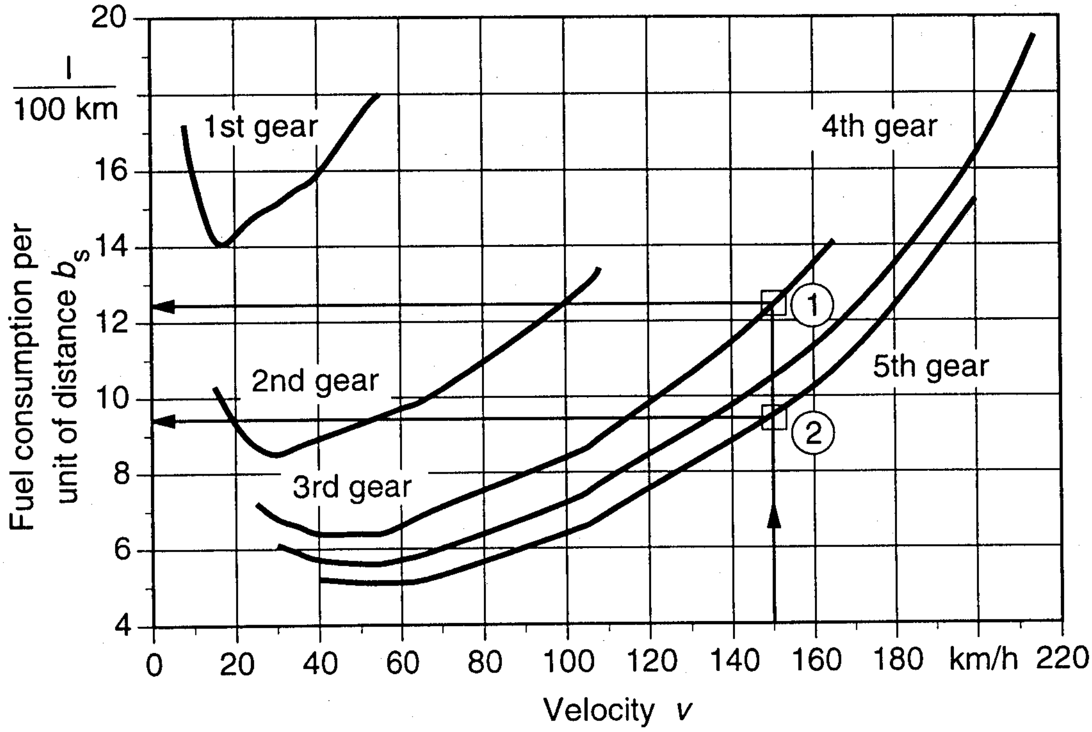


Figure 5.11. Fuel consumption related to gear for the vehicle used as an example in Figure 5.3. Operating point *1*, from Figure 5.2: (150 km/h in 3rd gear) gives a fuel consumption of 12.4 litres/100 km. Operating point 2 on the other hand gives 9.3 litres/100 km in 5th gear

In order to use Equation 5.18 to calculate fuel consumption for driving cycles with varying speed, the cycle must be broken down into small time intervals during which acceleration is assumed to be constant. The calculation of fuel consumption for standard cycles and at steady speed is an important application for driving simulation programs. See Chapter 15 "Computer-Aided Transmission Development, Driving Simulation".

5.3.2 Determining Fuel Consumption by Measurement

Fuel consumption is measured either on a roller test bench or in road tests. Three methods are used:

- gravimetric or volumetric measuring methods,
- flow measurement, and
- determining consumption from the carbon balance of the exhaust gas composition.

Standardised cycles are normally used to assess the consumption and emissions performance of vehicles, Table 5.4.

Table 5.4. Samples of important driving cycles specified for passenger cars

Cycle	Time (s)	Length (m)	ø-v (km/h)	v_{max} (km/h)	Special features	Aim
ECE cycle [5.3]	780	4 052	18.7	50	– Shift points specified	– Emissions – Urban consumption
CVS cycle FTP75 Urban cycle	1 877	17 884	34.3	91.2	– High speed – High acceleration	– Emissions – Urban consumption
Highway driving cycle HDC	765	16 463	77.5	88.5	– Typical for US highway	– Emissions – Motorway consumption

In these cycles the most important variable is the road speed profile, with any stationary periods, plotted over time. The cycles are either "simulated" on a roller test bench or reproduced on a test track driving at specified speeds.

In the USA the overall fuel consumption of all a manufacturer's vehicles, called "pool consumption", is regulated by law. There are no such requirements in Europe in 1998, but legal limits on fuel consumption are being considered.

Fuel consumption for trucks is normally measured on the road or by computer simulation. The wide variation in equipment and fittings makes it more difficult to compare consumption between commercial vehicles than between passenger cars. Calculations are normally based on the engine's specific consumption curve at full load. The fuel consumption of the test vehicles is usually also measured under part load at a series of constant speeds, such as 70, 80 and 95 km/h. For buses, especially schedule service buses, there are company-specific duty cycle specifications.

With commercial vehicles there is a goal conflict between road speed and fuel consumption. One possible evaluation option [5.4] is the efficiency factor:

$$\text{Commercial vehicle efficiency factor} = \frac{\text{Average speed } \varnothing v}{\text{Consumtion per unit distance } b_S} \,. \quad (5.19)$$

The higher the efficiency factor, the better.

5.3.3 Reducing Fuel Consumption

The transmission affects fuel consumption in two ways. One factor is its own transmission losses (Section 3.1.7), the other is providing suitable ratios for fuel-efficient utilisation of engine power. Geared transmissions are now so efficient that the former offers hardly any prospect of improvement. Transmission efficiency however remains a significant factor with continuously variable transmissions. But the main factor affecting consumption is still the driver! The main means available for reducing fuel consumption are as follows:

- Improving the efficiency of the internal combustion engine, in particular by reducing part-load consumption.
- Appropriate engine performance characteristics, i.e. the vehicle must be neither over-powered nor under-powered.
- Reducing driving resistance, for example rolling resistance and drag.
- Reducing the power draw of accessories such as servo pumps, air conditioning, etc.
- Improving the efficiency of the transmission. This relates principally to continuously variable transmissions, which includes torque converters.
- Adaptive control of ratio selection by automatic selection circuits and continuously variable transmissions.
- Traffic management systems to reduce stationary periods.
- Improved driving. Intelligent control systems, which protect the driver against his own misjudgement. There are many factors involved in determining how far this "usurping" of control can go.

5.3.4 Continuously Variable Transmissions

Unlike geared transmissions, continuously variable transmissions offer the possibility of selecting engine operating points on the demand power hyperbola according to a predefined strategy. The operating point is derived from the intersection of the line $T(P)$ with the control characteristic curve. In principle any point within the operating map covered by the overall gear ratio may be selected (Figure 5.12). In steady-state travel on the level, all the operating points are inside the operating map.

Transmission efficiency is a decisive factor with continuously variable transmissions (which here relates principally to pulley transmission). It is significantly worse than that of geared transmissions. This is partly due to the chain itself, but also the variable displacement pump and the contact pressure pump which the design requires; they are needed to ensure that the taper discs make contact at all power levels. The efficiency of the engine and the transmission offset each other. You could say "what the right hand gives the left hand takes away".

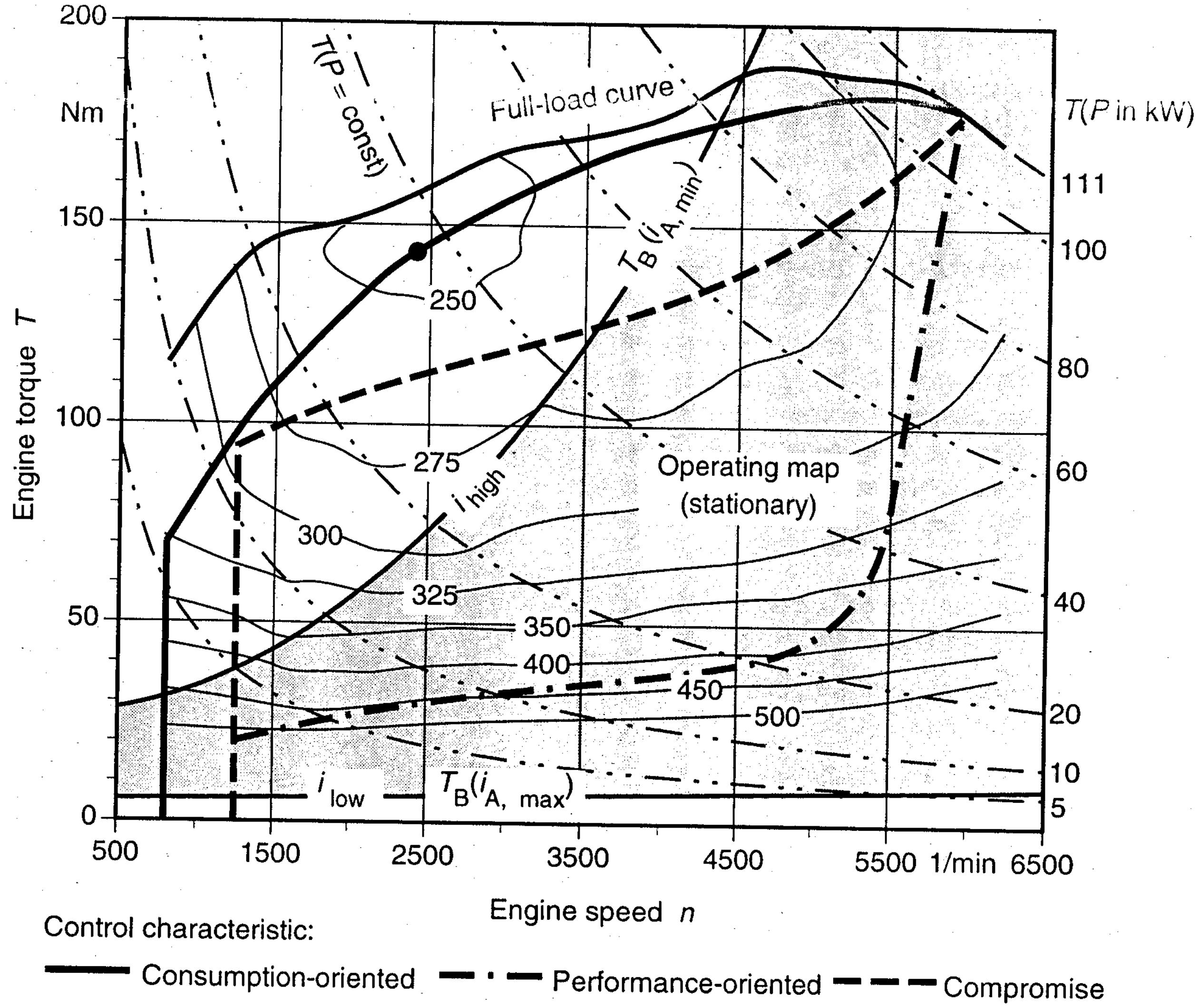


Figure 5.12. Engine performance map as shown in Figure 5.2. with control characteristics for a continuously variable transmission: economy-oriented, performance-oriented and a compromise solution

Different control characteristics result, depending on which of the two criteria is optimised – consumption or performance (Figure 5.12). The control characteristic optimum in terms of fuel consumption is given by the line of minimum fuel consumption. A control characteristic giving optimum performance gives rise to high excess traction at every operating point.

The control characteristic chosen is bound to involve a compromise taking into account driveability. A single control characteristic is not adequate for this purpose. Adaptive strategies are required for changing the ratio in line with the driving situation. See also Chapter 13.

5.4 Emissions

As with consumption, a distinction can be made in the case of emissions between those caused directly by the transmission (noise) and those which result from the engine operating point to which the ratio gives rise. The causes and remedies of transmission noise emissions are discussed in Section 7.5 "Developing Low-Noise Transmissions".

Emissions from internal combustion engines are harmful to the environment. The chief pollutants are

- ❍ carbon monoxide CO,
- ❍ nitrogen oxides NO_x,
- ❍ unburned hydrocarbons HC, and

with diesel engines also

- ❍ soot particulate.

Emissions of the greenhouse gas CO_2 are proportional to fuel consumption. Emissions for passenger cars are assessed and compared on the test cycles described in Table 5.4. The test establishes an average (g/test) value for each constituent pollutant.

In the case of commercial vehicle diesel engines, gaseous emissions are determined on a test bed. In the 13 Point Test to ECE-R49, 13 engine operating points are set, defined by engine speed and by engine load. The pollutant components are recorded at these operating points in (g/kWh) and given a weighting. The weighted emissions values for all 13 operating points are then averaged [5.5]. The legal limits of the EURO2 standard have applied to heavy trucks and buses since 1995/1996. New stricter limits enter into force in 1999, EURO3.

5.5 Dynamic Behaviour of the Power Train, Comfort

Customer expectations of smooth running are high, especially in the case of passenger cars and buses. This means paying particular attention to the vibration and thus, most importantly, the noise produced by the power train. The power train is an oscillatory system. The individual components have different masses, rigidities and damping factors. Mechanical substitute models of the power train can be developed using springs, dampers and inertial masses (Figure 5.13). The degree of complexity of the model relates to the purpose of the investigation.

The main source of oscillation is the irregular, sinusoidal running of the combustion engine (Figure 5.14). The drive torque of the combustion engine pulsates with the ignition frequency and stimulates these torsional oscillations.

With the emphasis in recent years on better fuel consumption and emissions performance there has been a sharp increase in the degree of irregularity of engine running, aggravating the oscillation characteristics of the power train. [5.6]. See also Section 7.5 "Developing Low-Noise Transmissions".

Hydrodynamic components in the power train (converter, retarder) have very good damping. For example in vehicles using a torque converter as the moving-off element, the torsional vibration of the engine is uncoupled from the transmission. As soon as the converter is bridged with a lock-up clutch, the vibration problems reappear.

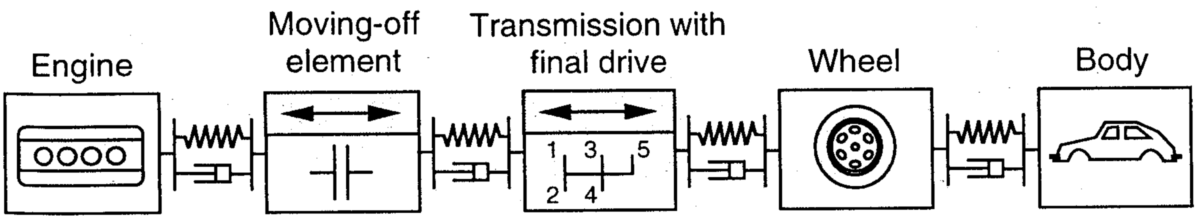


Figure 5.13. Simple vibration substitute model of a power train

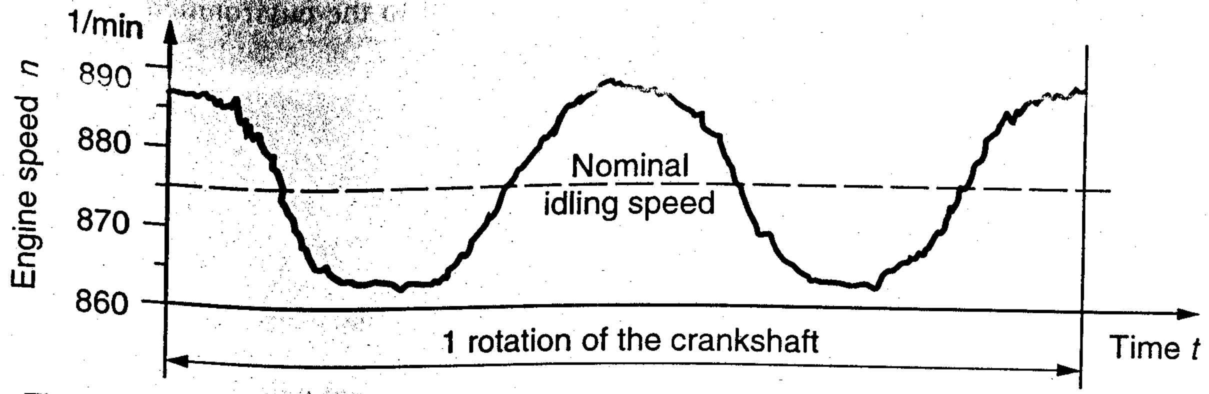


Figure 5.14. Internal combustion engine in neutral. Irregularity of a 1.8 litre 4-cylinder diesel engine during the course of one crankshaft revolution

The normal measures taken to decouple the engine from the power train are

- *torsion dampers* in the dry clutch driving disc,
- *dual mass flywheel*: elastic coupling of an additional mass,
- *dampers* to reduce resonance effects,
- *permanently slipping friction clutch* with electronic clutch systems: an electronically controlled adjusting mechanism filters out vibration peaks by using controlled slip,
- *torque converter* with automatic transmissions,
- *profile-controlled torque converter lock-up clutches [5.2].*

The transmission is not just a "passive" component in the power-train vibration system. Gearwheel transmissions themselves also cause parametrically excited vibrations. These mechanisms are explained in detail in Section 7.5. Vibration is also caused by the power transmission function of the universal joints as used in drive shafts. Random vibrations are also introduced to the power train (against the direction of power flow) from the road through the wheel.

The complex field of power-train dynamics is dealt with in greater detail in the literature [5.7, 5.8].

6 Vehicle Transmission Systems: Basic Design Principles

The ideal transmission design is determined
by the intended application, systematic thinking, and experience

This chapter presents a systematic exposition of basic design concepts for vehicle transmissions. These principles are related to specific examples in Chapter 12 "Typical Designs of Vehicle Transmissions".

6.1 Arrangement of the Transmission in the Vehicle

The important decisions at the concept phase of a new vehicle are the type of vehicle (saloon, coupé, sports car, etc.) and the type of drive (front-wheel drive, rear-wheel drive, etc.). The type of drive has a significant effect on handling, ride, economy, safety and space available. There are numerous factors influencing the design of the transmission, both with front-wheel and rear-wheel and with four-wheel drive. There are also four known alternatives for the relative position of the engine, transmission and final drive to each other. These basic configurations are set out below, based on the classification by transmission and type of use presented in Table 2.7.

6.1.1 Passenger Cars

The possible engine and drive configurations for passenger vehicles are shown in Table 6.1.

Table 6.1. Theoretical alternative engine and drive configurations

Configuration		Drive: Front	Drive: Rear	Drive: Front + Rear
Engine	Front	Front-wheel drive	Standard drive	4-wheel drive
	Rear	Not practicable	Rear-motor drive	4-wheel drive

The dominant technology for road-going passenger cars is currently *front-wheel drive* or *standard drive*. *Rear-motor drive* used to be common, but is now used mainly in sports cars. *Four-wheel drive* on the other hand is firmly established in new designs. Almost every range now includes a four-wheel drive model. Section 6.1.3 is devoted to a description of the great variety of four-wheel drive designs for passenger cars, both on-road and off-road.

The possible combinations of passenger car power-train components are shown in the morphological table (Table 6.2), including design variants for individual components.

Table 6.2. Morphological table for passenger car power trains

Parameters	**Configurations (passenger car)**				
Position of engine	Front – longitudinal	Front – transverse	Rear – longitudinal	Rear – transverse	______
Driven axle(s)	Front-wheel drive	Rear-wheel drive	4-wheel drive	______	______
Position of engine relative to gearbox	Engine in front of gearbox	Engine behind gearbox	Engine above gearbox	Engine beside gearbox	Engine longitudinal, gearbox transverse (T configuration)
Position of engine and gearbox relative to final drive	Engine, gearbox and final drive as a block	Final drive separate from engine and transmission (standard drive)	Engine separate from gearbox and final drive	______	______
Gearbox/final drive structurally combined	Final drive integral to gearbox	Final drive flange-mounted to gearbox	Gearbox and final drive separate	______	______
Final drive	Spur gears	Bevel gear helical bevel drive	Bevel gear hypoid bevel drive	Worm gears	Belt drive
Differential lock	Unlocked	Self-locking	Manual locking	Electronic locking	______
Differential gear	Spur gears	Bevel gears	Helical gears	Worm gears	______
Gearbox design	Single-stage	Two-stage, e.g. countershaft gearbox	Multi-stage, e.g. three-stage (3-shaft) gearbox	______	______

Figures 6.1 and 6.3 show various configurations of the power train and its components in the vehicle. Variants commonly encountered in practice are listed below.

Front-wheel drive, Figure 6.1:

- longitudinal engine in front of the axle, longitudinal transmission (Figure 6.1a),
- transverse engine beside the transmission (Figure 6.1d).

Standard drive, Figure 6.2:

- longitudinal engine in the front above/behind the front axle, transmission flanged to the engine longitudinally, final drive with differential to the rear axle (Figure 6.2g).

Rear-motor drive, Figure 6.3:

- longitudinal engine in front of the axle (Figure 6.3j),
- longitudinal engine behind the axle ("rear engine") (Figure 6.3k).

With rear-motor drive, a further distinction is made between rear-engine and mid-engine design depending on the position of the engine relative to the axle.

Transaxle design: The term transaxle is a general term for *transmission + axle*, referring to a combined transmission and axle drive unit. The concept accordingly applies to front-wheel drive as well as to transmissions with the final drive mounted at the rear axle. The term as used in Germany is more closely circumscribed, and relates only to the configuration of front longitudinal engine and transmission with rear-axle final drive (Figure 6.2h and Figure 6.2i). This incongruity of terms between German and English often leads to misunderstandings.

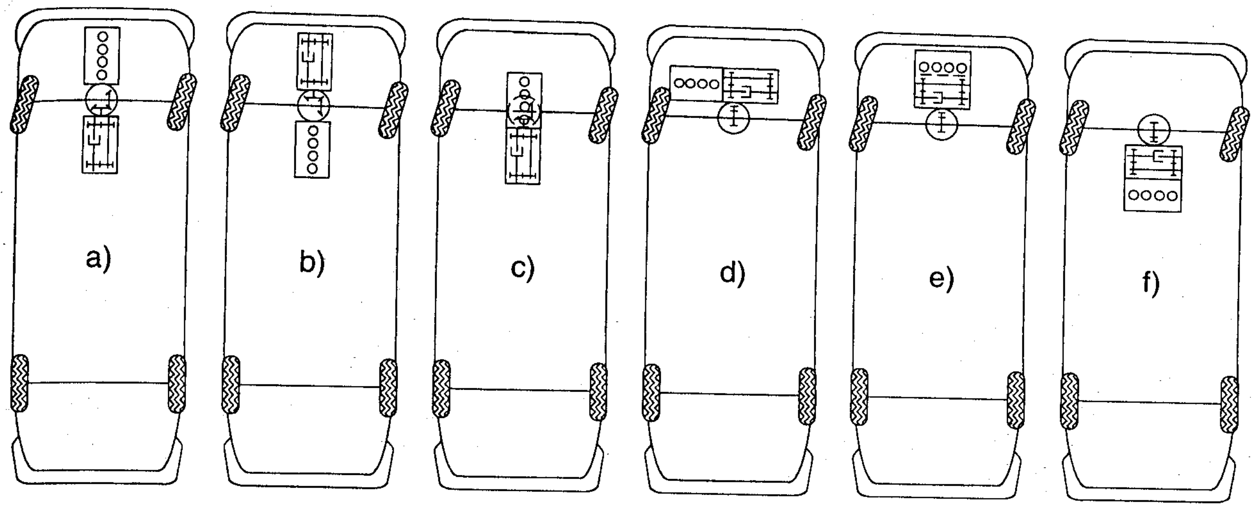


Figure 6.1. Front-wheel drive: **a)** Longitudinal engine in front of axle, longitudinal gearbox; **b)** Longitudinal engine behind axle, longitudinal gearbox; **c)** Longitudinal engine above axle, longitudinal gearbox; **d)** Transverse engine beside the gearbox; **e)** Transverse engine above the gearbox; **f)** Transverse engine behind the gearbox

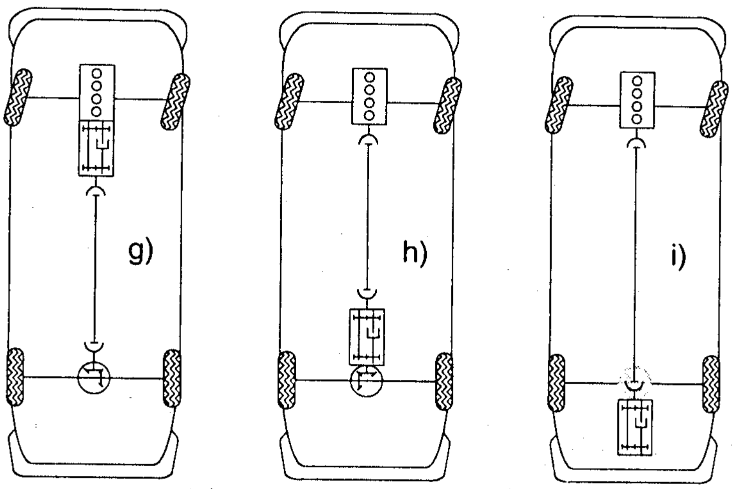


Figure 6.2. Standard drive: **g)** ("Standard drive") Longitudinal engine front mounted above/behind the front axle, gearbox flange mounted longitudinally to the engine, final drive with differential at the rear axle; **h)** Longitudinal engine front mounted above/behind the front axle, gearbox mounted longitudinally in front of or **i)** behind the rear axle with integral final drive and differential (transaxle principle).

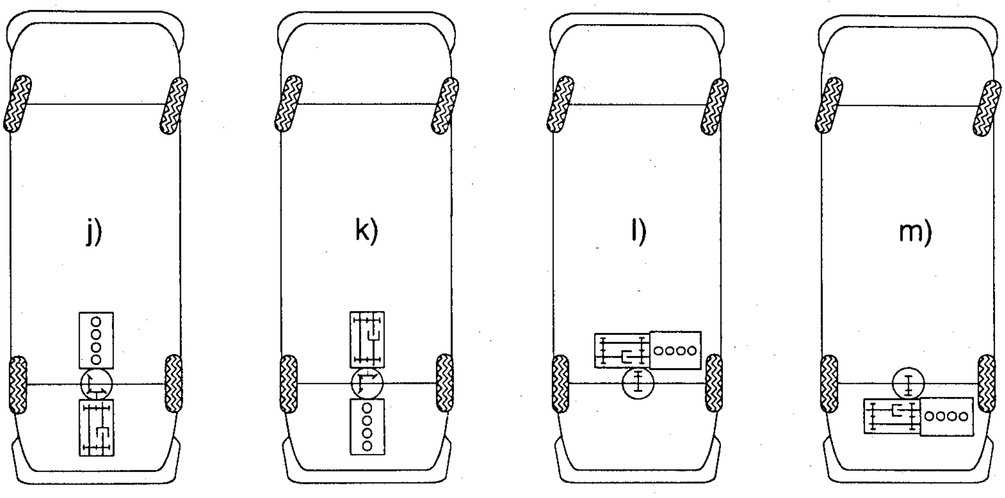


Figure 6.3. Rear-motor drive: **j)** Longitudinal engine in front of axle; **k)** Longitudinal engine behind axle ("rear engine"); **l)** Transverse engine beside the gearbox in front of the axle; **m)** Transverse engine beside the gearbox behind the axle

6.1.2 Trucks and Buses

The drive configurations for commercial vehicles (trucks and buses) up to approximately 4.0 t gross weight, are closely related to passenger car technology. The normal configurations are

- ❍ front-mounted longitudinal engine in front or/above the front axle, transmission longitudinally flanged to the engine, final drive with differential at the rear axle,
- ❍ transverse engine beside the transmission.

The all-wheel drives of trucks up to 4.0 t gross weight are likewise based on passenger car designs, and are explained in greater detail in Section 6.1.3.

For trucks with a gross weight of more than 4.0 t, the standard layout is now almost universal. In this arrangement, the engine torque is transmitted to the driving axle via a master clutch, gearbox and normally a cardan shaft (because of the rigid axle principle). Pure front-wheel drive, i.e. driving the vehicle through the front axle only (steering axle), occurs only very rarely in some bus designs and special-purpose designs. This option is not viable for heavy commercial vehicles because of the unfavourable front axle load distribution (less than 40% with two-axle vehicles, less than 30% with three-axle vehicles), and the resultant traction difficulties.

With multi-axle vehicles, almost all non-steering axles are driven; in the case of all-wheel drive (off-road mode) all axles are driven, including the steering axle. The normal applications for two-axle and three-axle vehicles are shown in Figure 6.4. The designation of the drive variants shown in Figure 6.4 is as follows: number of driven wheels (pairs of wheels) x the overall number of wheels (pairs of wheels). For example 2 x 4 represents two driven wheels with a total of four wheels. The engine can be located above or near to the front axle, and in some applications in the middle of vehicle (underfloor), which gives a favourable weight distribution.

Buses very often have a rear engine, either longitudinal or transverse. This allows greater freedom for designing the passenger accommodation. Figure 6.5 gives examples of drive configurations for buses.

Vehicles for use off-road and on building sites are almost always equipped with all-wheel drive. The engine is located at the front or in the middle of the vehicle, and the drive axles are steering drive axles at the front, and single or double rigid drive axles at the rear. The drive is delivered through a transfer gearbox (Section 6.8).

In contrast to passenger cars, the drive configuration of trucks and buses has little effect on the design of the vehicle transmission. The various possible power-train configuration options for trucks and buses can be derived from the morphological table (Table 6.3).

6.1.3 Four-Wheel Drive Passenger Cars

Four-wheel drive technology has established itself increasingly in recent years, for the following reasons:

- ❍ increased climbing performance,
- ❍ increased performance (traction) through full utilisation of static friction (only possible with four-wheel drive),
- ❍ improved moving-off traction,
- ❍ more precise handling,
- ❍ increased payload and trailer load,
- ❍ improved crash resistance due to energy absorption by the whole power train,
- ❍ identical roll steer effect under different weather conditions (dry surface, wet, ice and snow).

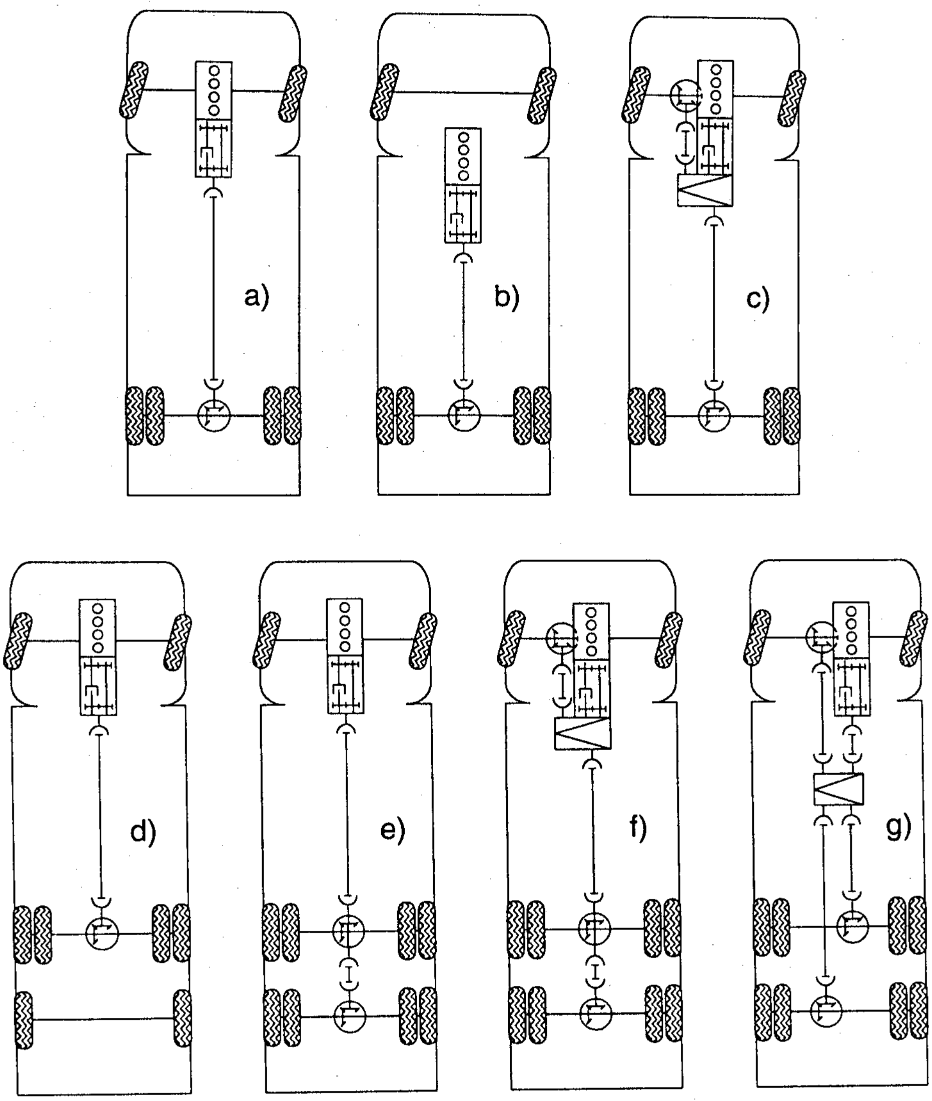


Figure 6.4. Drive designs for trucks with one or more powered axles: **a)** 2 x 4; **b)** 2 x 4, underfloor engine; **c)** 4 x 4, 4-wheel drive; **d)** 2 x 6, trailing axle; **e)** 4 x 6; **f)** 6 x 6, with through drive to second rear axle; **g)** 6 x 6, second rear axle with direct drive

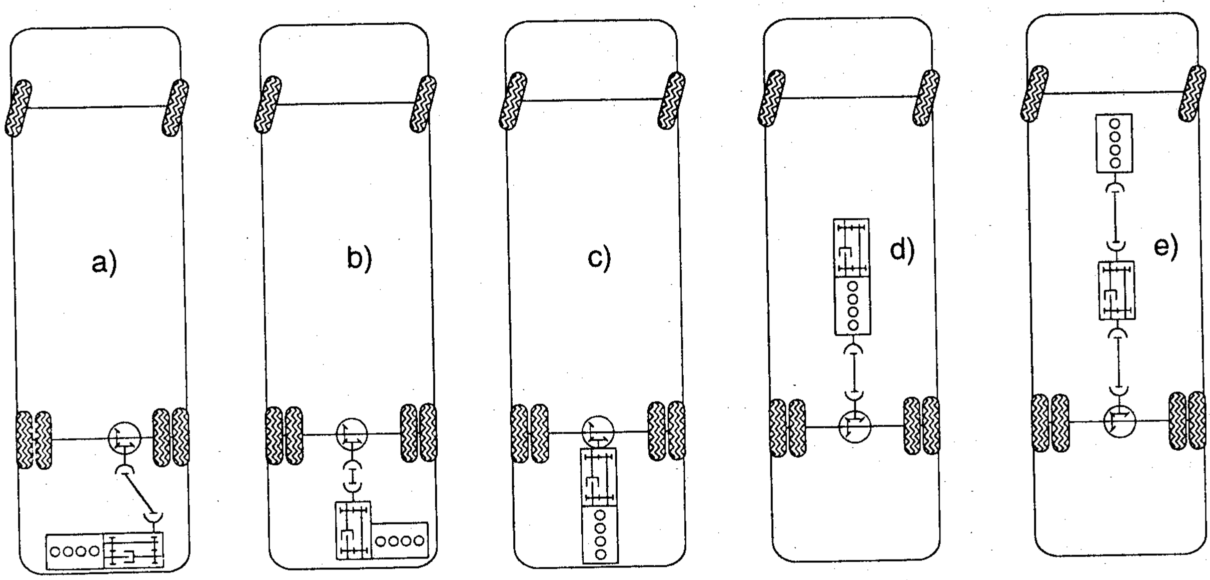


Figure 6.5. Drive designs for buses: **a)** Transverse engine behind axle, transverse gearbox; **b)** Transverse engine behind axle, longitudinal gearbox; **c)** Longitudinal engine behind axle, longitudinal gearbox; **d)** Longitudinal engine in front of axle, gearbox flange-connected longitudinally to engine; **e)** Longitudinal engine in front of axle, longitudinal gearbox

Table 6.3. Morphological table for truck and bus power trains

Parameters	**Configurations (trucks and buses)**				
Engine and gearbox configuration	Front-mounted – longitudinal	Front-mounted – transverse	Rear-mounted – longitudinal	Rear-mounted – transverse	Underfloor
Number of drive axles	One	Two	Three	———	———
Type of axle	Drive axle with/without through drive	Steering axle	Driven steering axle	Trailing axle	———
Through drive to second axle	Yes	No	———	———	———
Final drive	Bevel gear – helical bevel gear drive	Bevel gear – hypoid bevel drive	Worm gears	Double bevel gear	———
Reduction in centre drive	Single	Spur wheel reduction, shiftable	Spur wheel reduction, non shiftable	Planetary reduction, shiftable	Planetary reduction, non shiftable
Differential gear	Spur gears	Bevel gears	Helical gears	Worm gears	———
Differential locking	Unlocked	Self-locking	Manual locking	———	———
Hub gear	Without hub gear	Spur gear reduction, external toothed	Spur gear reduction, internal toothed	Planetary reduction, flat	Planetary reduction, spatial

The principal corresponding disadvantages are greater technical complexity, increased weight (fuel consumption) and increased space requirements.

In addition to dedicated four-wheel drive designs for off-road passenger cars and special vehicles (e.g. rally vehicles), four-wheel drives have been designed based on the following drive configurations [6.1–6.5]:

- front-motor drive with longitudinal engine in front of the axle,
- front-motor drive with transverse engine beside the transmission,
- standard drive with longitudinal engine over the front axle, longitudinal transmission flanged onto the engine, final drive with differential at rear axle,
- rear-motor drive with longitudinal engine behind the axle ("rear engine").

Figure 6.6 shows these drive configurations. Various alternative all-wheel drive designs are given in Section 12.7. Figure 6.6 shows that the all-wheel drive design is very largely dependent on the drive concept of the original vehicle. Further characteristic features are defined by the objectives when introducing all-wheel drive. For example a crucial factor for engineering complexity and thus for cost is whether the purpose of all-wheel drive is to improve handling, or merely to provide improved drive-away traction. The appropriate type of all-wheel drive can be selected from the systematic classification of all-wheel drives in Figure 6.7. The decisive criterion is the type of link between the two axles to be driven. The power flow between the front and rear axles can be clutch controlled ("hang-on system") or differential controlled. See also Section 6.8 "Transfer Gearboxes" and Section 6.10 "Differential Gears".

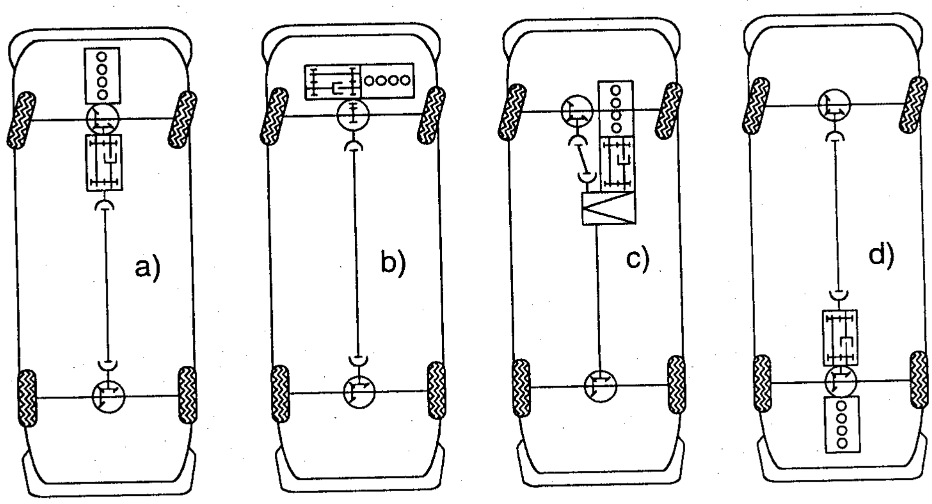


Figure 6.6. Drive designs for four-wheel drive passenger car: **a)** Front-motor drive with longitudinal engine in front of the axle; **b)** Front-motor drive with transverse engine beside the gearbox; **c)** Standard drive with front-mounted longitudinal engine over the front axle, longitudinally mounted gearbox flange mounted on the engine, direct drive at the rear axle; **d)** Rear-motor drive with longitudinal engine behind the axle

Differential-Controlled All-Wheel Drive

In *differential-controlled* systems, torque is distributed to the front and rear axle by a planetary gear differential or a bevel gear differential. With planetary gear differentials the drive torque can be split to the two drive axles as required, by selecting the ratio. Typical torque distribution ratios between front and rear axle are 50% : 50% to 33% : 66%. In bevel gear differentials, the torque distribution is fixed at 50% : 50%. Selecting a fixed torque ratio between the front and rear axles means the traction distribution is ideal for only one point, the design point.

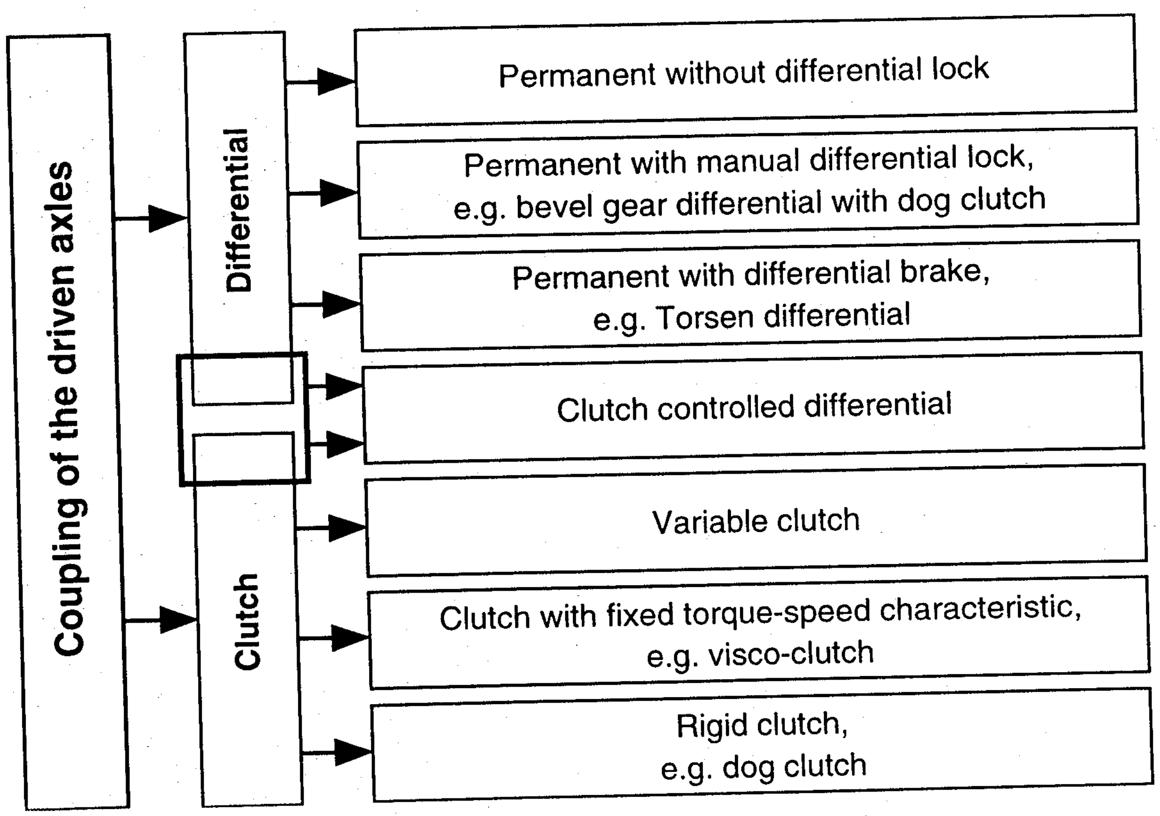


Figure 6.7. Systematic classification of passenger car all-wheel drives

The drive torque is thus not split in proportion to the axle load required by the current driving conditions. If the traction reserves are completely exhausted when there is a high degree of slip (which is theoretically only possible with variable torque distribution between the front and rear axle), the interaxle differential can be braked or locked. With a locking effect which engages as the speed difference increases (e.g. visco-lock), handling is not impaired, preventing permanent distortion of the power train such as can occur with positive locks. A TORSEN transfer differential (TORSEN stands for "torque sensing") acts in this regard as a self-locking differential.

Clutch-Controlled All-Wheel Drive

Clutch-controlled all-wheel drives are characterised by the fact that only one axle is permanently driven. The second axle is engaged manually or automatically as required. The most inexpensive option for engaging the second axle is to use a rigid controllable clutch. But this system can only be used for moving off, since deformation occurs in the power train with 100% lock, as in the case of the centre differential (see Figure 6.50).

The use of a visco-clutch gives another way of linking two axles, building up the coupling torque between the front and rear axles through viscous friction (slip controlled) depending on the speed differential between the front and rear axles. The transition to all-wheel drive is gradual as the speed differential between the front axle (FA) and the rear axle (RA) increases. Permanent distortions in the power train are not possible. The level of coupling torque depends on the clutch characteristic selected. It can be influenced by the level, viscosity and temperature of the oil (using silicone oil with low viscosity change with temperature fluctuations). A "soft" coupling characteristic (i.e. low coupling torque with high relative rotational speed) is desirable to avoid distortions in the power train, whereas a "hard" coupling characteristic (i.e. large coupling torque with small relative rotational speed) is necessary when an axle is spinning. The visco-clutch is protected against destruction under high load by the *hump effect*, i.e. heat-related friction engagement between the plates [6.6 and 6.7].

The last type of clutch controlled system involves clutches with externally adjustable coupling torque (e.g. multiplate clutches), where the coupling torque can be selected to match current driving conditions. This enables torque distribution between the front and rear axles to be adapted to the changes in dynamic axle load, i.e. depending on acceleration, gradient, loading, etc.

Hybrid

A further variant, located between the differential-controlled and clutch-controlled systems, is all-wheel drive using an "electronically controlled" multiplate clutch and a "lockable" differential.

Table 6.4. Generations of passenger car all-wheel drives, derived from [6.4]

Generation	Power split between FA and RA	Example of differential-controlled system	Example of clutch-controlled system
1	Torque distribution constant	Spur gear transfer differential	Selectable all-wheel drive
2	Torque distribution along system-dependent characteristic curve	TORSEN transfer differential	Visco-clutch, unregulated
3	Torque distribution along controllable characteristic curve	Transfer differential with controlled multi-disc lock	Visco-clutch, regulated
4	Torque distribution selectable	Vario drive	Multi-disc clutch regulated for FA and RA

This solution gives a high degree of ease of use with identical handling (two-wheel drive) of the basic vehicle. This "traction control system" however involves a high level of engineering complexity.

All-wheel drives for passenger cars are often divided into different generations. Table 6.4 describes the characteristic features of the particular all-wheel drive generation, giving an example. Table 6.5 shows a morphological table for passenger car all-wheel power trains.

Section 6.10 "Differential Gears" examines the function and design of the various types of differential and differential lock, including a description of the visco-clutch that acts both as a clutch and as a differential lock.

Table 6.5. Morphological table for passenger car all-wheel drives

Parameters	Configurations (4-wheel drive passenger cars)				
Position of engine	Front – longitudinal	Front – transverse	Rear – longitudinal	Rear – transverse	
Concept	Manual selection	Automatic engagement	Permanent		
Torque transmission FA - RA	Rigid clutch	Multi-disc clutch: regulated/ unregulated	Centrifugal clutch	Visco-clutch	Transfer differential
Torque distribution	Constant, e.g. 50% : 50%	Slip-dependent	Regulated according to wheel load		
Type of interaxle differential lock	Rigid clutch	Self-locking differential	Visco lock	Multi-disc clutch unregulated	Multi-disc clutch regulated
Type of interwheel differential lock	Rigid clutch	Self-locking differential	Visco lock	Multi-disc clutch	Locking function activated by brake action
Transfer differential	Spur gear differential	Bevel gear differential	Other		
Transfer box	Integrated in gearbox	Individual subassembly	Flange mounted on the gearbox		
Braking stability	Free-wheel	Multi-disc clutch			

6.1.4 Transverse and Longitudinal Dynamics with All-Wheel Drive

Section 6.1.3 lists some of the advantages of all-wheel drive compared to two-wheel drive. To further illustrate the advantages of all-wheel drive, the longitudinal and transverse dynamics of the vehicle and/or the tyre are examined below. This requires an understanding of the fundamental connection between circumferential force and lateral force on the wheel at high lateral acceleration. Wheel load, wheel load fluctuations, and torque and slip angle alignment are ignored for the sake of simplicity.

Figure 6.8 shows the conditions for a wheel at the friction limit for two-wheel drive (left) and for four-wheel drive (right). With four-wheel drive the circumferential force $F_{U2} = F_{U1} / 2$, given simplified assumptions. The wheel can transmit greater lateral forces F_S, until it reaches the friction limit at F_{res}.

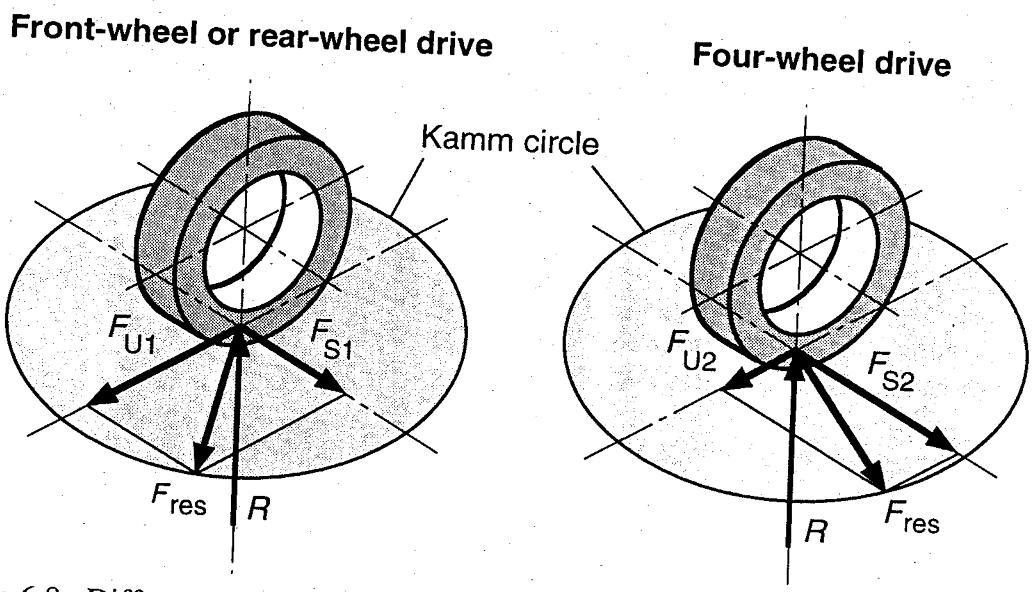


Figure 6.8. Difference between two-wheel drive and four-wheel drive

The maximum transmittable circumferential force $F_{U,\,max}$ is derived from Equation 3.9 as

$$F_{U,\,max} = \mu_H\ R\ . \tag{6.1}$$

The maximum lateral force $F_{S,\,max}$ is

$$F_{S,\,max} = \mu_H\ R\ . \tag{6.2}$$

When the circumferential force F_U and the lateral force F_S both occur simultaneously, they make up a geometrical sum (see Figure 6.8). It must not exceed $F_{res} = \mu_H\ R$ (KAMM circle). The Kamm circle represents the friction limit for the rolling wheel transmitting both circumferential and lateral forces at the same time. The following relationship applies:

$$F_{res} \geq \sqrt{F_U^2 + F_S^2}\ . \tag{6.3}$$

6.2 Transmission Formats and Designs

Completed transmissions are distinguished in terms of *format* and *design*. Transmission format relates to the morphology or external appearance of the transmission, or the configuration of input and output. The transmission design describes how the main functions of the transmission are engineered. It relates to the internal configuration. Transmissions can thus have different designs with the same format. The format selected for a design depends on various criteria; principally the vehicle design, the type of engine and the intended use (Figure 6.9).

6.2.1 Transmission Format

The format of the transmission (Figure 6.10) is determined primarily by the position of the transmission in the vehicle or in the power train (Section 6.1), and any additional geometrical constraints such as space limitations.

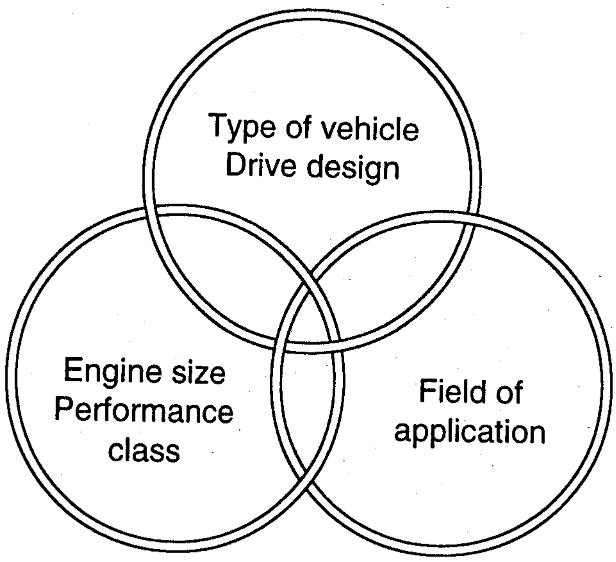


Figure 6.9. Specification factors for transmission format and design

The format is also affected by assembly considerations (both as regards the transmission itself and as regards its installation in the vehicle), by gearbox housing rigidity and noise emissions. Transmissions often comprise several individual gearboxes, which can also be housed in separate gearbox housings. In this case, the relative position of the individual housings is an important factor influencing the format of the transmission as a whole.

The format of a transmission concerns the design engineer principally when adapting or developing existing designs, for example adapting an existing transmission to a new vehicle with different dimensional constraints.

With standard drive (front-mounted longitudinal engine and transmission, rear wheel drive, Figure 6.2g), the coaxial transmission format is used. If there are two driven rear axles, or if all-wheel drive is used, then a transfer box is needed, which may be flange-mounted directly onto the gearbox or separate from it.

For front-wheel drive layouts, a transmission format is used in which the axle gearbox is integrated into the gearbox with the differential. Input and output are not coaxial in this case.

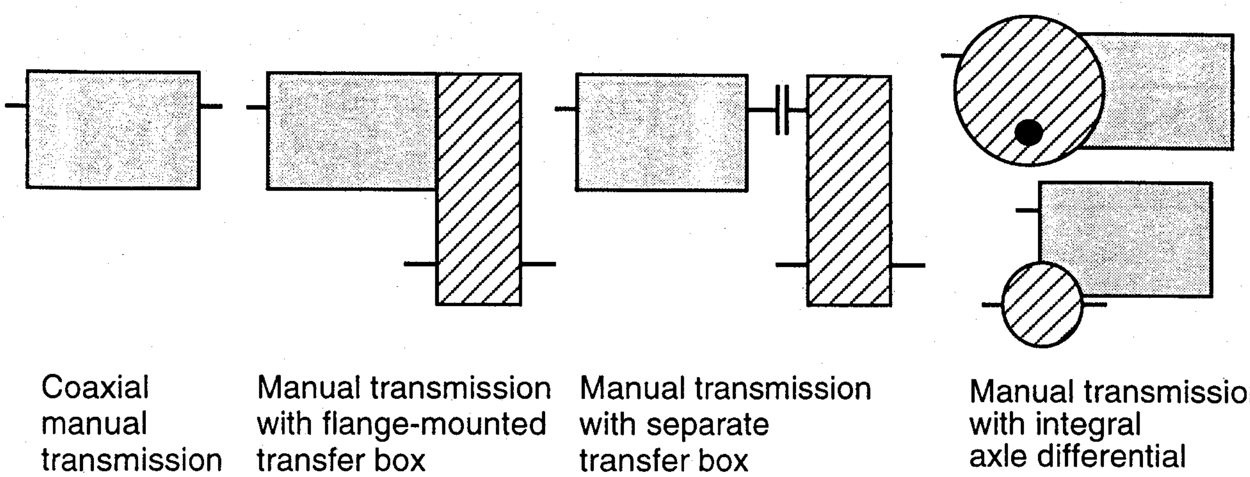


Figure 6.10. Examples of different transmission formats

6.2.2 Transmission Design

The transmission design is derived from the functional principles applied, to fulfil the main functions of the transmission. As already indicated in Section 2.3.3, a vehicle transmission has four main functions: "Moving off from rest", "Changing ratio/rotational speed", "Shifting/establishing power flow" and "Operating/controlling the gearbox".

The "Moving off from rest" function can be carried out mechanically, electro-mechanically or hydraulically. The "Changing ratio/rotational speed" function can be carried out using spur gears, planetary gears, hydrodynamic or hydrostatic transmissions or mechanical continuously variable transmissions. The "Shifting/establishing power flow" function can be divided into the two functional principles positive engagement or frictional engagement. The "Operating/controlling" function can be carried out by manual shifting or by an automatic system with associated control unit.

Their selection depends on the power to be transmitted, considering traction utilisation and ease of operation. Especially in the case of new developments, the design engineer has to decide the design or combination of designs of the transmission.

Hybrid designs are in principle always an option for carrying out the various main functions. In the last 100 years numerous possible solutions have been proposed for vehicle transmissions. These can be systematically represented in a morphological table (Table 6.6). The main functions are shown in the four rows of this table, and the associated design principles applied appear in the columns. By combining these principles to form a complete transmission, you get all possible combinations of transmission designs. Not all theoretical combinations are of significance or relevance in practice.

A preliminary selection can be made by assessing the design under consideration, and other alternatives. This preliminary selection follows on from the concept phase of transmission development.

In multi-range transmissions (Section 6.7.2), these main functions can take different forms for each individual range unit. Each individual range unit must have the following main functions: "changing ratio/rotational speed", "shifting/establishing power flow" and "operating/controlling the gearbox". Even with multi-range change transmissions, only one approach is used for the main function of enabling "moving off". The number of functional principles and their physical principles of operation can change as technology advances.

Table 6.6. Morphological table of approaches for the main functions. The principles underlying a conventional manual gearbox are highlighted in grey

Main function \ Principle of operation	1	2	3	4	5
Enable moving off from rest	Mechanical Dry	Mechanical Wet	Electro-mechanical	Hydro-dynamic	Hydro-static
Change ratio/speed	Spur gears	Planetary gears	Hydro-dynamic	Hydro-static	Mechanical continuous
Shift/establish power flow	Positive-engaged Sliding gears	Positive-engaged Shifting dog, synchronised	Positive-engaged Shifting dog, unsynchronised	Frictional engaged Multi-disc clutch	Frictional engaged Multi-disc brake
Activate/control the transmission	Manual shift	Manual shift, servo-assisted	Semi-automatic	Automatic hydraulic	Automatic electr./hydr.

6.3 Basic Gearbox Construction

Geared transmissions are categorised by their technical design or the number of ratio steps making up the individual gears:

- single-stage transmissions,
- two-stage transmissions,
- multi-stage transmissions.

The term "stage" refers here to a gear pair, or the power flow from one gearwheel to another. A stage generally involves power flow from one shaft to another. Figure 6.11 shows designs of three- and four-speed countershaft transmissions. The term "countershaft transmission" is defined in Section 6.4.

Single-stage transmissions are primarily used in front-wheel drive vehicles, since they require no coaxial transmission of the power flow, unlike standard drive or four-wheel drive vehicles.

In the standard power-train configuration (engine and transmission in the front, drive at the rear), the two-stage countershaft transmission with coaxial input and output shaft is virtually universal.

Multi-stage (more than two-stage) transmissions are just as suitable as single-stage transmissions for front-engine front-wheel drive vehicles. The number of gear stages they have depends upon the number of gears. The multi-stage design enables short gearboxes to be constructed. Multi-stage coaxial transmissions are used principally in commercial vehicles with front or rear-mounted range transmissions (see Section 6.7.2).

To decide on the type of transmission for a particular application, first the basic ratio change options need to be defined. The shifting elements involved also by definition constitute part of the transmission.

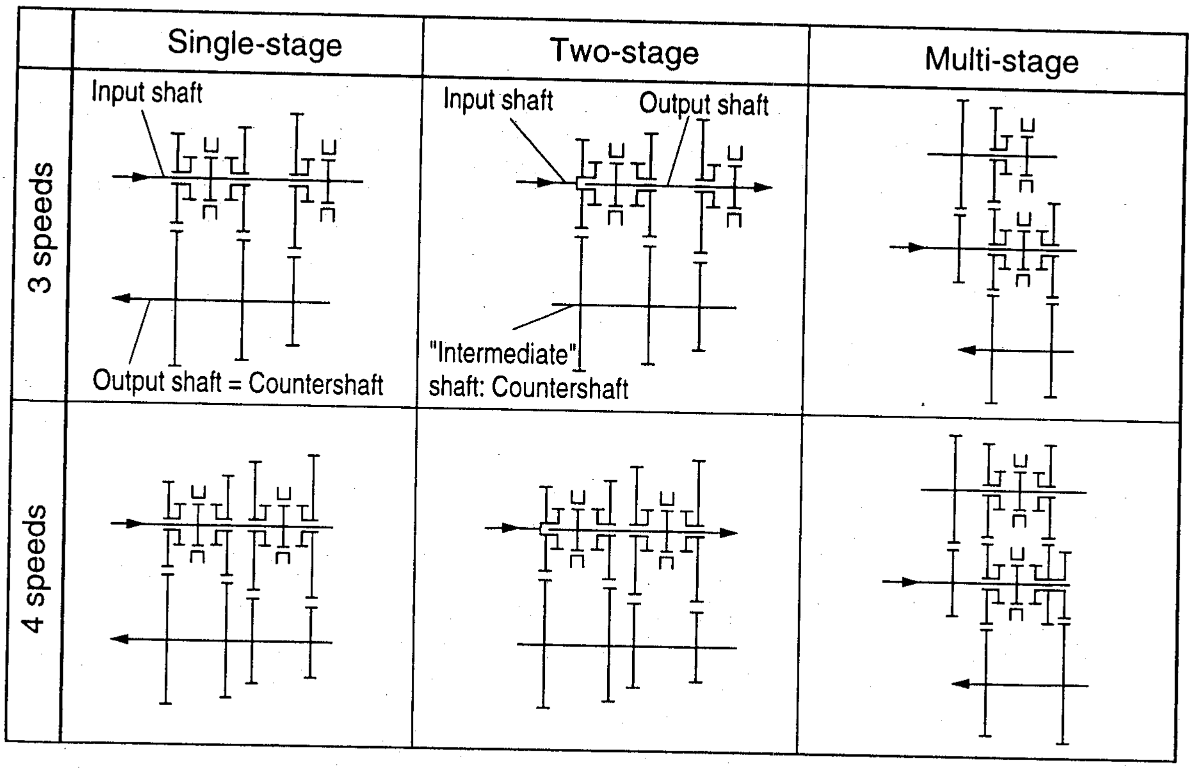


Figure 6.11. Configuration of the ratio stages for 3-speed and 4-speed gearboxes

6.3.1 Shifting with Power Interruption

The transmission is shifted without load, i.e. the power flow between the prime mover and the wheels is interrupted during the gear change operation. The vehicle coasts during the gear change operation. This can entail a loss of speed (Figure 6.12), depending on the difficulty of the terrain (gradient, high rolling resistance). In order to limit this loss of speed, the shifting operation must not take too long; the whole gear change operation must therefore be concluded in less than one second. For multi-range transmissions this means that the gear change operations in the individual range units must be carried out within 0.2 to 0.3 seconds (assuming they are in succession). This is one reason why the number of ranges in a transmission cannot be increased indefinitely, although this would lead to a reduction in the number of gear pairs needed (see also Section 6.7.2). The requirement for several individual shifting actions to occur synchronously at the various shifting points is demanding in engineering terms.

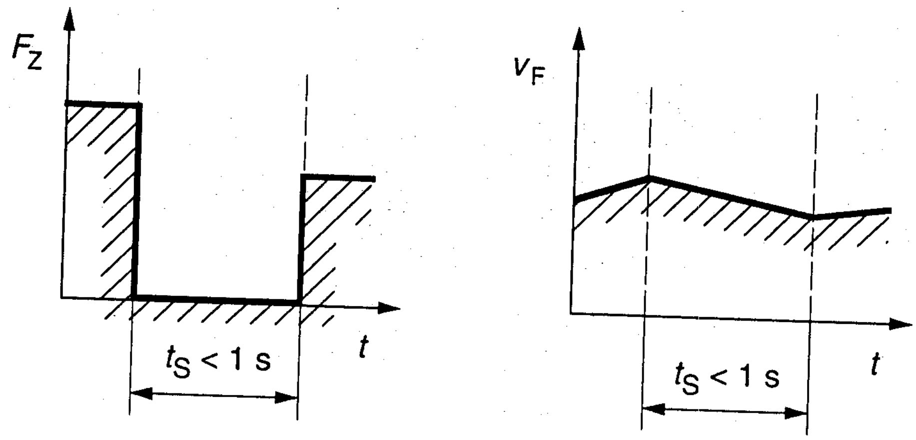


Figure 6.12. Qualitative traction and velocity profile when shifting up with power interruption

Transmissions with power interruption can be used wherever the application is such that vehicle speed does not decrease (or on downhill runs increase) significantly during the shifting process, and the shifting operation is reasonably practical for the driver.

In the case of fully automatic countershaft-type truck transmissions, shifting normally involves power interruption. Vehicle acceleration forces are relatively low, the vehicle mass is high, and ride quality is not the top priority.

Ride quality considerations make shifting without power interruption essential for fully automatic passenger car transmissions, whether countershaft or planetary gear transmissions.

6.3.2 Shifting without Power Interruption

As in the case of shifting with power interruption, the transmission ratio is changed in stages. But in this case, the power flow is not interrupted during the gear change operation (Figure 6.13).

Such transmissions are known as frictional transmissions or powershift transmissions. The transition from one ratio step to another is carried out without interrupting the power flow. The ratio steps can be engaged under load by means of additional braking or clutch elements. In this case the gear-set which is being shifted out of is disengaged from the power flow, whilst the new gear-set is engaged in the power flow.

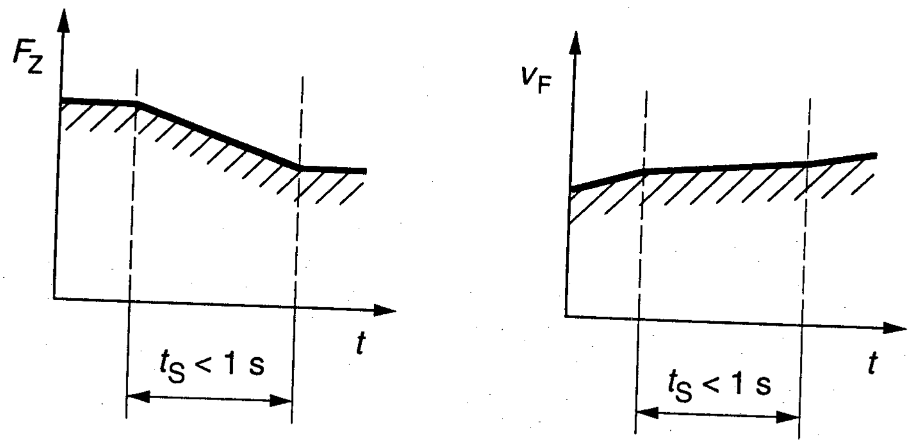


Figure 6.13. Qualitative profile of traction and velocity for upward powershift

There is no reduction in road speed. Examples of this type of transmission are conventional automatic transmissions and twin clutch transmissions. Powershift transmissions are well suited for fast shifting. Transmissions of this type are used in heavy vehicles, where vehicles operate in difficult terrain, and in all vehicles where the driver is to be relieved of gear shifting activity. They are fitted both with manual and with automatic gear selection.

6.3.3 Continuously Variable Transmissions without Power Interruption

Here ratio shifting is no longer in steps, but varies continuously (see also Sections 5.3.4 and 6.6.4). The traction is adapted to the tractional resistance without any intervention by the driver (Figure 6.14). This type of characteristic output conversion represents the theoretically ideal solution. The best known current example of this principle of operation is the hydrodynamic torque converter. Various mechanical variants are known in the form of friction gears or pulley transmissions. The mechanical variants are always based on converting the rotational speed to continuously variable diameters.

Hydrostatic transmissions comprising a combination of pump and motor also provide continuously variable regulation of rotational speed. Usually a hydrostatic transmission is coupled to a planetary gear to increase the overall gear ratio and to preselect different operating ranges, some with power split.

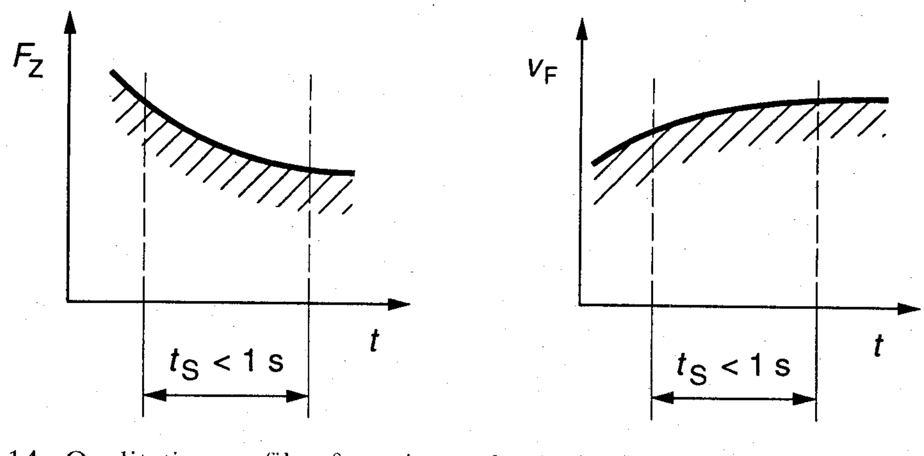


Figure 6.14. Qualitative profile of traction and velocity for continuously upward "powershift" with CVT

6.4 Gear-Sets with Fixed Axles, Countershaft Transmissions and Epicyclic Gears

Geared transmissions are divided into:

- gear-sets with fixed axles and
- epicyclic gears.

These terms relate to the axes of the gearwheels engaged in the transmission. In the case of *gear-sets with fixed axles* the positions of the axes of all the gearwheels in the transmission are fixed relative to the gearbox housing. In *epicyclic* or *planetary gear-sets*, a revolving bar (spider) carries the axes of the planetary gears.

Countershaft transmissions: The term countershaft transmission [6.8] relates to a transmission with only one input shaft and only one output shaft, and a countershaft running in a fixed position in the housing (Figure 6.11). Countershaft transmissions are thus gear-sets with fixed axles. In the case of single-stage countershaft transmissions, the output shaft and the countershaft are combined so they could also be called "reduced" countershaft transmissions.

Planetary transmissions: In planetary transmissions there are always three or more planetary gears on a spider (Figure 6.15) to ensure uniform and lower stress. The number of planetary gears and the number of teeth on each have no effect on the transmission ratio; they merely reverse the direction of rotation at this point. The axes of the planetary gears thus complete a rotational movement around the main axis of the gearbox.

There are also hybrid designs combining elements of gear-sets with fixed axles and epicyclic gears. Where the location of the spider is fixed, an epicyclic gear by definition becomes a gear-set with fixed axles.

Planetary transmission provide nine combinations of possible states of motion in one planetary gear-set. These are derived from the fact that in principle the position of the ring gear, the spider or the sun wheel can be fixed, to act as a "frame". The two remaining transmission components can be used as input or output of the planetary gear-set. The ratios of the individual states of motion cannot be selected independently of each other, but are defined by the numbers of teeth on the sun gear and the ring gear (Table 6.7).

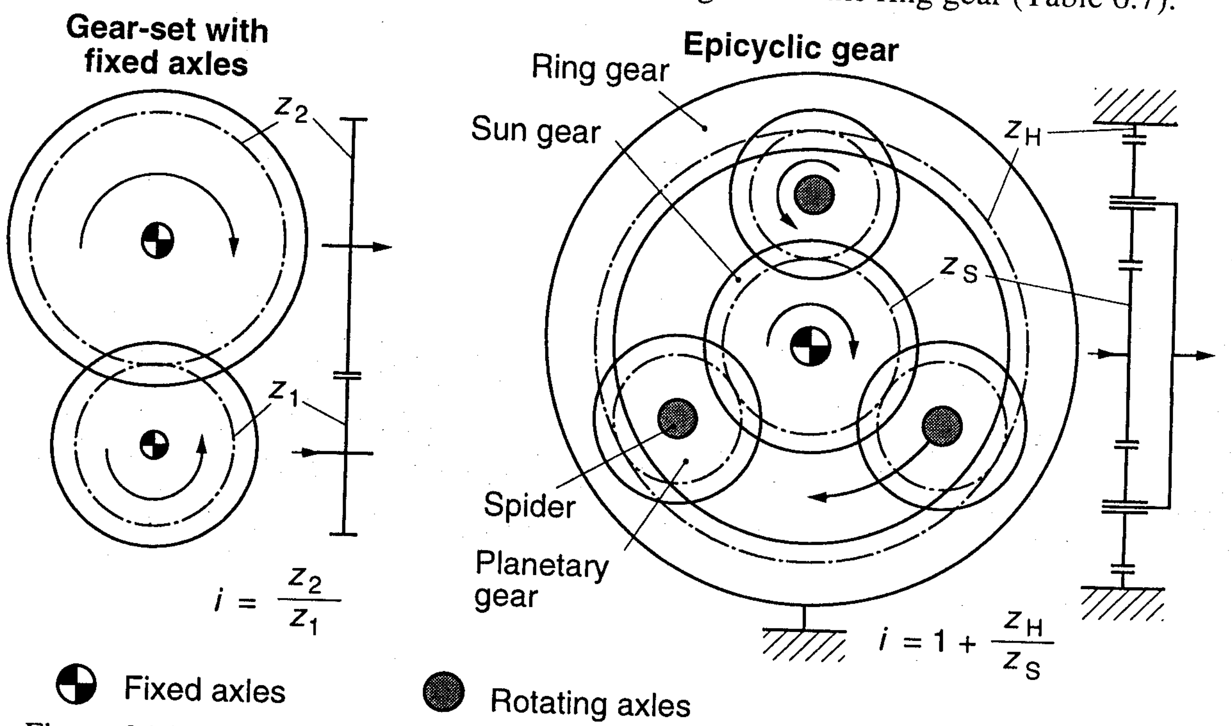
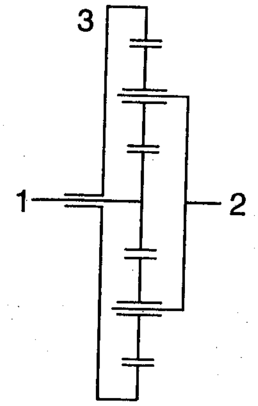


Figure 6.15. Gear-set with fixed axles and epicyclic gear

Table 6.7. States of motion and ratios of a simple planetary gear-set [6.9]. (The number of teeth on internal geared wheels is to be entered as a positive value in the formula)



1 ... Sun gear

2 ... Spider

3 ... Ring gear

State of motion	Type of gearbox	Input	Output	Frame	Planetary step ratio
a)	Gear-set with fixed axles	1	3	2	$i_P = \frac{n_1}{n_3} = i_S = -\frac{z_3}{z_1}$
b)		3	1		$i_P = \frac{n_3}{n_1} = \frac{1}{i_S} = -\frac{z_1}{z_3}$
c)	Planetary transmission single drive	1	2	3	$i_P = \frac{n_1}{n_2} = 1 - i_S = 1 + \frac{z_3}{z_1}$
d)		2	1		$i_P = \frac{n_2}{n_1} = \frac{1}{1 - i_S} = \frac{1}{1 + \frac{z_3}{z_1}}$
e)		2	3	1	$i_P = \frac{n_2}{n_3} = \frac{1}{1 - \frac{1}{i_S}} = \frac{1}{1 + \frac{z_1}{z_3}}$
f)		3	2		$i_P = \frac{n_3}{n_2} = 1 - \frac{1}{i_S} = 1 + \frac{z_1}{z_3}$

This table does not list the three trivial states of motion in which the transmission rotates as a block. Furthermore, not all transmission ratios are suitable for use in motor vehicles.

If none of the parts in a planetary gear-set is in a fixed position, then it is referred to as a differential drive, or a summarising gearbox, transfer gearbox or differential gear. If several planetary gear-sets are linked together, the result is a so-called coupled gear. This sort of gear makes it possible to achieve different ratios between input and output, depending on how the individual transmission components are linked together and which components are in a fixed position. The components are linked together by clutches, and the components are linked to the housing by brakes. The great variety of possible ratios available in transmissions with just one planetary gear-set is further substantially increased in coupled gears, but not all the ratios that can be derived from the transmission are relevant in motor vehicles. There are other important designs in addition to the simple planetary transmissions discussed here. You may wish to refer at this point to Section 6.6 and the relevant literature [6.8] to [6.10].

Traditional automatic transmissions with various gear ratios are made up of several individual planetary gear-sets. The ratios of the individual gear steps cannot be freely selected independently of each other, since the same gearwheels are used for several gear steps.

A section from a Wilson transmission is shown in Figure 6.16 as an example of such a planetary transmission. The spider of the first planetary gear-set is connected to the ring gear of the second. The two gears are only shifted by the closing of the corresponding brake.

Planetary coupling gears can also be power split, as shown in the illustration (Figure 6.16) above in the case of second gear. *Reactive power* can also occur in calculating the power values in the various paths. Reactive power can be envisaged as power, flowing in a circuit, which is not detectable from outside. But it stresses the components through which it flows, and impairs the overall efficiency of the transmission. Planetary transmissions can reach very low overall levels of efficiency, which in extreme cases can even become negative. This represents transmission interlock, which in certain circumstances can be desirable if the transmission is not to be moveable from the output side.

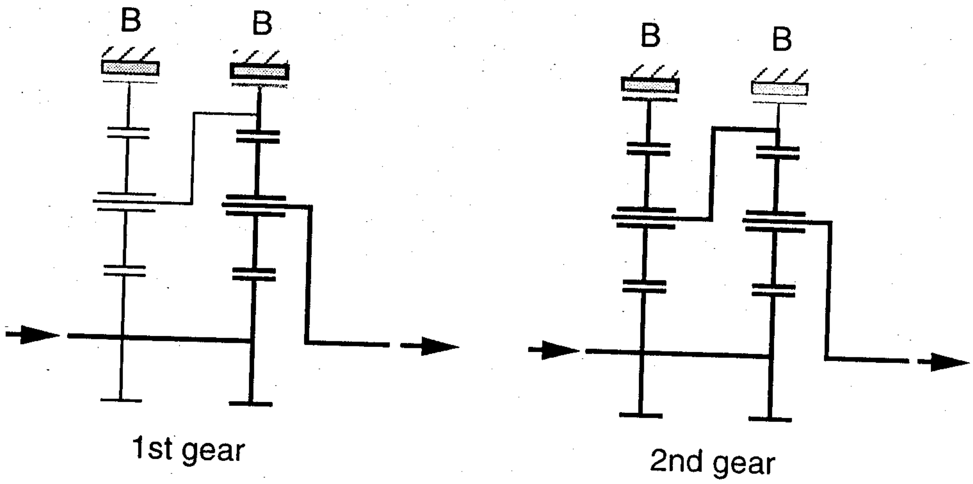


Figure 6.16. Planetary coupling gear: Section from a Wilson commercial vehicle gearbox (British Leyland; four forward gears: four planetary gear-sets, four belt brakes, one clutch)

6.5 Fundamental Approaches for Part Functions, Evaluation

In the concept phase of developing a transmission, basic approaches are established; see Figure 14.14 in Chapter 14 "Systematic Engineering Design". A large number of transmissions can be created by combining the individual approaches used for the main functions, as shown in the morphological table in Section 6.2 (Table 6.6). The number of viable alternatives is however significantly reduced when a *technical/economic evaluation* is carried out. This can be demonstrated using the example given in Table 6.8 for the main functions "Enable moving off" and "Change ratio". This is given as an example, and does not claim to be comprehensive.

A complete evaluation of all proposed solutions for the main and ancillary functions of the transmission should be carried out after the concept phase. The design phase proper can begin when this evaluation has been completed.

Table 6.8. Example of assessing approaches to the sub-functions "Enable moving off" and "Change ratio". 0...not possible; 1...very unfavourable; 2...unfavourable; 3...moderate; 4...favourable; 5...very favourable

Function	Approaches					
	Gear-wheel	Pulley drive	Friction clutch	Fluid clutch	Torque converter	Hydrostatic gearbox
Convert torque	5	4	0	0	4	4
Vary slip	0	0	4	3	3	3
Efficiency	5	4	4	3	3	4
Service life	4	3	3	4	4	2
Reliability	2	3	2	4	4	3
Ease of use	4	2	3	3	3	2
Space demand	5	3	4	4	4	2
Price	5	2	4	2	2	1
Total points	30	21	24	23	27	21

Table 6.8 suggests that the gear pair commends itself as by far the most cost-effective element for torque conversion. The disadvantage that this eliminates all but geared transmissions becomes a secondary consideration. Friction clutches are still the best available compromise for moving off and for speed synchronisation. The torque converter also has many advantages.

6.5.1 Reverse Gear

There are numerous designs for implementing the ancillary function of reverse gear. Figure 6.17 shows six different variants.

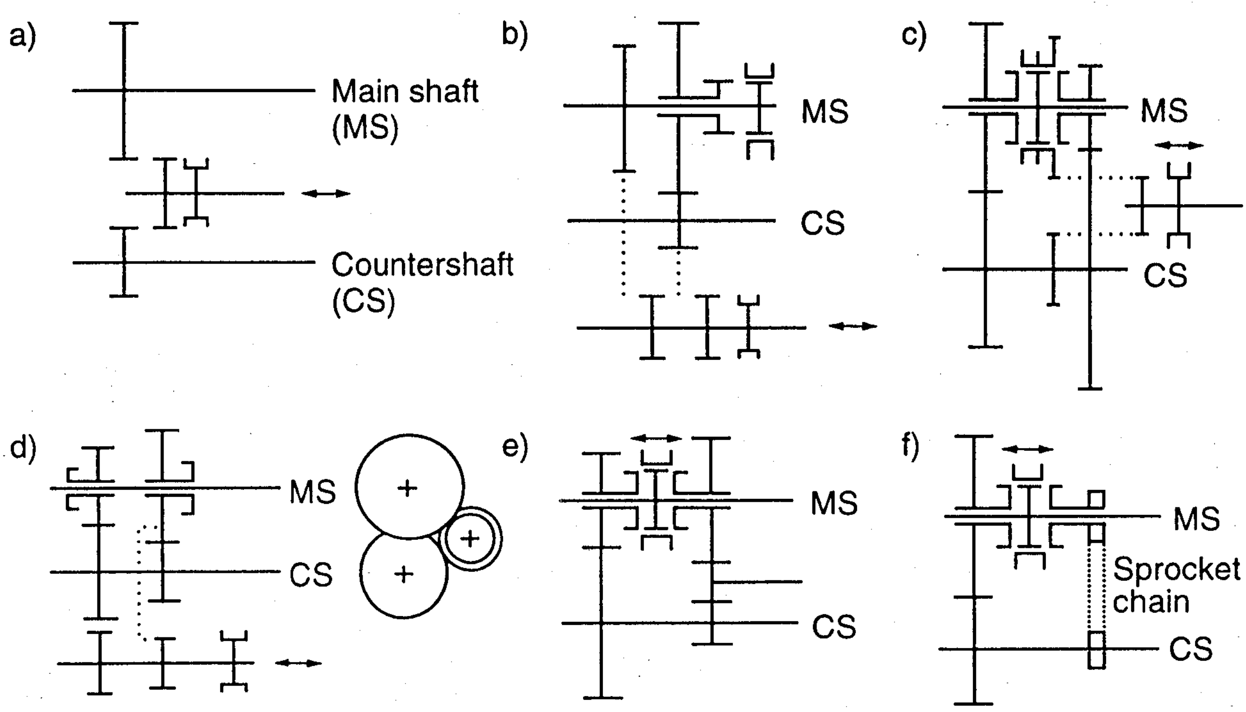


Figure 6.17. Alternative reverse gear configurations. **a)** An axial sliding gear is inserted between each fixed wheel of the main shaft and the countershaft ; **b)** Shiftable shaft with two pinions between a reverse gearwheel of the main shaft and a forward gearwheel of the countershaft; **c)** The sliding gear is inserted between a fixed wheel of the countershaft and a toothed sliding sleeve of a synchroniser on the main shaft; **d)** Sliding shaft with two pinions between a forward gearwheel of the main shaft and a forward gearwheel of the countershaft; **e)** Reverse gear with intermediate pinion constantly engaged, shifting with sliding sleeve; **f)** Reverse gear using gear chain, shifting with sliding sleeve

The required reversal of the direction of rotation of the gearbox output shaft is usually achieved by inserting an idler gear into the power flow. The general rule for toothed gearing is that increasing or reducing the number of ratio steps by one reverses the direction of rotation of the output shaft. Not all the variants shown in Figure 6.17 are of equal significance in practice. The following highly simplified assessment in Table 6.9 is intended to highlight their strengths and weaknesses.

If reverse gear does not use a gearwheel of a gear step of the forward gears, the cheaper spur gear toothing can be used for reverse, because of the relatively small proportion of time spent in reverse gear. The resultant increased noise level is acceptable.

Table 6.9. Advantages and disadvantages of various types of reverse gear (ref. Figure 6.17); + advantage, – disadvantage

Evaluation criterion / Solution	a)	b)	c)	d)	e)	f)
Easy to synchronise					+	+
Can be synchronised at rest	+	+	+			
Saves components compared to a)		+	+	+		
No ratio or toothing constraints	+				+	+
No axial space requirement			+			
Sufficient shaft clearance to accommodate the toothing			–			
Reverse gear must be helical cut		–	–	–		
Practicability				–		

6.6 Passenger Car Transmissions

Passenger car transmissions are classified into the following main designs and types:

- conventional 4–6 speed manual transmissions,
- semi-automatic transmissions,
- fully automatic transmissions:
  - conventional 3-speed to 5-speed automatic transmissions (consisting of torque converter and rear-mounted planetary gear),
  - 3-speed to 6-speed automatic countershaft transmissions,
- mechanical continuously variable transmissions.

The dominant design for passenger cars in the USA is the conventional automatic transmission. 75 to 80% of all vehicles are fitted with them. The same trend was predicted in Europe in the late '60s, but failed to materialise. Here the market share of automatic transmissions in passenger cars has remained virtually constant at around 15%.

Multi-range transmissions, as discussed in Section 6.7.2 for commercial vehicles, are in principle also feasible for passenger cars.

6.6.1 Manual Passenger Car Transmissions

Manual passenger car transmissions include all transmissions in which both the process of changing gear and the process of engaging the master clutch and moving off are carried out manually by the driver. They are all fitted with spur gears. Transmissions with dog clutch engagement are now practically unheard of in passenger cars. All transmissions are offered with synchromesh. Occasionally the reverse gear is still not fitted with synchromesh.

Passenger car manual transmissions can be sub-divided into further categories (see also Section 6.3). This subdivision relates mainly to the design of the main gearbox itself, not to any integral final drives, differentials and intermediate shafts needed to drive them. With this limitation, the following categories result:

- single-stage countershaft transmissions with 4 to 6 gears and integral final drive,
- two-stage, (coaxial) countershaft transmissions with 4 to 6 gears with and without integral final drive,
- three-stage countershaft transmissions with 4 to 6 gears and integral final drive.

Single-stage countershaft transmissions are used in passenger cars in which the engine is located on the drive side, which is to say in rear-wheel drive vehicles with rear engines, or in front-wheel drive vehicles with front engines. This applies to both the normal engine configurations – transverse and longitudinal; they also occur in "transaxle transmissions" (e.g. Porsche 944 with gearbox behind the axle) (Figure 6.2i). In the case of single-stage countershaft transmissions, the final drive is usually integrated into the gearbox housing.

In the transmission diagrams used in this chapter, integral final drives, where present, and reverse gears of the various transmissions are represented by "faint" lines for the sake of completeness. It should be noted that in reverse gears the shafts of the idler gears are located in a different plane to the main shafts (compare also with Section 6.5). The location and size of the idler gears are intended only to give an impression of the fundamental design.

Two-stage countershaft transmissions are used in passenger cars with standard drive. They normally contain no integral final drive components since they are generally flanged directly onto the front-mounted engine, and linked to the drive axle by a propeller shaft. One exception is two-stage transmissions mounted on the rear axle to give more even weight distribution with front-mounted engines (e.g. Porsche 928; transmission in front of the axle) (Figure 6.2h). Parts of the final drive are integrated in them.

In three-stage transmissions, a part of the transmission is relocated to a third countershaft located elsewhere. These transmissions are used in passenger cars where space constraints dictate a very short overall gearbox length.

The synchroniser packs are each allocated to one shift level, and serve mostly to shift two neighbouring gears. In each shift level there is usually a first and second gear, third and fourth gear, fifth and reverse gear, or alternatively fifth and sixth gear. There are also designs which use a separate clutch element for the fifth and reverse gear, which can be unsynchronised in reverse gear.

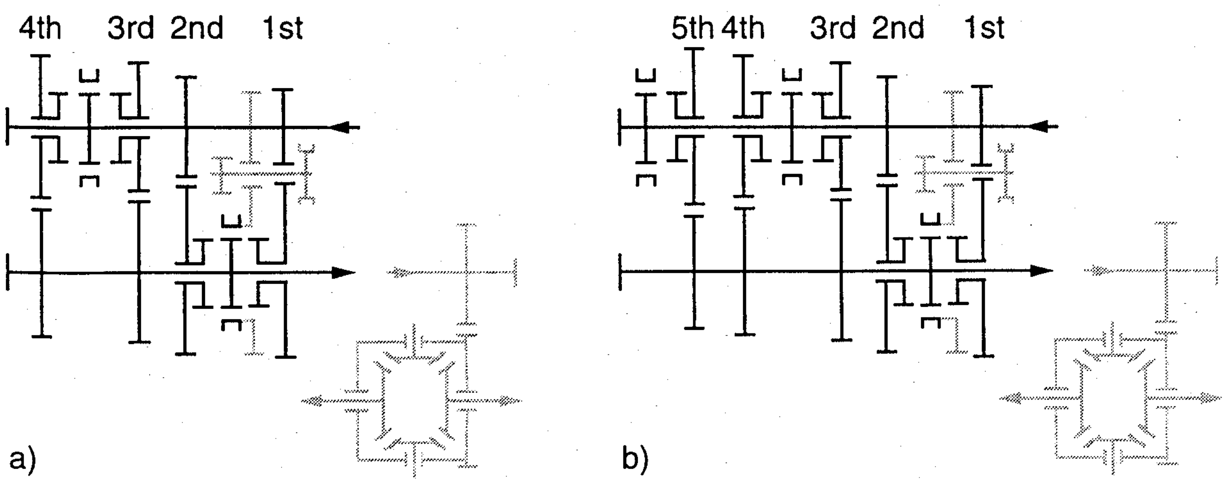


Figure 6.18. **a)** Single-stage 4-speed gearbox (VW); **b)** Single-stage 5-speed gearbox (VW), *production design Figure 12.5*

The example of a single-stage 4-speed gearbox is the VW unit, as used for example in the VW Golf (6.18a). In this gearbox, the gear pair of the first gear is located directly beside a shaft bearing. The total number of gear pairs remains the same compared to a two-stage 4-speed transmission, since although the gear pair of the *input constant gear CG,* (sometimes called *head set*), (Figure 6.19) is not required, one is needed for the fourth gear. Two-stage transmissions have a direct gear.

The single-stage 5-speed transmission (Figure 6.18b) differs from the single-stage 4-speed transmission only in having an additional gear stage, which is "attached on to" the drive housing side opposite the input side. This does not require any design changes in the original gear unit. Numerous 5-speed gearboxes have been developed from existing 4-speed gearboxes.

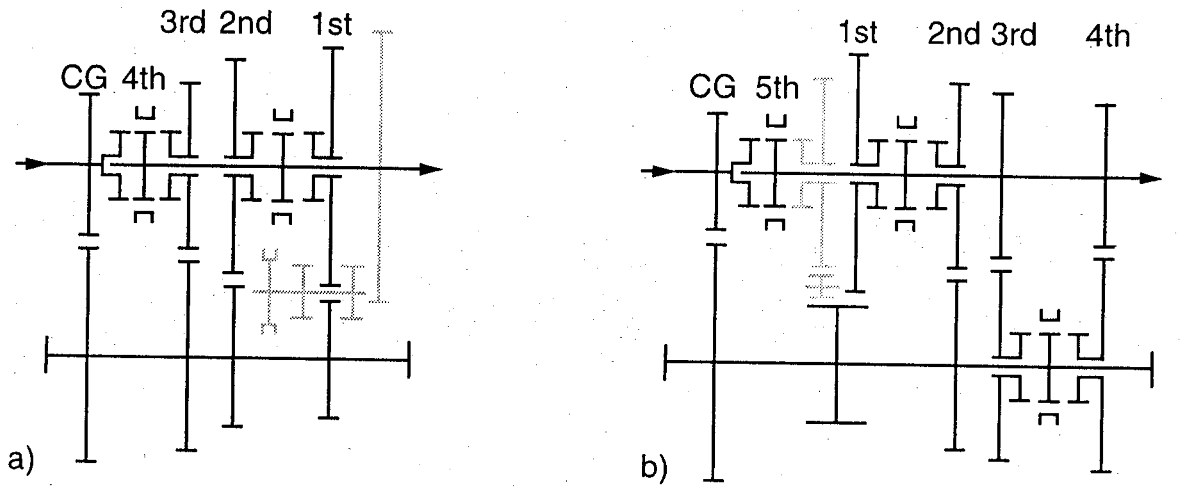


Figure 6.19. **a)** Two-stage 4-speed gearbox (Getrag); **b)** Two-stage 5-speed gearbox with direct 5th gear, "sports gearbox" (ZF), *production design Figure 12.1*

One example of the two-stage 4-speed gearbox is the Getrag gearbox in Figure 6.19a. In accordance with the design principle of placing gear pairs with high torque changes near bearings in order to minimise shaft deflection, the gear pair of the first gear is located on the gearbox output side. The fourth gear is the direct gear. In the 5-speed gearbox shown on the right in Figure 6.19, the fifth gear is the direct gear. Frequently the fifth gear is speed increasing (overdrive) and fourth gear is the direct gear.

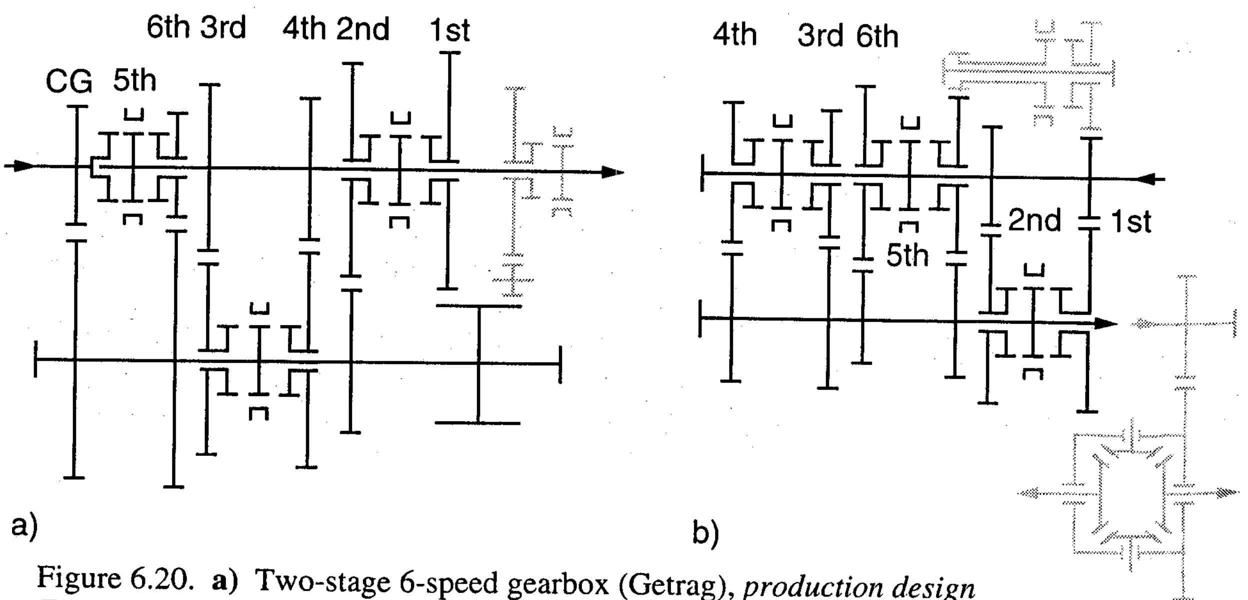


Figure 6.20. **a)** Two-stage 6-speed gearbox (Getrag), *production design Figure 12.4*; **b)** Single-stage 6-speed gearbox (Opel), *production design Figure 12.6*

There are manual gearboxes for passenger cars with up to six forward gears (Figure 6.20). In the two-stage countershaft transmission (Figure 6.20a), the gear step of the first and second gear are near a shaft bearing. It should also be borne in mind that such transmissions are used principally in high-performance passenger cars, and therefore have a high torque design.

Figure 6.20b shows a single-stage countershaft transmission with final drive. The reverse gear is located on a lay shaft to save space for the front transverse configuration. The transmission is three-stage for the reverse gear.

The advantage of the three-stage passenger car gearbox design is its short overall length (Figure 6.21). In fact this transmission is only three-stage in first and second gear, since in these gears the power flow runs through the countershaft. In third and fourth gears the power flow goes directly from the input to the output shaft, making the transmission functionally single-stage. The reverse gear acts as a two-stage gear unit since the power flow runs directly from the countershaft to the output shaft.

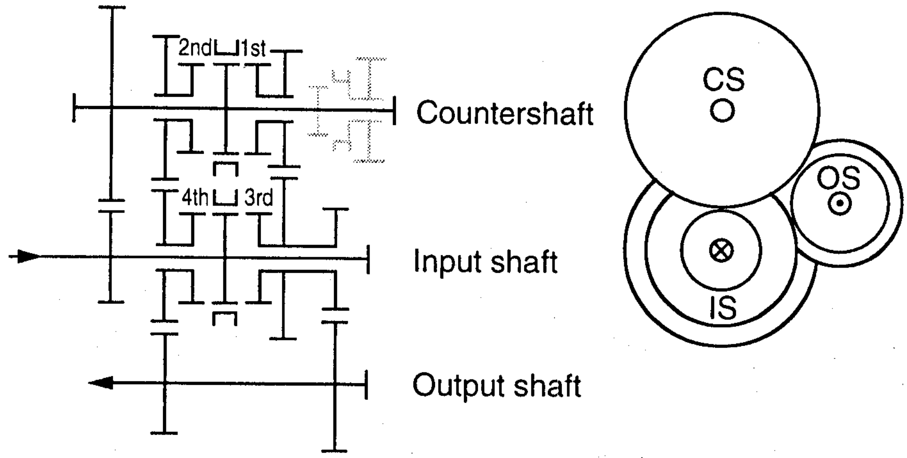


Figure 6.21. Gearbox diagram: three-stage 4-speed gearbox (Volvo)

6.6.2 Semi-Automatic Manual Passenger Car Transmissions

The term "Semi-automatic transmission" relates to the two operations "Engaging the clutch/Moving off" and "Changing gear". One of these operations is automatic in semi-automatic transmissions (see Table 6.12 "Levels of automation of passenger car and commercial vehicle manual gearboxes"). They may be broken down as follows:

- transmissions with automatic master/gearshift clutch, manual gear shift,
- transmissions with driver-controlled clutch, automatic gear-shifting process.

In the first variant the driver merely sets the desired gear by shifting the control lever, and drive take-up, moving off and changing gear is carried out automatically. In the second variant the driver preselects the gear or follows an automatic gear selection recommendation, activating the complete gear change with the clutch. In private passenger cars the first variant is principally used, whereas in commercial vehicles the second variant is common (see also Chapter 13 "Engine/Transmission Management"). Conventional manual gearboxes can be converted by retrofitting automatic controls for clutch operation or gear shifting.

In the VW torque converter clutch transmission (1967) there is a mechanical gearshifting clutch mounted behind a torque converter (Figure 6.22). When the gearshift lever is activated, the gearshifting clutch is automatically disengaged, interrupting the power flow to the manual gearbox. The gear can now be shifted manually. When the operation is complete, the gearshifting clutch engages again automatically.

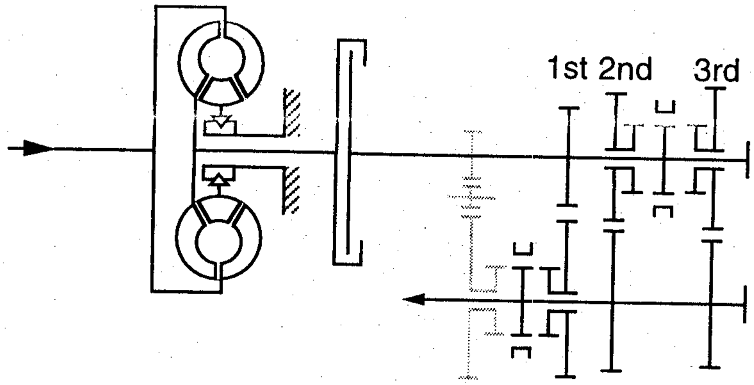


Figure 6.22. Gearbox diagram: 3-speed torque converter clutch gearbox (VW 1967)

The converter has three main functions to fulfil in this process:

- ❍ Enable moving off in any gear.
- ❍ Refine the coarse stepping (three forward gears) of the manual gearbox.
- ❍ Damp the torsional vibration when engaging the gearshifting clutch.

The main gearbox is a single-stage 3-speed gearbox developed from a 4-speed transmission by converting what was originally first gear into a reverse gear. In practice this transmission concept had to contend with high fuel consumption. The reason for this was the constant power flow through the converter – there was no lock-up clutch – and the fact that with this transmission it was possible to move off in second or third gear. This design therefore never became popular in passenger cars.

Transmissions with an automatic clutch are also to be found in Formula 1 racing cars. The driver activates a gearshift lever to manually control shifting up and down. In these transmissions only the gearshift clutch operation is automated, not the moving-off process.

Developments in passenger cars aim at automating the mechanical gearshifting clutch and master clutch (such as the Mannesmann Sachs EKS "Electronic clutch system" or the LuK EKM "Electronic clutch management"). The systems are now in mass production (production design, Figure 12.16a). In these systems the driver only changes gear manually – the clutch is controlled by an automatic system, both when changing gear and when moving off.

6.6.3 Fully Automatic Passenger Car Transmissions

The term "fully automatic transmission" is applied to geared transmissions in which the two part functions "moving off/engaging drive" and "changing gear" are carried out automatically in accordance with fixed or adaptive programmes.

Fully automatic transmissions have some advantages compared to conventional manual transmissions, such as

- ❍ reduced driver stress, and consequently improved road safety and ride comfort,
- ❍ faster shifting than the average driver,
- ❍ "smarter" shifting than the average driver, and therefore lower fuel consumption.

Gear shifting without any (noticeable) power interruption (powershift) is desirable in passenger cars to improve passenger comfort (see also Section 6.3.2 and Section 9.7).

Countershaft-Type Automatic Transmissions

Fully automatic passenger car gearboxes of one-, two- or three-stage countershaft design have the advantage of being very compact, allowing free choice of ratio, and comprising standard elements. Transmissions with high numbers of gears are simpler to build than with conventional automatic transmissions.

Well known examples are the Honda (Hondamatic, Figure 6.23a), the GM Saturn transmission and the W5A 180 transmission of Mercedes A-Class (production design, Figure 12.16b). In these transmissions a countershaft transmission is mounted after the converter. The conventional synchromesh units are replaced by multiplate clutch packages. The oil feed to the rotating multiplate clutches through the shafts is a soluble problem.

Automatic wet and dry clutches can be used as the moving-off element as well as the converter. Another example of powershift passenger car transmission and (depending on the degree of automation) fully automatic countershaft-type passenger car transmission is the *twin-clutch transmission* (Figure 6.23b). Twin-clutch transmissions were developed for commercial vehicles and passenger cars. The Porsche electronically controlled twin-clutch transmission (PDK = Porsche twin-clutch transmission) has been successfully used in racing passenger cars. The transmission input shaft is split into a solid shaft and a hollow shaft. There are two routes available for the power flow. In the example shown in Figure 6.23, the clutch C_1 is used for the second and fourth gears, the clutch C_2 for the first and third gears. The gears in the part that is currently not active can be preselected. Gear changing is then carried out by shifting from the one clutch to the other.

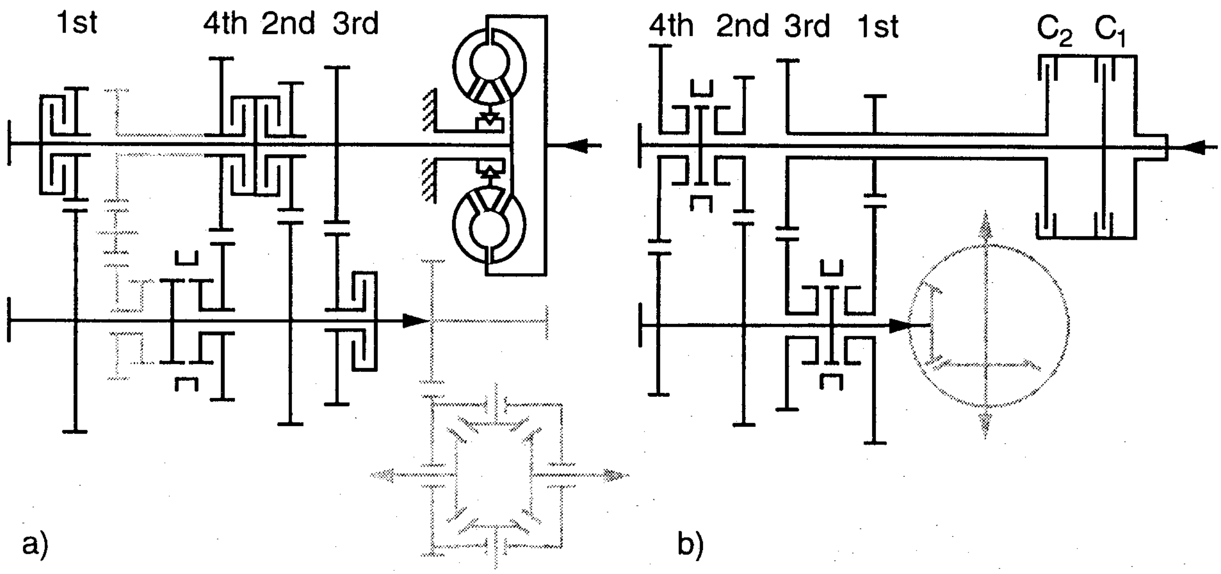


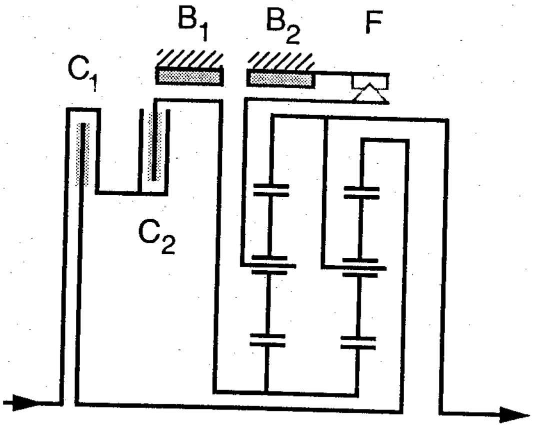
Figure 6.23. **a)** *Hondamatic* countershaft-type automatic gearbox (Honda);
b) Twin-clutch transmission for installation on the transaxle principle (Porsche)

Conventional Automatic Transmission

Fully automatic passenger car transmissions consisting of a torque converter with a planetary type gearbox enabling shifting without power interruption are known as conventional automatic transmissions or just "automatic transmissions". Fully automatic transmissions are now predominantly of conventional design.

Conventional automatic transmissions consist of the components listed above, with the power flow active in the particular gear step being defined within the planetary gear-sets by clutches and brakes (Figure 6.24).

The Simpson planetary gear-set has manufacturing advantages because it has the same number of gearwheels in both transmission parts. Both transmission parts run on a common wider sun gear. The design most commonly used in automatic transmissions is the Ravigneaux planetary gear-set, (Figure 6.25). This makes it possible to achieve up to four practically useable forward gears and one reverse gear.



Gear	1st	2nd	3rd	R
B_1		X		
B_2	(X)			X
C_1	X	X	X	
C_2			X	X
F	X			

Figure 6.24. Gearbox diagram of a Simpson planetary gear-set. *C* Clutch; *B* Brake; *F* Free-wheel; *X* Element engaged in power flow

The Ravigneaux set is a so-called reduced planetary gear. These are planetary transmissions in which the construction resources are "reduced" since parts of the individual simple planetary gears are the same size and can therefore be grouped together [6.8]. By using gear-sets of this type that can shift up to five gear steps, the overall dimensions of such planetary gears are relatively short. The selection of ratios is now restricted. Since the individual gearwheels are used for several gears, the resultant gear steps have to suffice.

Since in the case of automatic transmissions the converter carries out part of the change in transmission ratio, they theoretically require fewer gear steps than the manual transmission.

Some of the 5-speed automatic transmissions, which are becoming increasingly popular, use an additional planetary gear-set. Most of the space taken up by automatic transmissions is occupied by the multiplate clutch packs in the clutches and brakes required to shift the gears.

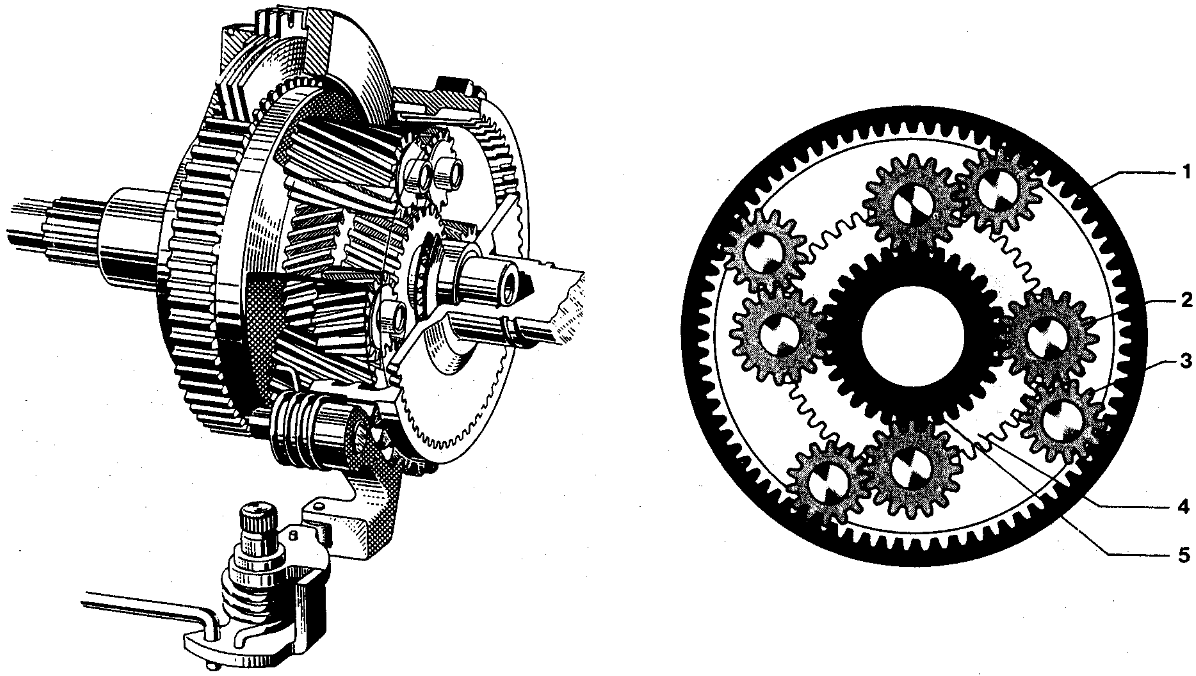


Figure 6.25. Ravigneaux planetary gear-set. *1* Common ring gear; *2* Narrow planetary gear; *3* Broad planetary gear; *4* Large sun gear; *5* Small sun gear [6.11]

There are two different types of brake as standard, the belt brake and the multi-disc brake. In the belt brake a metal belt runs once or twice around a brake drum, and brakes the drum by tightening the belt. This braking process is not easy to control, since the braking action is very rapid because of the self-reinforcing physical principle involved. In view of the increasing requirement for easy gear shifting, the multi-disc brake is becoming increasingly common. Although this takes more space than the belt brake, the shifting action is improved because of the finely controlled braking process. The multi-disc brake is based on the same components as the multi-disc clutch, which serves to link the moving parts of the transmission together. See also Section 9.1 "Shifting Elements".

The clutches and brakes discussed above for shifting the various gear steps are hydraulically controlled by hydraulic fluid. This fluid is supplied under pressure by a primary (engine driven) pump. The power it absorbs is no longer available to propel the vehicle, and thus represents an efficiency loss for the transmission as a whole. The effect of this pump on the overall level of efficiency is comparable to that of the torque converter. An overview of the losses in automatic transmissions is given by the highly simplified block diagram of a conventional automatic transmission in Figure 6.26.

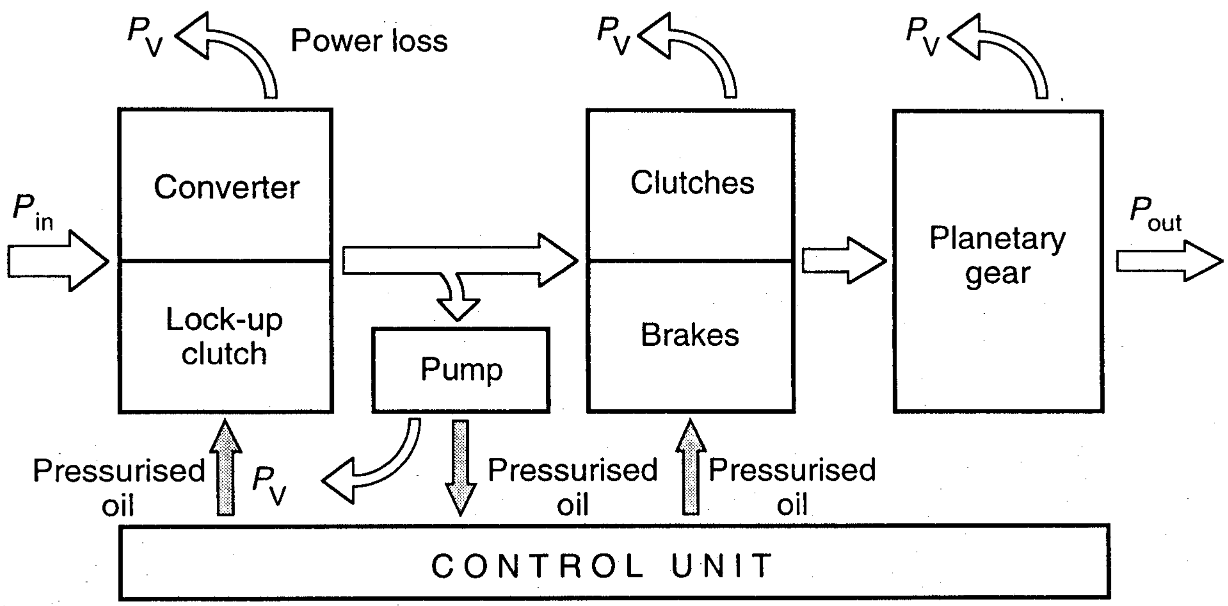


Figure 6.26. Block diagram and power losses in a (conventional) automatic transmission

The functioning of a 4-speed automatic transmission is considered in detail below based on the ZF 4 HP 14 automatic transmission (Figure 6.27). This transmission is designed for use in front-wheel drive passenger cars, which would become apparent in the transmission diagram only after the planetary gear. The components devoted to the final drive are not shown, since they have no effect on the principle of operation of the automatic mechanism. The components involved in the particular gear step are shown by heavier lines.

The ZF 4 HP 14 four-speed automatic transmission consists of a torque converter with integral torsion damper *T*. To improve efficiency, the transmission has no torque converter lock-up clutch, but works with *power split*, (see Section 10.7). Please refer to Chapter 10 for a precise functional description of torque converters. There is also a crescent design oil pump linked to the pump shaft of the converter (not shown in the diagram) to provide the pressurised oil necessary to shift the gears. The actual kinematic transmission is a 4-speed Ravigneaux set. The clutches are multiplate clutches shifted by oil pressure.

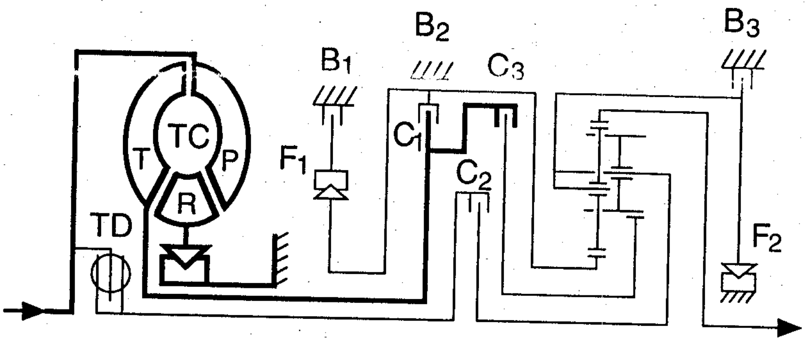


Figure 6.27. Gearbox diagram: 4-speed automatic transmission (ZF) in neutral;
TC Trilok converter:
P Pump, *T* Turbine,
R Reactor
TD Torsion damper
F Freewheels
B Brakes
C Clutches

The brakes are of both designs, multi-disc brakes B_1 and B_3 and belt brake B_2. The clutch linings and brake linings in automatic transmissions have an extremely long service life if correctly designed, since they run in oil, and are almost non-wearing.

In 1st gear, the spiders of both planetary gears are retained by the freewheel F_2, by means of which the planetary gear-set functions as a gear-set with fixed axles (Figure 6.28). The input power flows through the converter and the engaged clutch C_3 to the large sun gear of the Ravigneaux set, and back out of the planetary gear-set via the ring gear to the output. The effective ratio $i = 2.41$.

In second gear, the small sun gear rests against the gearbox housing by means of the free wheel F_1 and the brake B_1. The input power flows through the converter and the engaged clutch C_3 to the large sun gear, as in the first gear. The bar of the planet gear-set now rotates, and the planet gear-set functions as a reduced planetary coupled gear. The power flows again via the ring gear to the output, and the effective ratio is $i = 1.37$.

The third gear is the most interesting from the point of view of its functioning. The transmission functions with power split, i.e. a part of the drive power flows through the torsion damper *T* and the engaged clutch C_2 into the planetary gear-set which functions as a differential drive. The second power split flows from the converter through the clutch C_3 to the large sun gear of the planetary gear-set. Both power branches, or rotational speeds, "overlap" in the planetary gear-set, and are fed to the output at the ring gear. This operating status of the power split is not to be confused with that of a closed torque converter lock-up clutch *TCC*. In the torque converter lock-up clutch the impeller *P* and turbine wheel *T* of the converter are linked together, locking up the converter (Figure 6.29). The transmission ratio in third gear depends to a small degree on the slip in the converter, and is therefore not constant. The ratio in third gear thus varies between $i = 1.0$ and 1.09.

In 4th gear the converter runs without load, and power transmission to the planetary gear-set is purely mechanical through the torsion damper *TD* and the clutch C_2. The Ravigneaux set functions as a simple planetary gear driven through its spider, and whose sun gear is supported at the housing through the brake B_2. Output is through the ring gear. The ratio in 4th gear is $i = 0.74$, constituting an overdrive.

In reverse gear the Ravigneaux set again works as a simple planetary gear reversing the direction of rotation. The power flows through the converter and the clutch C_1 to the small sun gear. The spider is supported against the housing through the brake B_3. The output is through the ring gear. The reverse gear ratio is $i = -2.83$.

A further example is the 5 HP 18 five-speed automatic transmission for standard drive (Figure 6.29). In contrast to the 4 HP 14, this transmission has no power split. The converter can be removed from the power flow by means of a torque converter lock-up clutch *CC*. See also Figure 5.9 "Interaction of engine and converter". The fact that the torque converter lock-up clutch is never shown as engaged in the ZF 5 HP 18 does not indicate that this is never a function, but rather that it can be engaged optionally in each range.

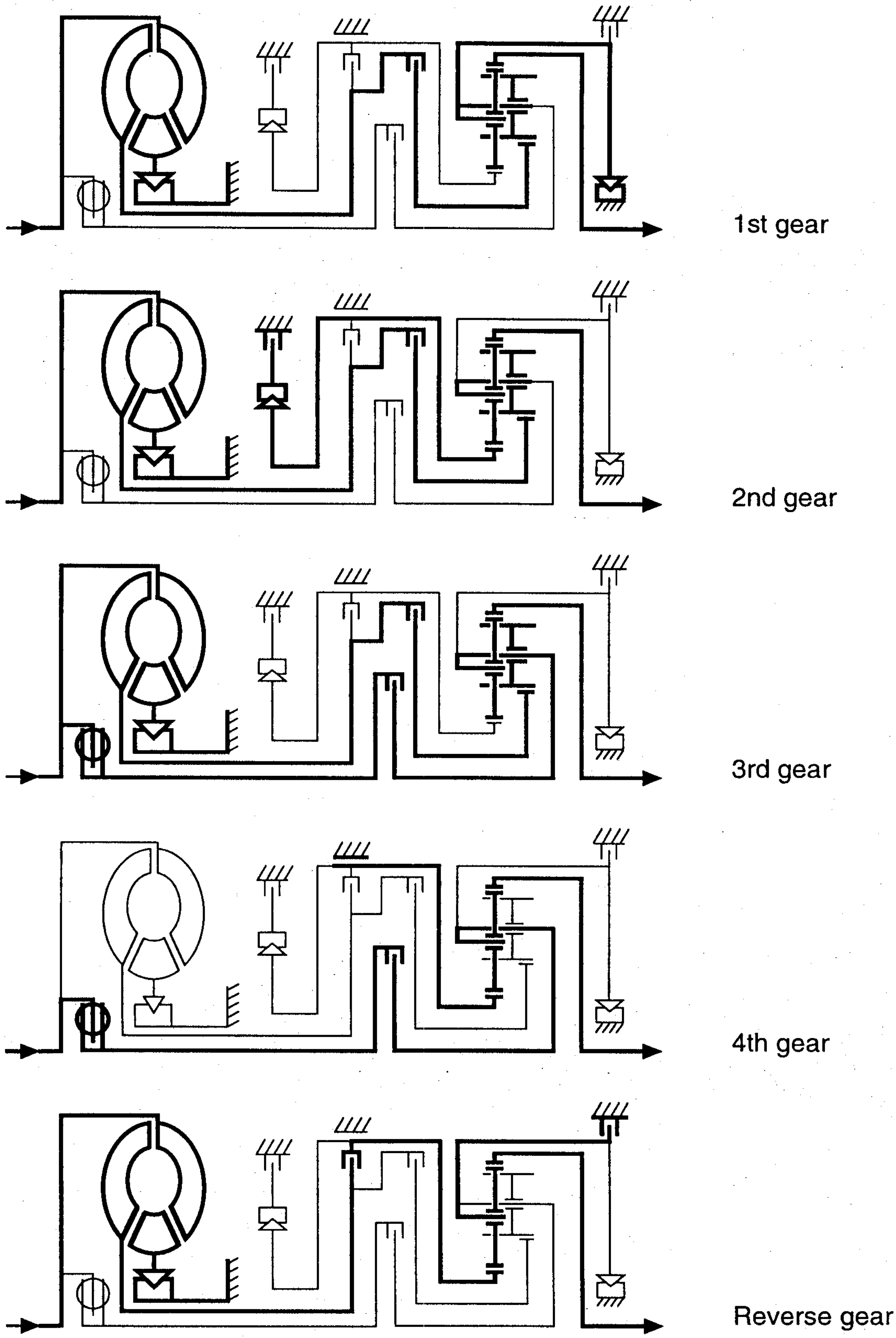


Figure 6.28. 4-speed automatic transmission (ZF); power flow in the gears

An important, if not the most important, assembly in an automatic transmission is the control unit. It is responsible for activating the brakes and clutches in the transmission. Their control has a direct influence on the "shifting quality" of the transmission as perceived by the driver (see also Section 13.3 "Transmission Control").

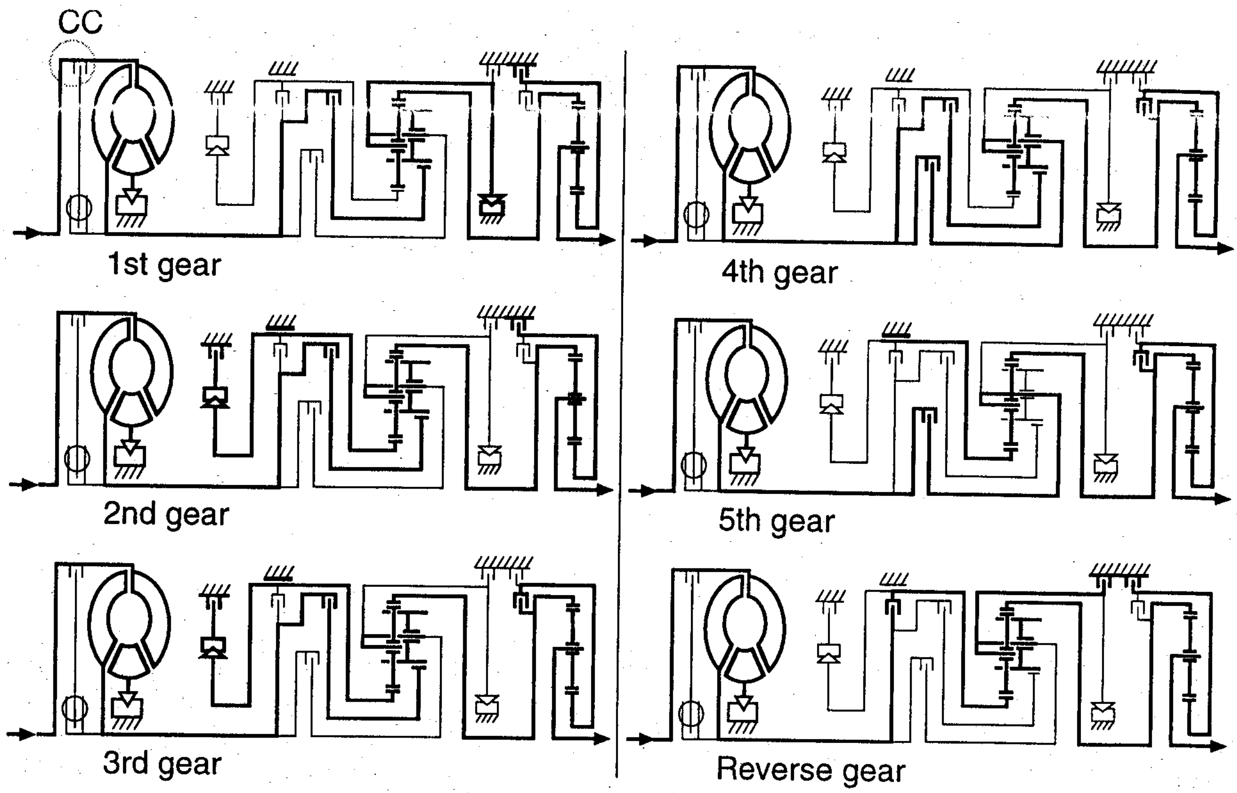


Figure 6.29. Gearbox diagram and power flow of a 5-speed automatic transmission (ZF) [6.12], *production design Figure 12.17*

In principle two types of control can be distinguished:

- ❍ hydraulic control units,
- ❍ electronic/hydraulic control units.

In hydraulic control units the input information to be processed is converted in a purely mechanical manner into proportional oil pressures which activate the shift elements through the hydrostatic servos (in principle always pistons under pressure). The entire control algorithm is embodied in the design of the hydraulic control unit. The assembly is very complex, so reference is made at this point to the relevant literature [6.13]. This was the design used in the first automatic transmissions. It has now reached a very advanced stage of development, and normally functions without fault throughout the life of the transmission. Two factors have led to development of electronic transmission controls: the disadvantages of hydraulic systems (such as the rigid control algorithm, embedded as it is in the hardware, and the fact that they cannot adapt to any mechanical wear), and the development of electronically managed engines. Electronic systems can adapt more easily to various engines or different operating conditions (adaptive gearshift programmes), and they can contribute to engine management (load reduction when shifting gear). They also have the general advantage of processing all available information, through to controlling the shifting process, taking into account the vehicle as a complete system. The shift elements are still activated hydraulically even with electronic control units.

The shifting profile of an automatic transmission is shown in simplified form in Figure 6.30. The shift points are principally dependent on the vehicle speed and the load on the engine. The position and shape of the shifting characteristic curve are also adapted to current driving conditions in modern controls by evaluating other parameters such as lateral and longitudinal acceleration, or rate of change of the accelerator position. The driver can influence the shifting characteristics, e.g. the selection of an economy or performance driving style.

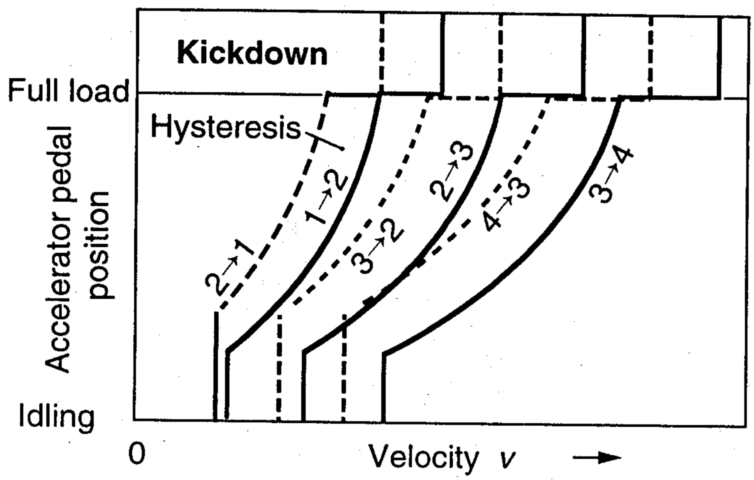


Figure 6.30. Qualitative profile of the shifting characteristic curves of an automatic transmission

The hysteresis arising from the two different shifting characteristic curves for shifting up and down between two gears is necessary to avoid constant shifting backwards and forwards at an operating point.

6.6.4 Continuously Variable Passenger Car Transmissions

The power available from an internal combustion engine cannot be fully exploited by the finite number of switching steps in traditional geared selector gearboxes. With a continuously variable transmission the engine can be operated at the ideal operating point for economy or performance as required (see also Sections 4.5 and 5.3.4).

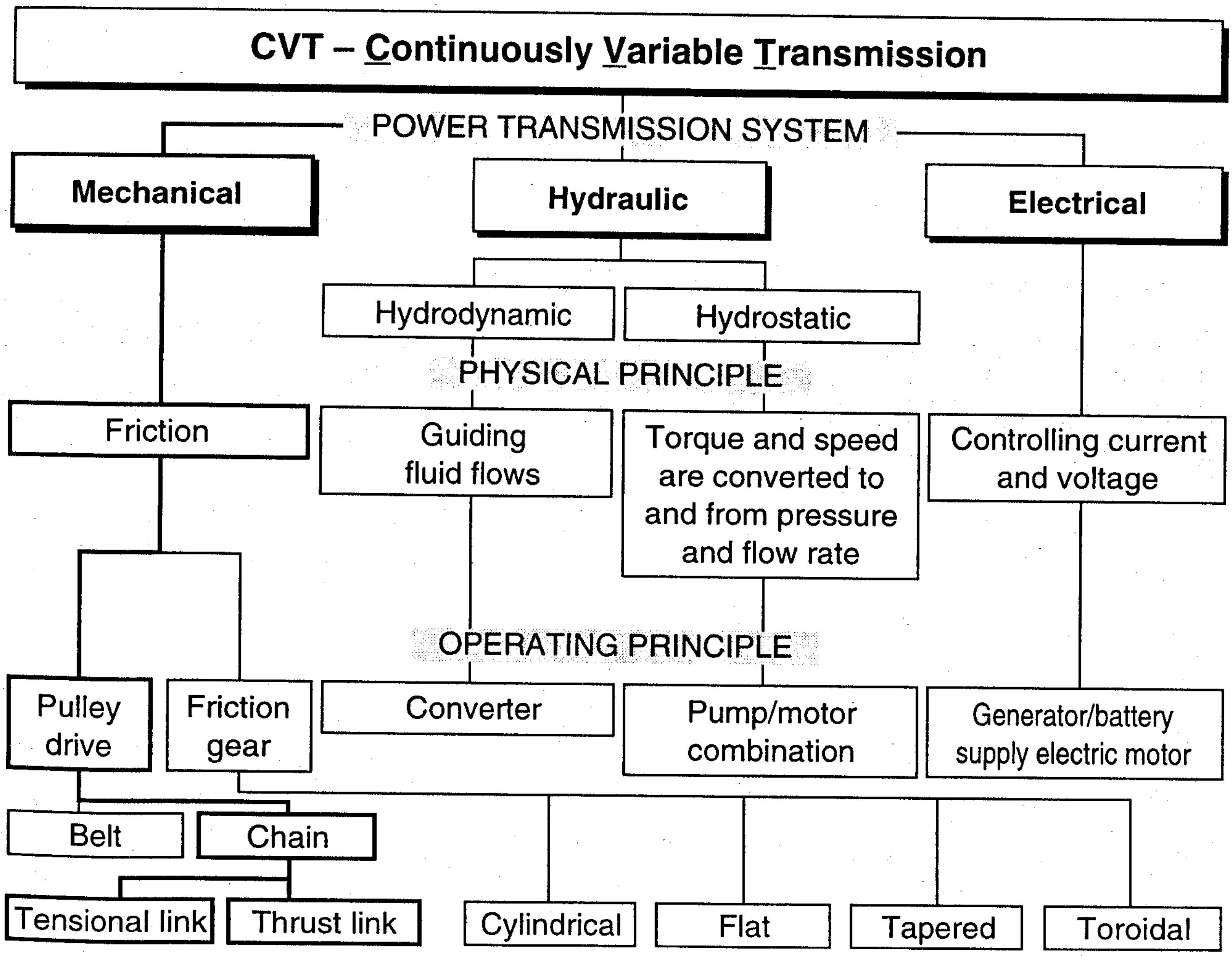


Figure 6.31. Overview of CVT designs

These transmissions are referred to as CVT (Continuously Variable Transmissions). Figure 6.31 gives an overview of various CVT designs. The continuously variable transmissions now used in passenger cars are almost without exception pulley transmissions. For the next generation CVT there is a growing interest in toroidal systems because of their higher torque capacity [6.27–6.29]. Figure 6.32a displays a simplified diagram of a toroidal variator in double configuration. The variation of ratio is achieved by swivelling the friction gears (rollers).

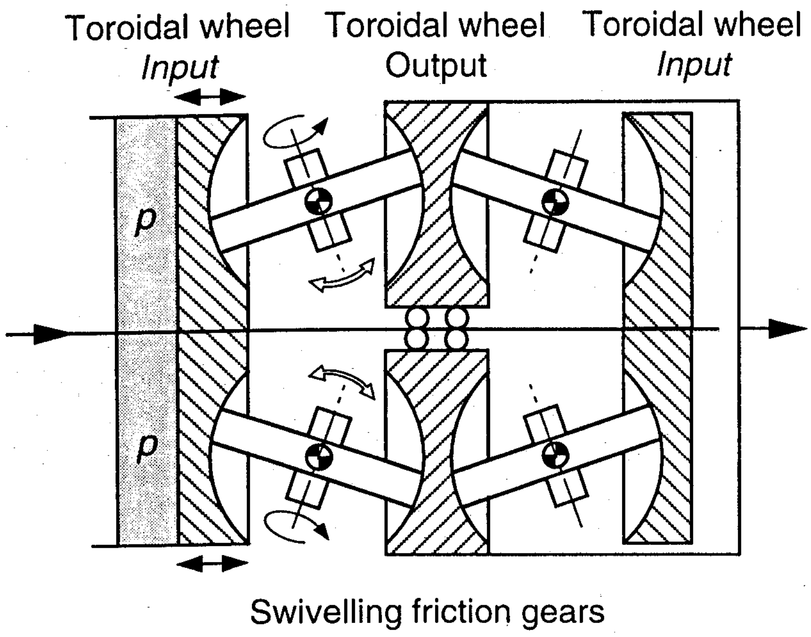


Figure 6.32a. Principle of toroidal variator

Hydrodynamic transmissions (converters) are only used with rear-mounted gearboxes, and are no longer proper continuously variable transmissions in this combination. There are also hydrostatic/mechanical powershift transmissions of the continuously variable type for use in passenger cars [6.14, 6.15]. They are of no practical significance at present. We therefore propose to investigate the continuously variable pulley transmission in greater detail in this chapter; this type of transmission could be of increasing importance in the near future if expectations as regards reduced consumption are fulfilled.

The central component of the chain converter transmission is the *variator*. It consists principally of taper discs and a chain. Power is transmitted frictionally through the chain, which runs between two axially adjustable taper discs. Through the axial adjustment of the taper discs, the chain runs on variable diameters, infinitely varying the ratio (Figure 6.32b). The power-related pressure of the taper discs on the chain requires a lot of attention, since excessive pressure reduces the efficiency of the chain, leading increased power consumption, and thus power loss by the contact pressure pump. It is essential to prevent the chain slipping, since this would inevitably lead to destruction of the transmission. This makes the construction and reliability of the contact pressure pump, and its control, a critical factor in these continuously variable transmissions.

With chains, a distinction is made between tensional link chains and thrust link chains. Tensional link chains are more efficient, since less power is required to adapt the chain to the ratio radii. The extremely short pitch of thrust link chains (Figure 6.32b) requires more (lost) work for this purpose. But the short pitch has many advantages at the "meshing impact" and in terms of associated noise generation.

In order to increase the overall gear ratio of the continuously variable transmission beyond the normal 5.3 to 6.0 of the variator, mechanical spur gears or planetary gears are front or rear mounted. Power splitting is also possible. (This is also valid for toroidal variators).

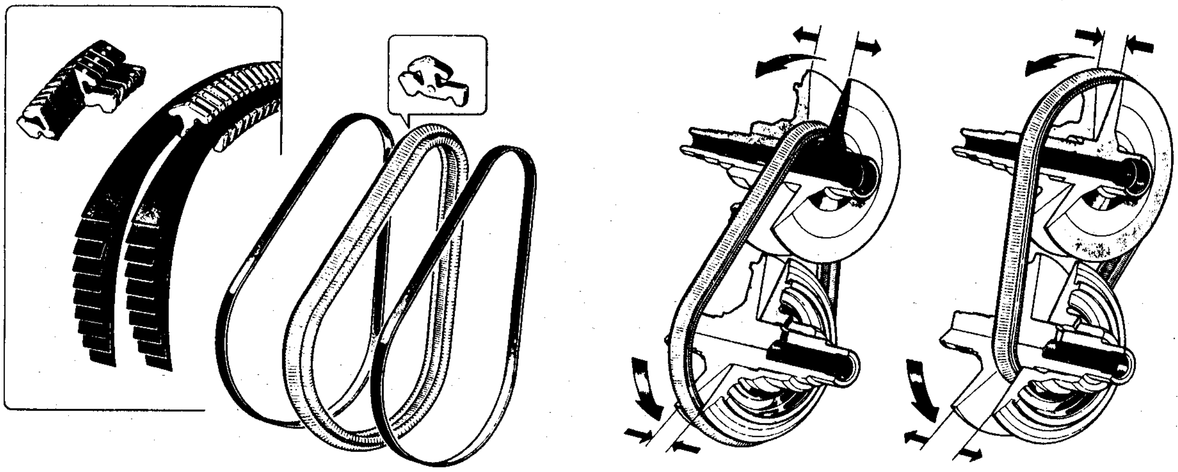


Figure 6.32b. Elements of a thrust link chain, and principle of operation of the variator [6.16], *production design Figure 12.13*

CVT with Rear-Mounted 2-Speed Twin Clutch Transmission, Figure 6.33

The ratio i_G of the rear-mounted gear step is equal to 1 when clutch C_1 is closed, since this corresponds to the direct drive. With the clutch C_2 engaged, the rear-mounted transmission gears down or makes the reverse gear available, depending on the position of clutch C_R.

The profile of the total ratio i as a function of the taper disc radius ratio i_V and the gear selected in the rear-mounted two-speed transmission is shown in the right-hand diagram in Figure 6.33.

The following formulae also apply to the CVT examples below. The overall total ratio

$$i = \frac{n_1}{n_2} = i_V \; i_G \; , \tag{6.4}$$

(the ratio of the input speed n_1 to the output speed n_2) of the transmission is made up of the ratio in the taper disc transmission (variator)

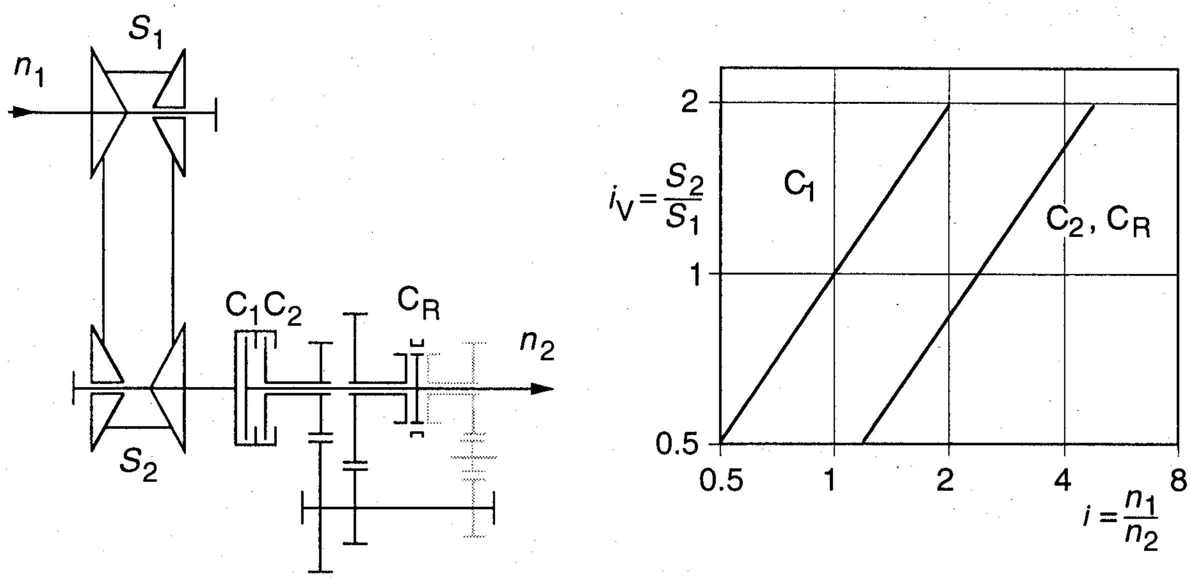


Figure 6.33. Gearbox diagram: "CVT with rear-mounted two-speed twin-clutch transmission" and diagram of the ratio profile

$$i_V = \frac{S_2}{S_1}, \tag{6.5}$$

(the ratio of the current taper disc radii S_2 to S_1) multiplied by a possible ratio i_G of a rear-mounted or front-mounted gear step.

CVT with Power Split (Geared Neutral Transmission), Figure 6.34

The total ratio i of this transmission, when operated with power split, is derived from i_V and the ratio i_G of the planetary gear-set functioning as a differential drive, where the clutch C_1 is closed. Depending on the ratio i_V, the total ratio i can also become negative, corresponding to reverse gear. The point where the sign changes is the geared neutral point. The transmission requires no additional moving-off element. With the clutch C_2 engaged, the planetary gear-set rotates as a block, and the simple equations given above apply. The profile of the total ratio as a function of the taper disc radius ratio i_V and the active clutch is shown in the diagram on the right of Figure 6.34.

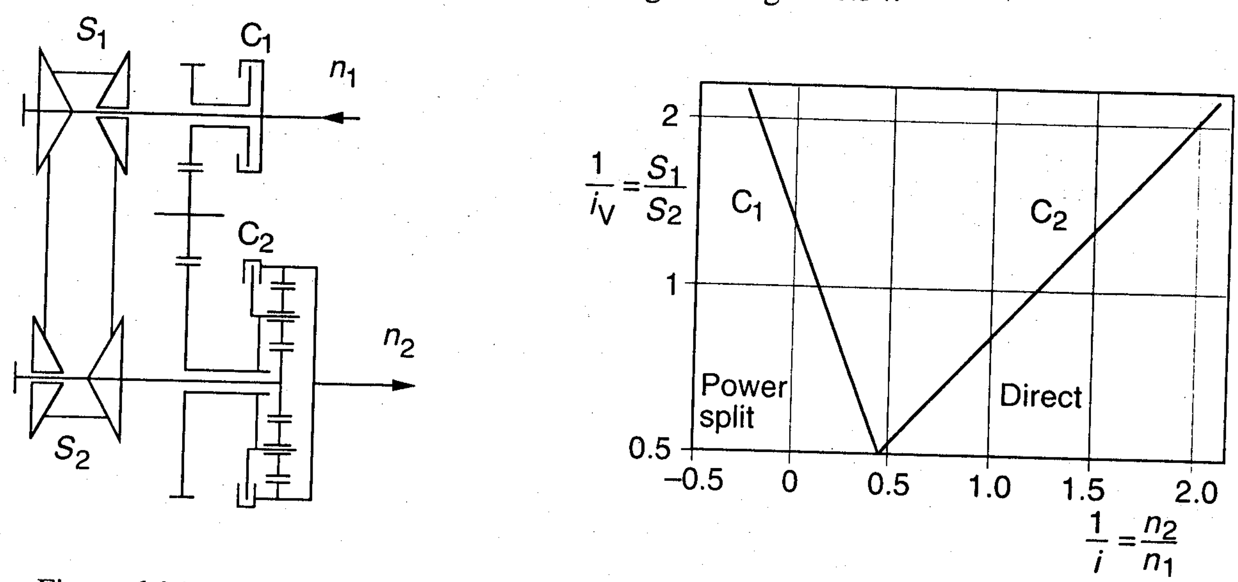


Figure 6.34. Gearbox diagram: "CVT with power split" and diagram of the ratio profile (shown as $1/i$ since $n_2 = 0$ at the geared neutral point)

CVT with Two Power Paths ("i^2-Transmission"), Figure 6.35

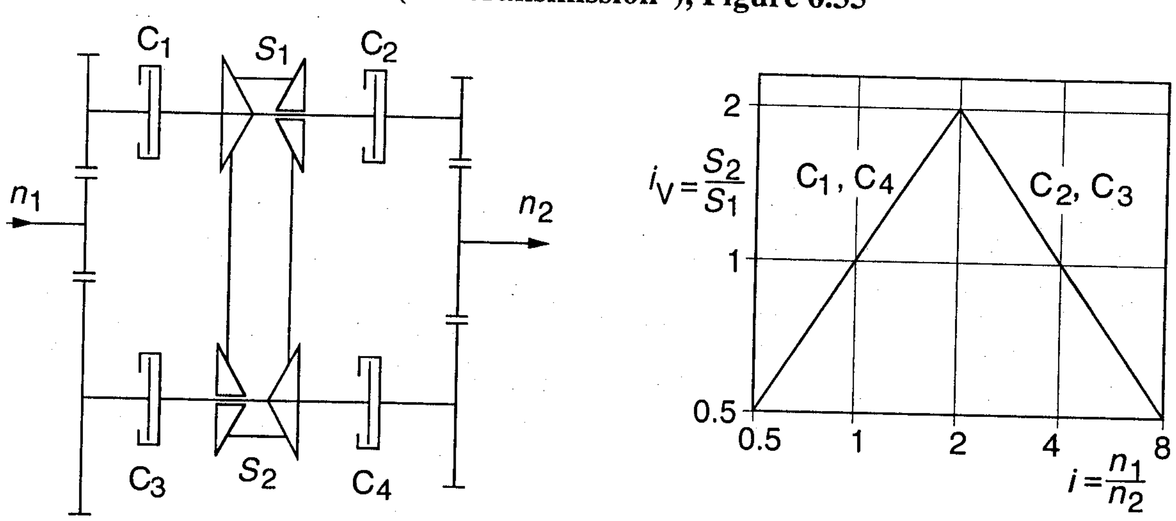


Figure 6.35. Gearbox diagram: "CVT with two power paths" and diagram of the ratio profile (also known as "i^2-transmission")

The total ratio i of this transmission is derived from i_V multiplied by the ratios of the two gear stages located in the power flow. Here either the two clutches C_1 and C_4 or the two clutches C_3 and C_2 are closed. The profile of the total ratio i as a function of the taper disc radius ratio i_V and the closed clutches is as shown in the diagram on the right of Figure 6.35.

6.7 Commercial Vehicle Transmissions

For transmissions of commercial vehicles up to 4.0 t gross weight rating, the explanations in the preceding Section 6.6 "Passenger Car Transmissions" apply. The following passage relates to

transmissions for commercial vehicles exceeding 4.0 t gross weight.

Table 6.10 lists common types of commercial vehicle transmissions. Depending on how their idler gears are positively locked to the shafts, manual transmissions can be subdivided into

- non-synchronised constant-mesh transmissions and
- synchronised transmissions

and by the shifting system (see also Section 9.1) into

- direct shifting: gearshift lever at the transmission housing,
- indirect shifting: gearshift lever and transmission physically separated (remote shift).

Table 6.10. Market shares and applications of commercial vehicle gearboxes

Gearbox type	Constant-mesh gearbox	Synchromesh gearbox	Torque converter clutch gearbox	Automatic + retarder
Number of gears	6 – 9 – 12 – 16	6 – 9 – 12 – 16	5 – 8	3 – 7
Market share, world-wide	Still large	Increasing	Very small	Small
Applications	Haulage; vehicles outside Europe	Local traffic; haulage in Europe	Heavy transporters; construction vehicles	Construction and delivery vehicles; city buses; heavy transporters

Non-synchronised constant-mesh gearboxes are common in long-distance trucks because of their robustness. Constant-mesh and synchromesh gearboxes have virtually equal market shares. (See also Section 2.2.1 "Market Situation" and Chapter 9 "Gear Shifting Mechanisms").

In heavy trucks, constant-mesh and synchromesh gearboxes are commonly semiautomatic and fully automatic. This is because the quality of gearshift action required is not as high as in passenger cars, and because the necessary auxiliary energy for the actuators is already available from the commercial vehicle's compressed-air system.

Torque converter clutch transmissions are hydromechanical transmissions with power interruption. They can also be fitted with an integral retarder if required. Their use is restricted to heavy special-purpose vehicles.

Conventional automatic transmissions are hardly used in trucks because of their higher price, their lower reliability due to the large number of parts, and the increase in fuel consumption of approximately 5%. But they are common in scheduled service buses and heavy special-purpose vehicles, where they significantly relieve driver stress.

6.7.1 Single-Range Transmissions

In the case of 4-speed to 6-speed gearboxes, the single-range design with input constant gear is standard (Figure 6.36). They are designed so that the ratio of a particular gear is derived from the individual ratios of two gear pairs. The first gear pair, the *input constant gear CG*, stays engaged in all gears with the exception of direct gear, and drives the countershaft at a constant ratio. When shifting into another gear, only the ratio of the second gear pair changes. Such a single-range transmission with input constant gear is referred to as a two-stage countershaft transmission, or simply countershaft transmission.

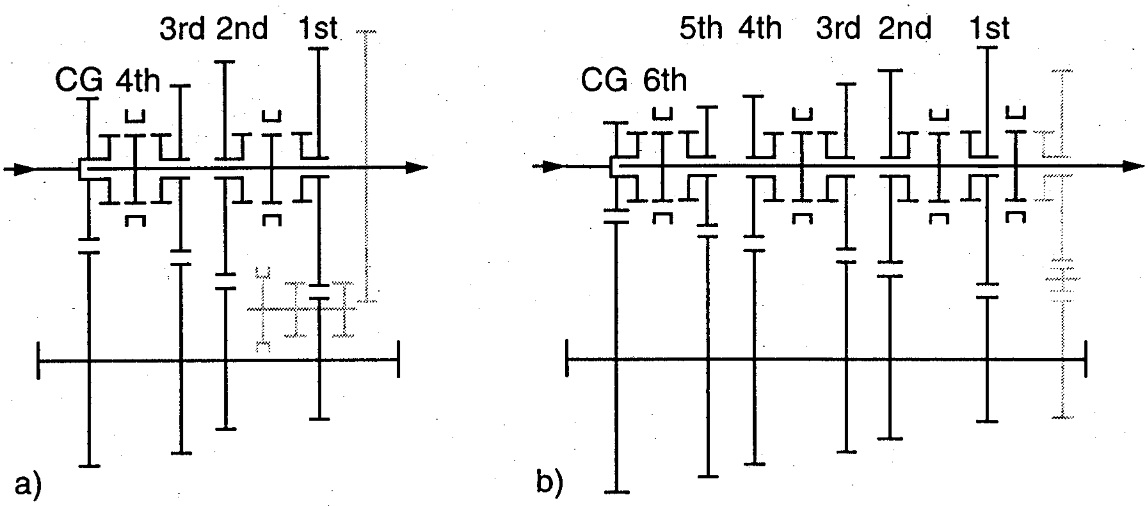


Figure 6.36. Single-range change design. **a)** 4-speed gearbox; **b)** 6-speed gearbox, *production design Figure 12.7*

There are numerous possible gearwheel configurations for any given number of gears. In general the emphasis is on locating high torque conversion in the vicinity of bearings, to minimise shaft deflection.

In commercial vehicles, these transmissions are called direct drive gearboxes or overdrive gearboxes (Figure 6.37), depending on whether in countershaft transmissions the top gear is a direct gear with a ratio equal to one or a ratio less than one (speed increasing ratio).

Alternatively such a transmission can also be fitted with a output constant gear CG_{out} (Figure 6.38). This means that the constant ratio is located behind the gear pairs for the individual gear steps.

Such a configuration has the following advantages:

- The reduced moment of inertia, which is crucial for synchromesh stress and is a function of the square of the gear step, is less with transmissions with an output constant gear.
- There is less deflection of the shafts on which the gear pairs for the switching steps are mounted.

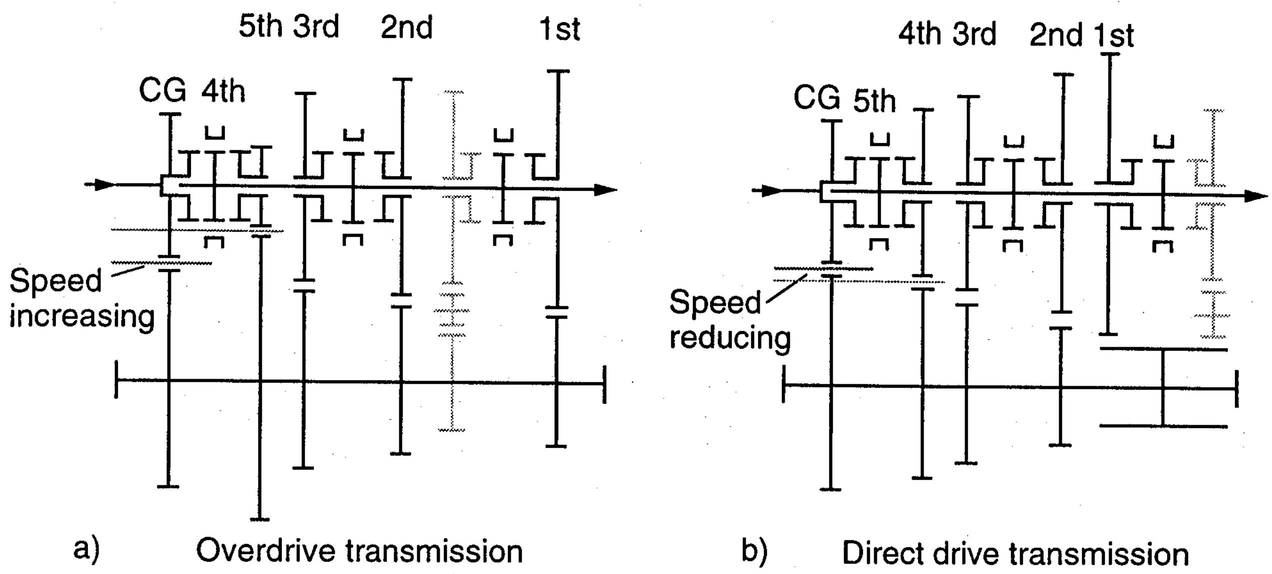


Figure 6.37. 5-speed countershaft transmission. **a)** Overdrive transmission – top gear is speed increasing; **b)** Direct drive transmission – top gear is direct drive. In the case of passenger cars this relates to overdrive (a) and sports transmissions (b)

These advantages have two countervailing disadvantages:

- Gearwheels, countershaft and bearings rotate at higher speeds than in gearboxes with an input constant gear.
- The output constant gear must be of more robust design, since there are already high torque levels. Like the input constant gear, it must be of durable design since it is always in the power flow, with the exception of direct drive.

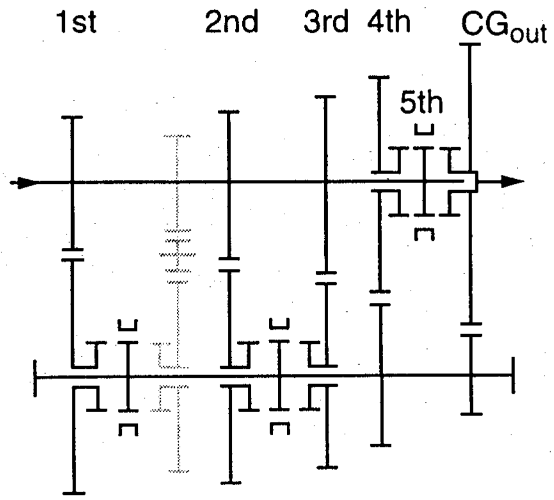


Figure 6.38. 5-speed gearbox with output constant gear CG_{out}

6.7.2 Multi-Range Transmissions

In the case of multi-range transmissions the task is to provide as many gear steps as possible with as few gear pairs as possible. Multi-range design is suitable for transmissions with more than 6 gears, and can be coaxial or non-coaxial.

Multi-range transmissions are constructed by combining single-stage, two-stage or multi-stage single transmissions (Figure 6.39), where a single transmission, which is by design self-contained, is called a *range*. The system boundaries are however fluid and cannot always be precisely defined. Both in a front-mounted splitter unit and in a front-mounted range-change unit, the second constant can also be used as a main gearbox gear pair (see the power flows in Figure 6.44). In the splitter unit a distinction is made between the "High" position (fast) and the "Low" position (slow). The splitter unit can be speed-reducing or speed-increasing. The range-change unit is always speed-reducing. The appropriate design must always be selected for each range unit. It is easy to arrange to link a countershaft transmission to a planetary gear transmission.

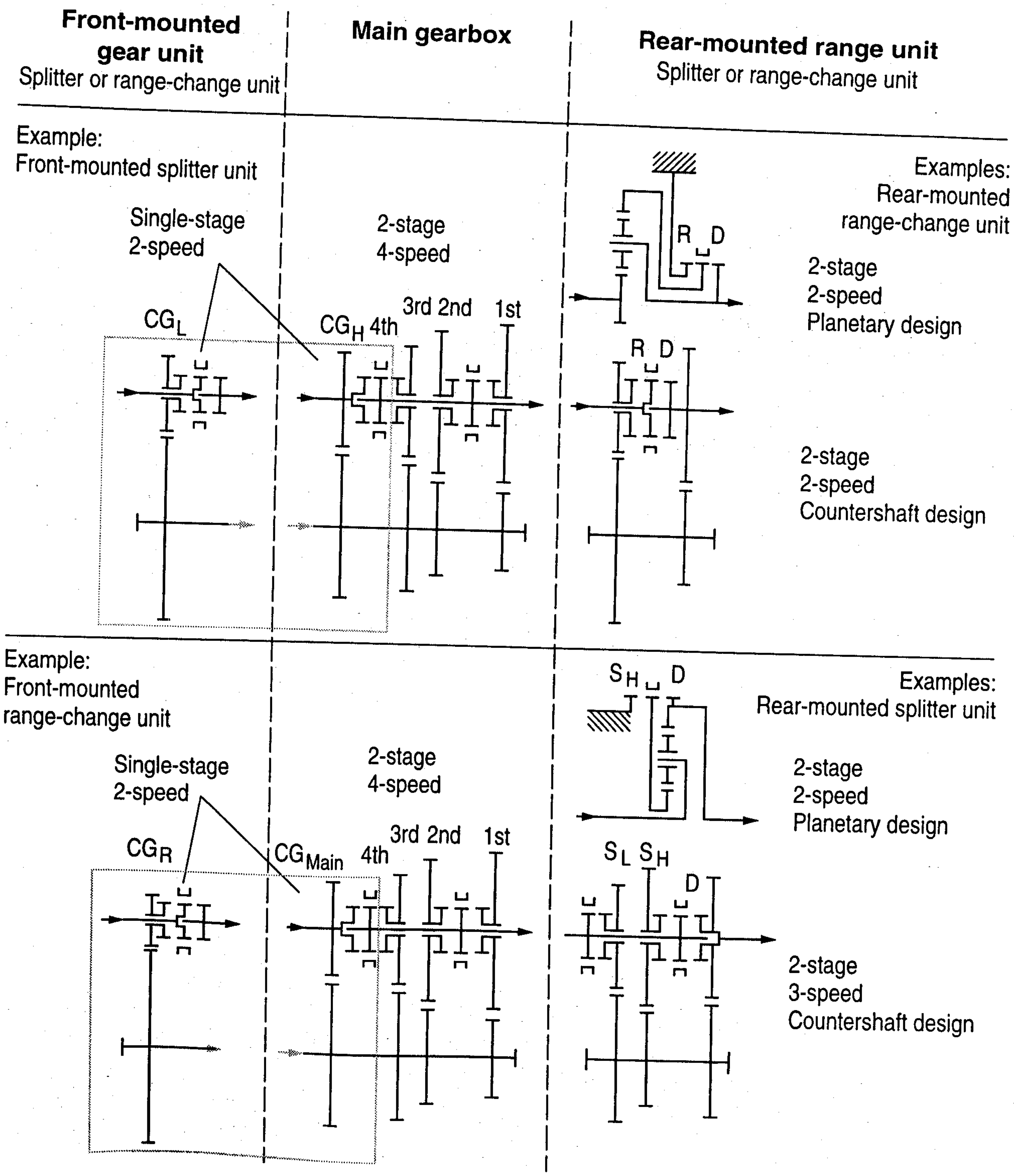


Figure 6.39. Combination of two-stage main gearbox with single-stage front-mounted or two-stage rear-mounted range transmissions. CG_H Front-mounted splitter unit constant high; CG_L Front-mounted splitter unit constant low; R Range; D Direct; CG_R Range constant; CG_{main} Main gearbox constant; S_H Rear-mounted splitter unit high; S_L Rear-mounted splitter unit low

A distinction is made between front-mounted and rear-mounted range units

- *splitter unit*
  compressing the gear sequence and
- *range-change unit*
  expanding the gear sequence.

Figure 6.40 shows the effect of the various ranges on the gear sequence, using the example of the three-range gearbox as shown in Figure 6.44 (see also the traction diagram of a commercial vehicle with front-mounted splitter unit in Figure 5.6).

The logarithm of the ratio is shown on the right of each graph in Figure 6.40 to illustrate the geometrical gear steps in multi-range transmissions.

Splitter Unit: Compressing the Gear Sequence

A splitter unit always leads to a compression of the gear sequence (6.40a). The splitter unit can be fitted before or after the main gearbox (Figure 6.39). The gear step in the splitter unit is less than that of the main gearbox (Figure 6.40a) (half as large with geometrical gear steps). The number of gears in the main gearbox is multiplied by the number of gear steps in the splitter unit. The splitter unit is normally fitted with two gears.

In practice, front-mounted splitter units are almost always used. The reason for this is that front-mounted splitter units only have a small ratio change of approximately 1.1 to 1.2. This means that the rear-mounted main gearbox is loaded either with only a slightly higher torque, or in the case of a speed-increasing ratio, even with a lower torque than without a splitter unit. If the splitter unit is rear-mounted to the main gearbox (bottom of Figure 6.39), it must be designed for the highest torque multiplication reached in the main gearbox. That is a more expensive solution than a front-mounted splitter unit.

Range-Change Unit: Expanding the Gear Sequence

The function of a range-change unit is to expand the gear sequence. This is achieved by the ratio step in the range-change unit being as big as the range of ratios in the main gearbox, multiplied with the gear step in the main gearbox (Figure 6.40b). The gear sequence with the range change unit engaged follows smoothly on from that of the main gearbox. Overlaps in the ratios of individual gears are avoided by using geometrical gear steps (Section 4.4.2). Range-change units are always speed-reducing. The torque multiplication in the range-change unit amounts to approximately $i_R = 3$–4. If the range-change unit was designed to be front-mounted, the high torque values would pass through the main gearbox. The range-change unit is therefore always fitted at the output end of the main gearbox. The range-change unit can be of countershaft design or a compact planetary gear unit.

Mixed Gear Sequence

Gear sequences can also be created with one or more range gearboxes, in which the gears do not all follow each other with the same gear step (geometrical stepping). Non-geometrical stepping can lead to part of the gears no longer being useable because the gear step to a neighbouring gear is too small.

Evaluation

Table 6.11 shows various combinations of range gearboxes. Splitter units and range-change units can be used singly or in combination as front-mounted or rear-mounted range units. Some of these possible combinations are however not viable in practice. A simplified evaluation of this multi-range change transmission was carried out to determine the most suitable design.

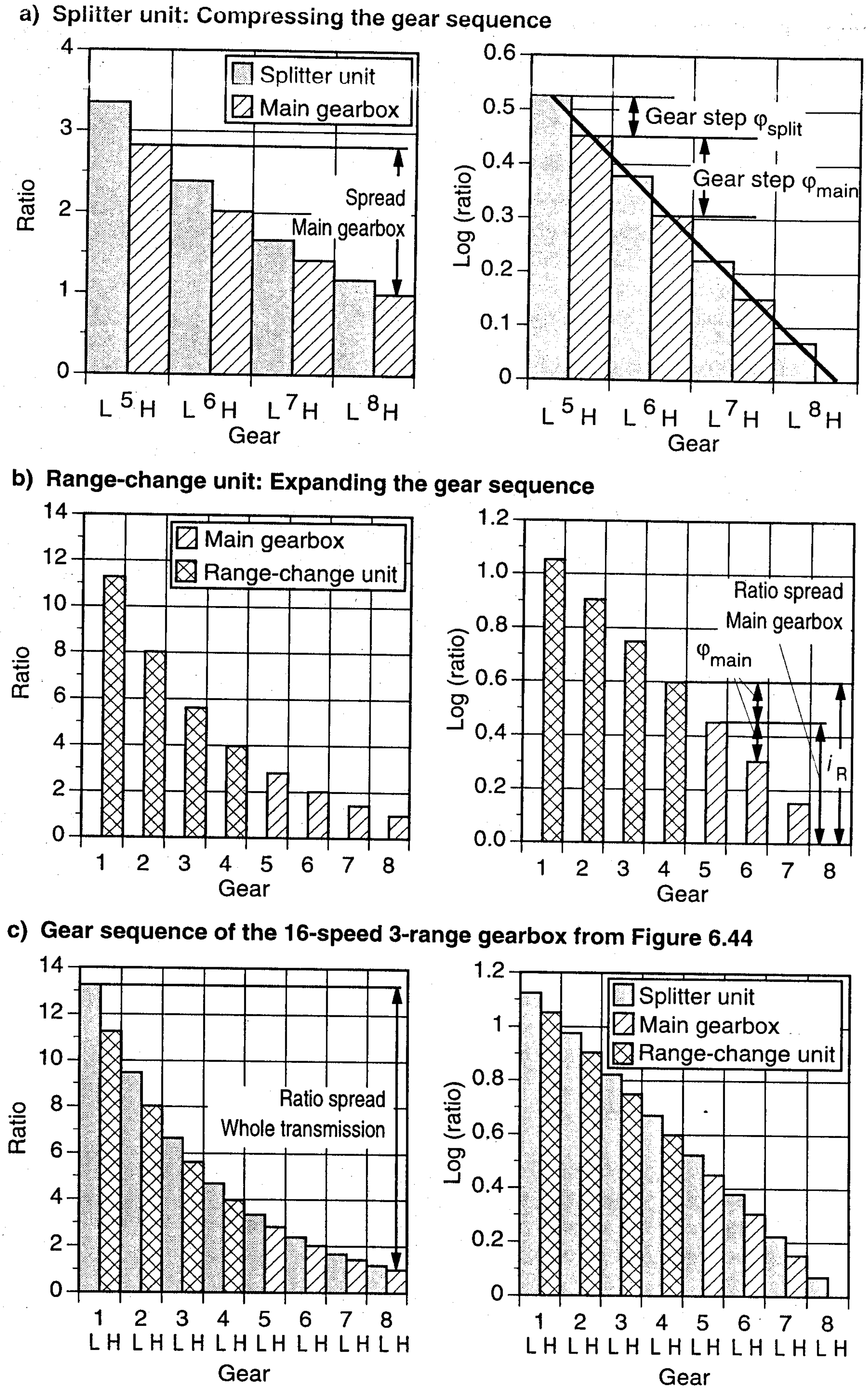


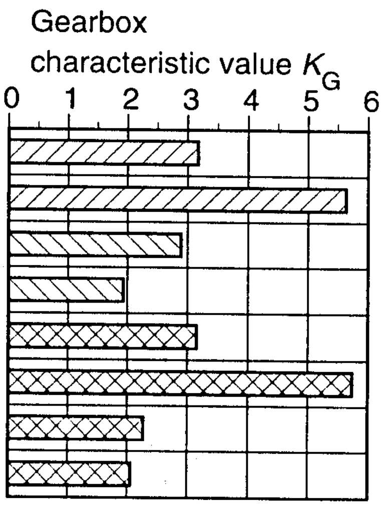
Figure 6.40. Compressing and expanding the gear sequence with splitter unit and range-change unit. Based on the example of the 16-speed commercial vehicle gearbox shown in Figure 6.44. L = Low (slow); H = High (fast)

This was based on a 4-speed main gearbox, with the simplifying assumption that the gear centres in each group are the same, and that for the torque multiplication in the splitter unit $i_{\mathrm{split}} = 1.2$, in the range-change unit $i_{\mathrm{R}} = 3.5$, and in the main gearbox $i_{\mathrm{G,H}} = 4$.

The characteristic value K_{G} is determined by the number of gear pairs per range unit, an incremental torque conversion factor, and the total ratio achievable. The resultant values represent a relative measure of the physical dimensions of the transmission, and are not absolute indications of size. They serve rather to compare the transmissions with each other. The smaller the transmission characteristic value K_{G}, the smaller the dimensions of the transmission. Front-mounted and rear-mounted range units, splitter units and range units were tried in different configurations. The results are shown in Table 6.11. The various possible configuration variants can be assessed in terms of their dimensions using this table.

Table 6.11. Various combinations of range transmissions and their characteristic values K_{G}. *F* Front-mounted; *R* Rear-mounted; *S* Splitter unit; *R* Range-change unit; *Main* Main gearbox

No.	Combination FS	FR	Main	RS	RR	Gear characteristic value K_{G}
S1	x		x			3.17
S2			x	x		5.63
R1		x	x			2.89
R2			x		x	1.93
RS1	x	x	x			3.15
RS2		x	x	x		5.74
RS3			x	x	x	2.27
RS4	x		x		x	2.05



The combinations reviewed can be subdivided into the following three categories:

- main gearbox with splitter unit (S1 and S2),
- main gearbox with range-change unit (R1 and R2),
- main gearbox with splitter unit and range-change unit (RS1 to RS4).

This reveals the combinations front-mounted splitter unit (S1), rear-mounted range-change unit (R2) and front-mounted splitter unit with rear-mounted range-change unit (RS4) as the most favourable in their respective categories in terms of physical dimensions.

Figure 6.41 shows common configurations of two- and three-range gearboxes. A high degree of flexibility can be achieved using the modular principle of two or three individual transmissions flanged together. Multi-range transmissions are in principle possible in passenger cars as well.

In multi-range transmissions shifting times become extended, since several junctions have to be shifted in some gears. The overall shifting time should also be less than a second under unfavourable conditions. Multi-gear transmissions can be constructed with a small number of gear pairs if several junctions are shifted simultaneously when changing gear. Theoretically,

$$z = 2^{(p-1)} \tag{6.6}$$

gears can be produced with p gear pairs.

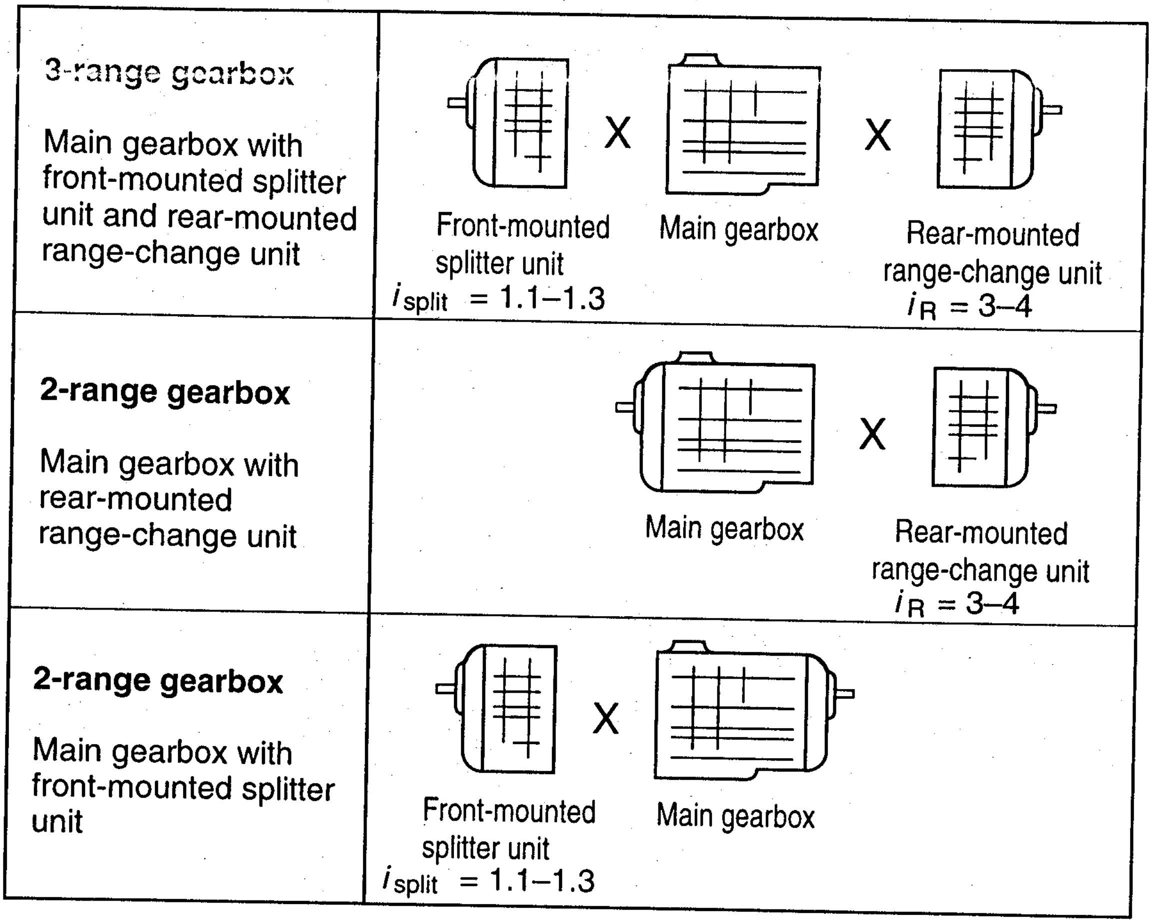


Figure 6.41. Conventional configurations with two-range and three-range gearbox

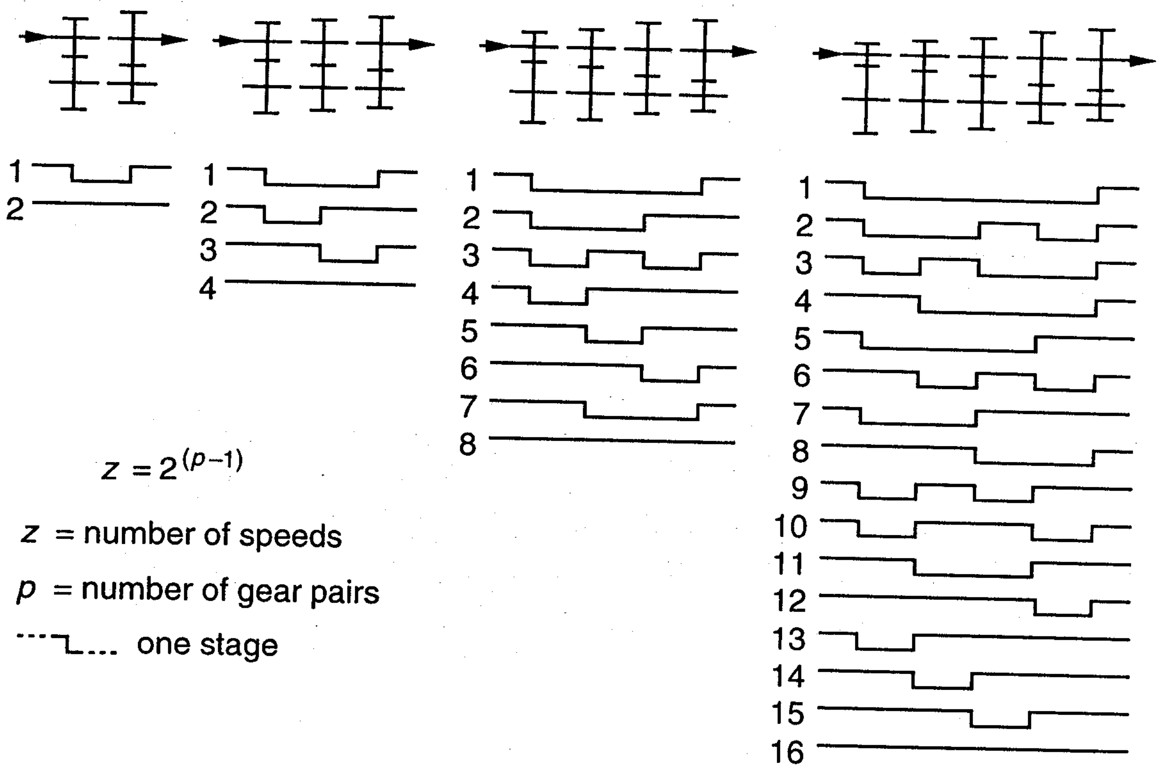


Figure 6.42. Gearbox diagrams and power flows of coaxial multi-stage transmissions

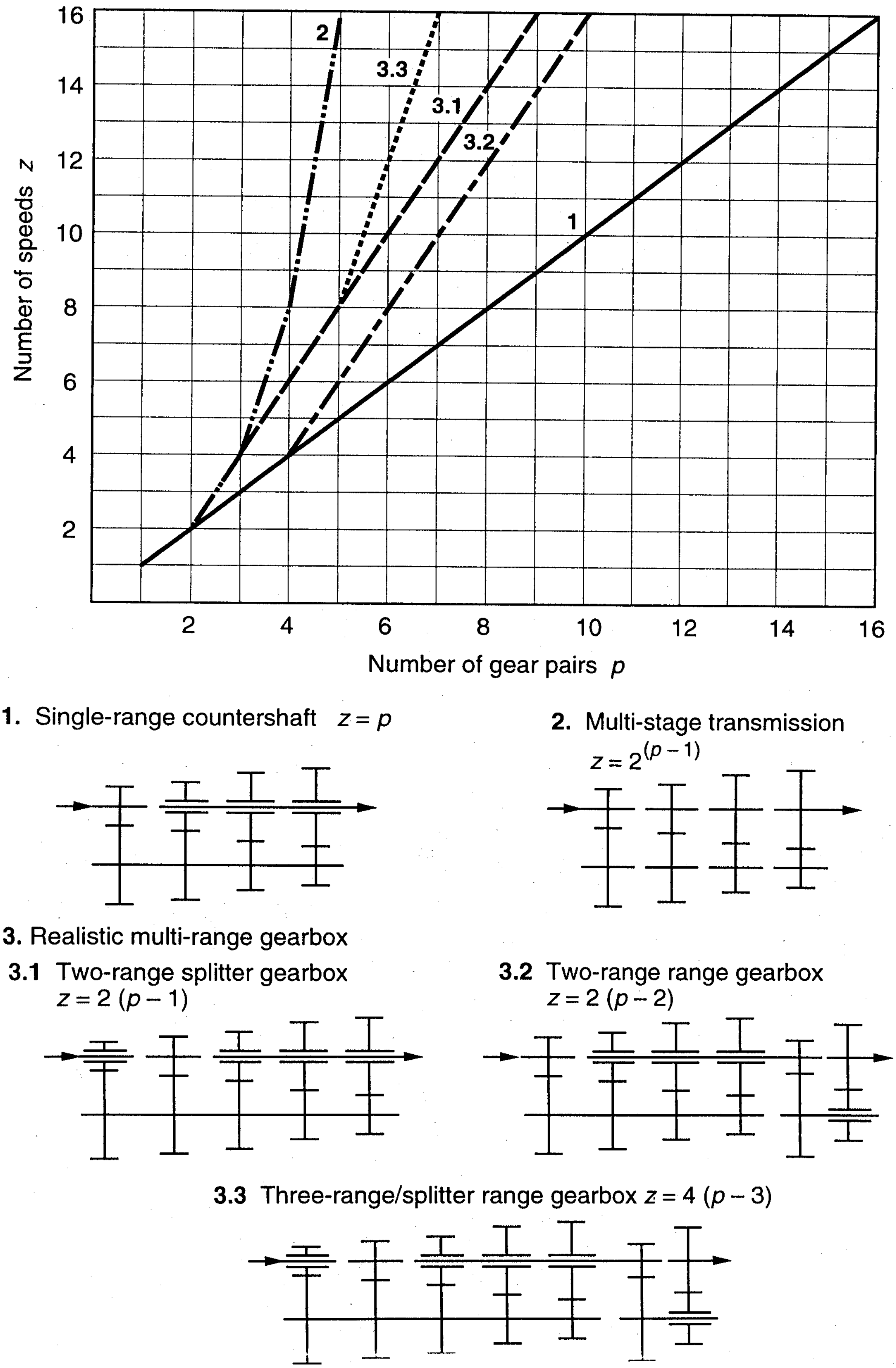


Figure 6.43. Effect of the range type on the number of gearwheels and speeds

Equation 6.6 applies when all gearwheels can be shifted and each gearwheel has its own shaft. Such a transmission, in which the power is transmitted several times from one shaft to the other in individual gears, is also known as a *multi-stage transmission*. In multi-stage transmissions the ranges are reduced to individual gear pairs. In addition to the high level of engineering complexity for the shaft junctions, several junctions have to be shifted at the same time when changing gear. Multi-stage transmissions with up to 5 gear pairs and 16 gears are shown in Figure 6.42. Shifting two or more junctions at the same time can lead to high shifting times.

Depending on the design, the following transmissions can be constructed using $p = 6$ gear pairs (Figure 6.43):

- ❍ single-range countershaft transmission, Fig. 6.43/1: 6 forward gears
- ❍ multi-stage transmission, Figure 6.43/2: 32 forward gears
- ❍ two-range splitter gearbox, Figure 6.43/3.1: 10 forward gears
- ❍ two-range range gearbox, Figure 6.43/3.2: 8 forward gears
- ❍ three-range splitter/range gearbox, Fig. 6.43/3.3: 12 forward gears

The multi-stage transmission is of no practical interest because of the many junctions to be shifted. If a splitter unit and a range-change unit are combined with a 4-speed main gearbox, this results is a 16-speed transmission. Here the ratio spreads of the three groups are selected in such a way that all 16 selectable combinations of gear steps are arranged in steps useful for the driver (Section 4.4.2 "Geometrical Gear Steps") (Figure 6.40c).

6.7.3 Practical Design of Two- and Three-Range Transmissions

The normal designs are two- and three-range transmissions with up to 16 gears (2 x 4 x 2) [6.17] (see also Figure 4.2). A larger number of gears is in principle possible, but in practice no longer relevant since it involves excessively frequent gearshifting by the driver.

The ZF 16 S 109 16-speed commercial vehicle transmission (Figure 6.44) can serve as an example of a 16-speed three-range type gearbox. The main gearbox is a 4-speed countershaft transmission. The two gear pairs of the countershaft type two-speed splitter unit are located on the transmission input side. A planetary type two-speed range-change unit is connected on the transmission output side.

The following principle applies to the design of commercial vehicle transmissions: The transmission must be designed in such a way that the largest possible number of gear pairs is acted on with a small change in ratio, and the smallest possible number of gear pairs with a high change in ratio.

The planetary design of the range-change unit in particular ensures compactness, bearing in mind that the range-change unit must have a large gear step, which is easy to achieve in a planetary design. The short overall length also ensures minimum shaft deflection in range-change units subject to high torque.

The countershaft transmissions discussed heretofore had only one countershaft located in the power flow. The transmission diagram shown in Figure 6.45 of the Eaton Twin Splitter transmission has two countershafts both for the 4-speed main gearbox and for the 3-speed rear-mounted splitter unit.

The power transmitted is split between both countershafts, and flows back to the main transmission shaft. The power split enables the gearwheels to be approximately 40% narrower than in a conventional countershaft transmission. The transmission is physically shorter, but wider. Short transmissions are advantageous especially in tractors. The shorter the transmission, the more favourable the proportions (the deflection angle resulting from the vertical offset and the longitudinal distance to the final drive) for the propeller shaft connected to the transmission.

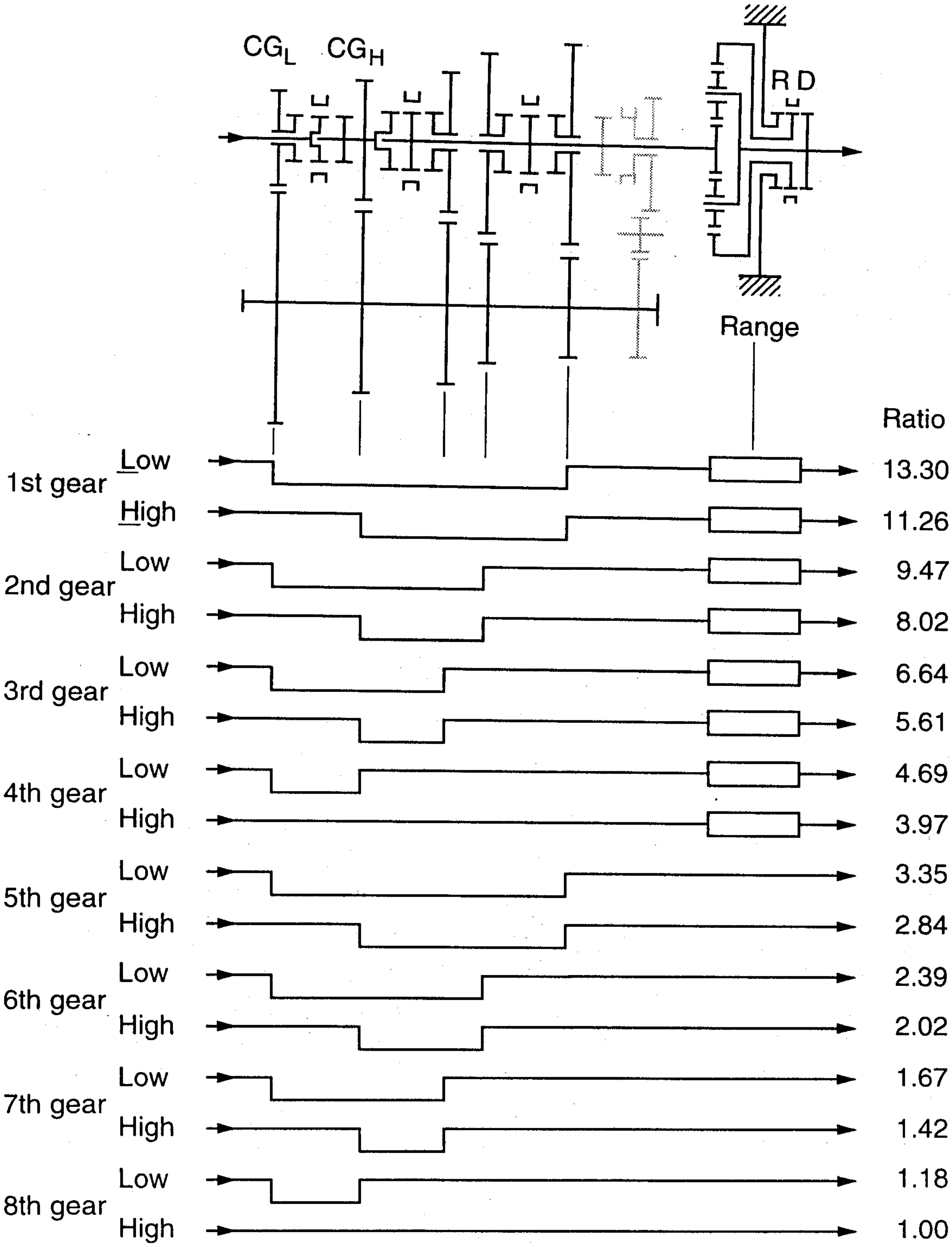


Figure 6.44. Gearbox diagram, power flows and ratios of a 16-speed three-range gearbox (ZF), 2 x 4 x 2 = 16 speeds, *production design Figure 12.8*

In order to ensure uniform loading (load compensation) of the gearwheels in both branches of the power splitter, the main shaft does not run in radial bearings but is merely radially guided. It is centred between the two countershafts when under load. Since the main shaft is not capable of absorbing large axial forces, straight cut spur gears are used. To still achieve good running characteristics, gearwheels with a high contact ratio $\varepsilon > 2.2$ and high contact gearing are used.

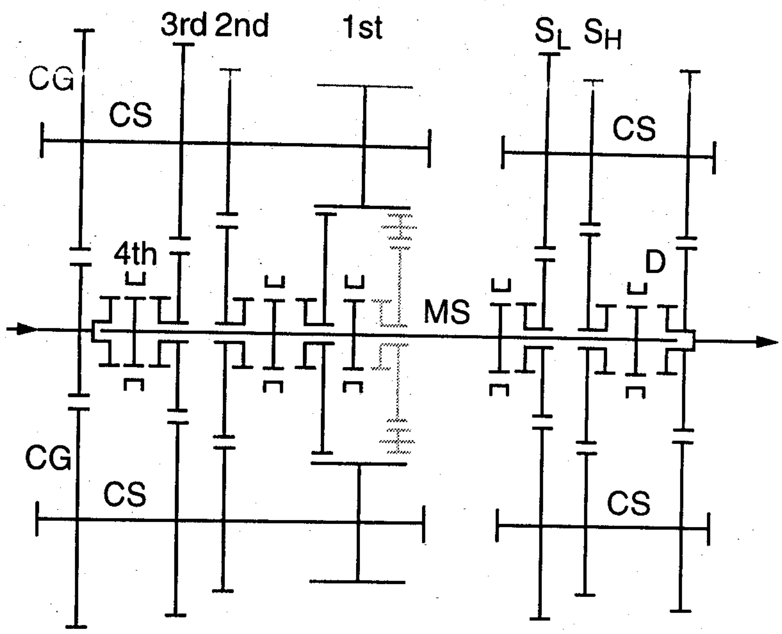


Figure 6.45. Gearbox diagram: Twin Splitter (Eaton).
4 x 3 = 12 speeds.
2-countershaft gearbox,
3-speed rear-mounted splitter unit;
CG Constant gear
CS Countershaft
MS Main shaft
S_L Rear-mounted splitter unit low
S_H Rear-mounted splitter unit high
D Direct
Production design Figure 12.9

The main gearbox of the twin splitter has four forward gears and one reverse gear. The rear-mounted splitter unit has three gears: one direct gear $i_D = 1.0$, one speed-increasing gear $i_{S,\,H} = 0.79$ and one speed-reducing gear $i_{S,\,L} = 1.28$. This gives 12 forward gears. Rear-mounted splitter units are not usually used, because of the face widths required (see also Table 6.11). It is nevertheless used in this case because of the low overall face width resulting from the power split. The main gearbox is constant mesh, the rear-mounted splitter unit is synchronised. The transmission is available as a semi-automatic.

Figure 6.46 shows the gearbox diagram of the Fuller (Eaton) RT 9513 Roadranger transmission. It also has two countershafts. The range-change unit is at first sight very similar to the Twin Splitter transmission. But in the Roadranger transmission the range-change unit is in the form of a combined splitter unit/range-change unit ($i_{S,\,H} = 0.87$; $i_{S,\,L} = i_D = 1.0$; $i_R = 3.38$).

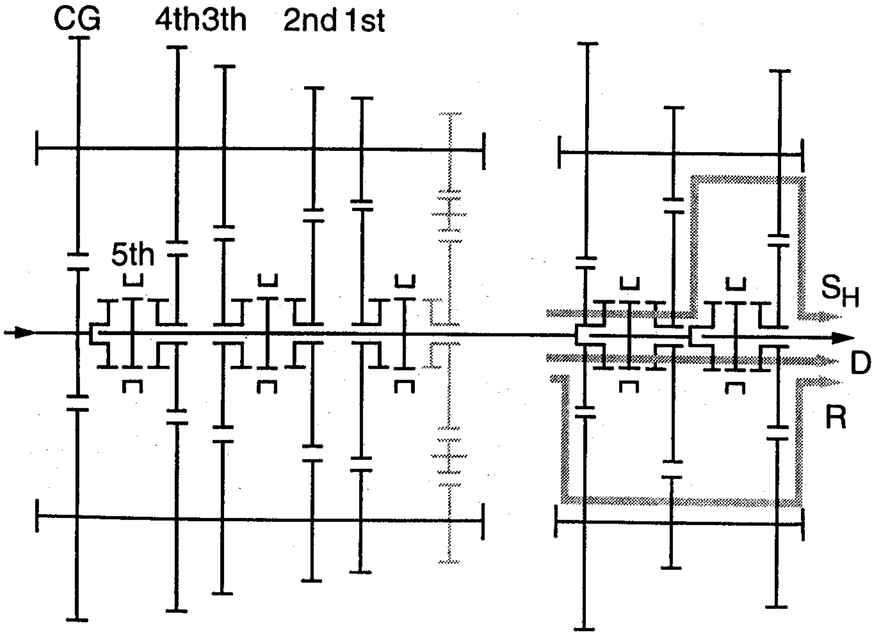


Figure 6.46.
Gearbox diagram:
Roadranger (Fuller)
4 x 3 + crawler
= 13 speeds.
2-countershaft gearbox,
3-speed rear-mounted
splitter/range-change
unit;
CG Constant gear
S_H Rear-mounted splitter unit high
D Direct
R Range

The main gearbox of the Roadranger transmission has five forward gears and one reverse gear, with the first gear being used only once in combination with the range-change unit as a crawler gear. The other four forward gears are combined with the splitter unit. This results in 13 forward gears for this transmission. The reverse gear can also operate with range-change unit, providing two reverse gears.

6.7.4 Semi-Automatic Manual Commercial Vehicle Transmissions

In line with the explanations in Section 6.2.2 the designation "semi-automatic" relates to the operations "Engaging the clutch/Moving off" and "Changing gear". In semi-automatic transmissions one of these processes is automatic. This results in the breakdown shown in Table 6.12 listing the various degrees of automation of transmissions from manual gearboxes (automation level 0) through to fully automatic transmissions (automation level 4).

Table 6.12. Degrees of automation of manual passenger car and commercial vehicle gearboxes

Degree of automation	Method of engaging drive	Shifting clutch action	Gear selection method
0	Foot-activated master clutch	Foot-activated clutch operation	Manual activation of a shift lever
1	Foot-activated master clutch	Automated clutch operation	Manual activation of a shift lever
2	Automated master clutch	Automated clutch operation	Manual activation of a shift lever
3	Automated master clutch	Gear change initiated by foot-activated clutch operation	Manual gear pre-selection from keypad
4	Automated master clutch	Automated clutch operation	Automated gear selection and engine management

In automation level 2 the driver just engages the desired gear by activating the gearshift lever, then the engaging action and moving off take place automatically. One example of this design is the ZF torque converter clutch shown in Figure 6.47, combined with the 16-speed transmission in Figure 6.44.

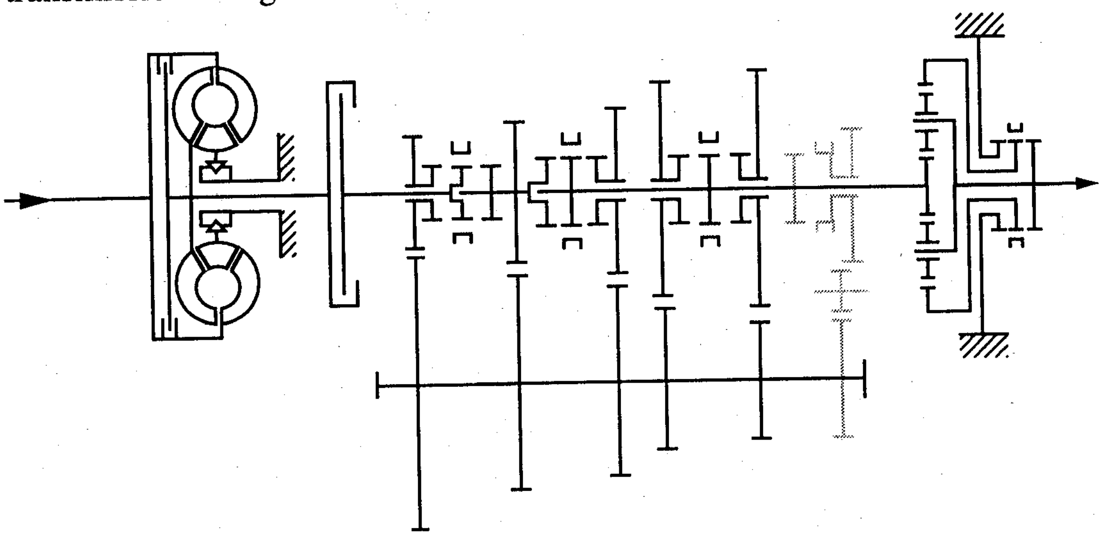


Figure 6.47. Gearbox diagram of a 16-speed commercial vehicle gearbox with torque converter clutch (ZF), *production design Figure 12.10*

In this transmission, moving off is a function exclusively of the converter, relieving the load on the clutch, which is used only to interrupt the power flow when changing gear (see also Figure 6.22).

In automation level 3 the driver selects the gear or follows an automatic gear (shift) recommendation. By activating the clutch the driver triggers an automatic shift into the recommended or selected gear. (See also Section 9.1 "Shifting Elements").

6.7.5 Fully Automatic Commercial Vehicle Transmissions

In fully automatic commercial vehicle gearboxes, both engaging the clutch/moving off and changing gear are automated. The following designs can be distinguished:

- conventional 3-speed to 7-speed automatic transmissions (consisting of torque converter and rear-mounted planetary gear),
- fully automatic commercial vehicle countershaft transmissions,
- continuously variable transmissions.

See also Section 6.6.3 "Fully Automatic Passenger Car Transmissions". Conventional automatic transmissions have powershift. The electronically controlled commercial vehicle countershaft transmissions have shifting with power interruption. Countershaft transmissions have the advantage of being based on conventional range gearboxes and being able to provide up to 16 speeds with acceptable production engineering requirements. By comparison, conventional automatic transmissions are available with a maximum of 7 speeds.

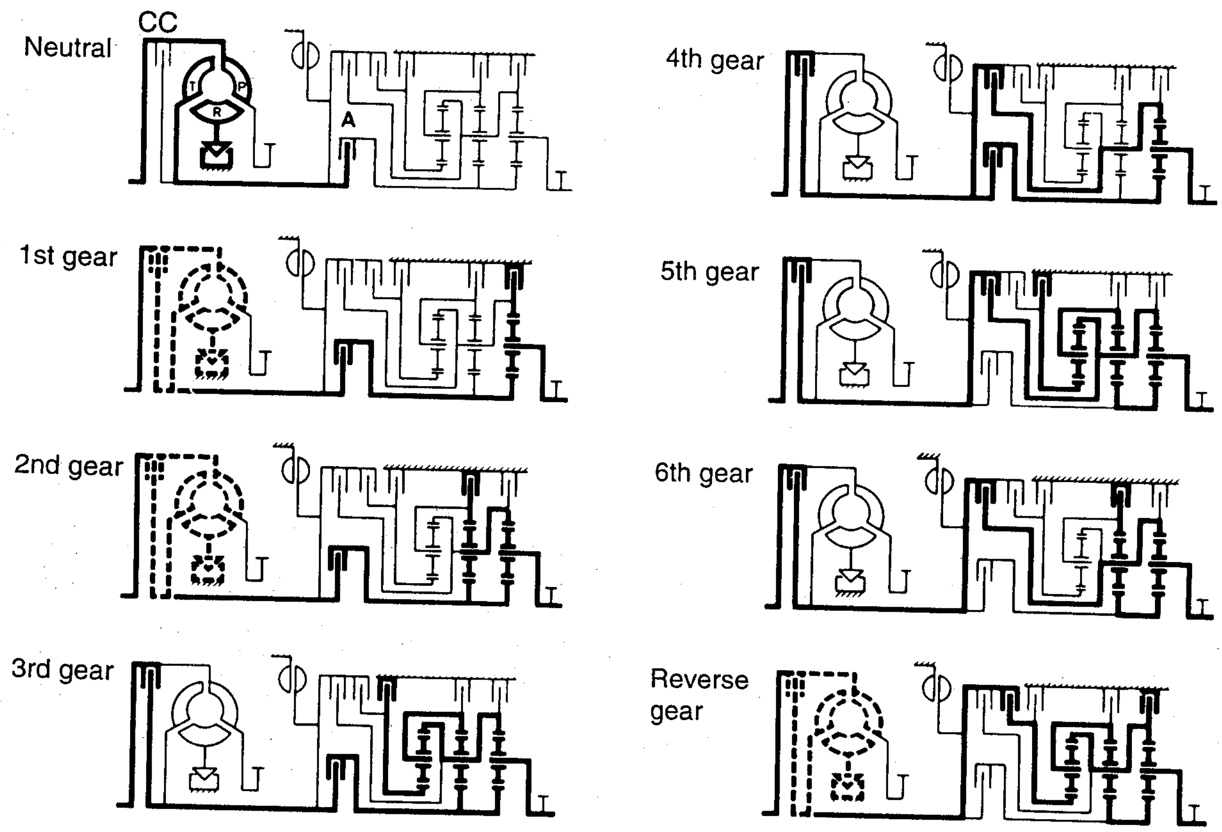


Figure 6.48. Power flow in a conventional 6-speed commercial vehicle automatic gearbox (ZF). In the starting gears 1 and 2, the converter only operates until the lock-up clutch closes (dotted line), *production design Figure 12.14*

The gearbox diagram of the ZF 6 HP 600 transmission from the Ecomat range (Figure 6.48 [6.18]) is typical of conventional automatic transmissions. This is a 6-speed transmission with torque converter and lock-up clutch, and integral hydraulic retarder.

The designs of automatic transmissions for commercial vehicles are very similar to those for passenger cars; they are by definition produced only in the form of converter with powershift planetary gear transmission. There are versions with up to 7 gears for engine ratings up to over 700 kW [6.19]. (Production design Renk REMAT, Figure 12.15). Transmissions for such high power ratings are developed specifically for use in special-purpose vehicles such as fire-fighting appliances and building-site vehicles. The resources required to achieve a higher number of speeds in this standard design makes this impractical.

6.7.6 Continuously Variable Transmissions for Commercial Vehicles

Mechanical continuously variable transmissions are currently not in prospect for commercial vehicles (greater than 4.0 t gross weight). Chain converter transmissions as used in passenger cars (up to 300 Nm) do not have the necessary torque capacity for use in commercial vehicles. There are test vehicles with toroidal variators (frictional wheel) which have higher torque capacity than chain converters.

In practice therefore only continuously variable hydrostatic transmissions are used, principally in combination with gearwheel transmissions. Purely hydrostatic transmissions can be considered only for power take-offs, because of their poor efficiency.

Figure 6.49 shows the SHL continuously variable hydrostatic power split transmission produced as prototypes up to the year 1995 by Voith [6.20]. This transmission transmits the input power partly through a 4-shaft planetary differential, and a hydrostatic pump/motor unit mechanically linked to it. Three different operating ranges can be preselected by connecting the transmission components to each other in different ways.

The continuously variable change in ratio is achieved by varying the flow rate and displacement capacity of the hydraulic components. The percentage power transmission of the hydraulic components ranges from 100% to 0% in the first operating range, and 0% to 25% in the second operating range, and 0% to 27% in the third operating range.

Note also that in this transmission the braking energy can be stored in an accumulator in the form of pressurised oil, and fed back to the hydraulic motor when required. If the kinetic energy is not stored during braking it can be converted into heat by the hydraulic units acting as pumps against a valve. This relieves the conventional service brakes, acting like a retarder.

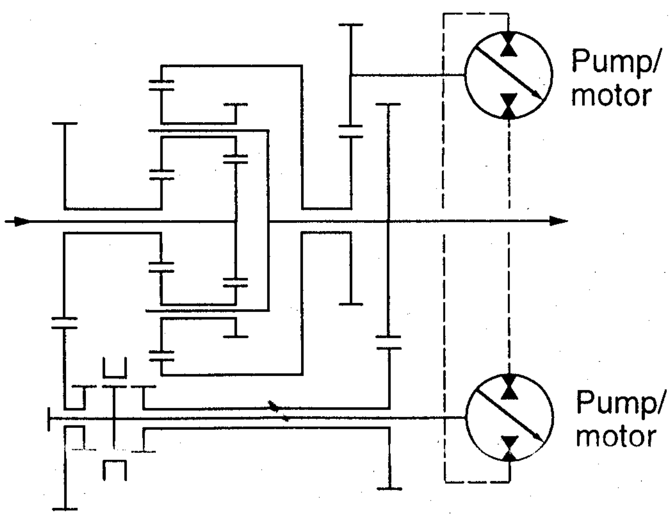


Figure 6.49. SHL continuously variable hydrostatic power split transmission (Voith)

In the late 1990's hydrostatic power split transmissions are entering the agricultural tractor market.

Electric transmissions pursue another avenue of continuously variable conversion. The mechanical parts in the power transmission chain are in this case replaced by electrical and electronic components [6.21].

6.8 Transfer Gearboxes and Power Take-Offs

The power delivered by the engine is fed to the driven wheels. In vehicles with more than one driven axle (all-wheel drive passenger cars and commercial vehicles) the power has to be *distributed* to the various powered axles. Over and above that there is often the requirement, especially in commercial vehicles, to also split the power to auxiliary units.

In automotive engineering a distinction is made between

- *differential gear units* (Section 6.10):
  - interaxle differential = *transfer box:* *longitudinal split* (viewed in the direction of travel) of the power to more than one powered axle,
  - interwheel differential = *differential gear unit:* *transverse power split* to the drive wheels of one axle,
- *power take-offs* power split from the actual power train to auxiliary units, e.g. concrete mixer.

Transfer Gearboxes

Transfer boxes can be broken down into four groups according to their construction (Figure 6.50). To avoid serious distortion in the power train when cornering, rigid power distribution can only be used for axles that are close together (tandem axles).

Front axle drives (or rear axle drives) that can be engaged as required are in use in vehicles which require all-wheel drive only part of the time, in poor traction conditions. The distortions are partly offset by wheel slip.

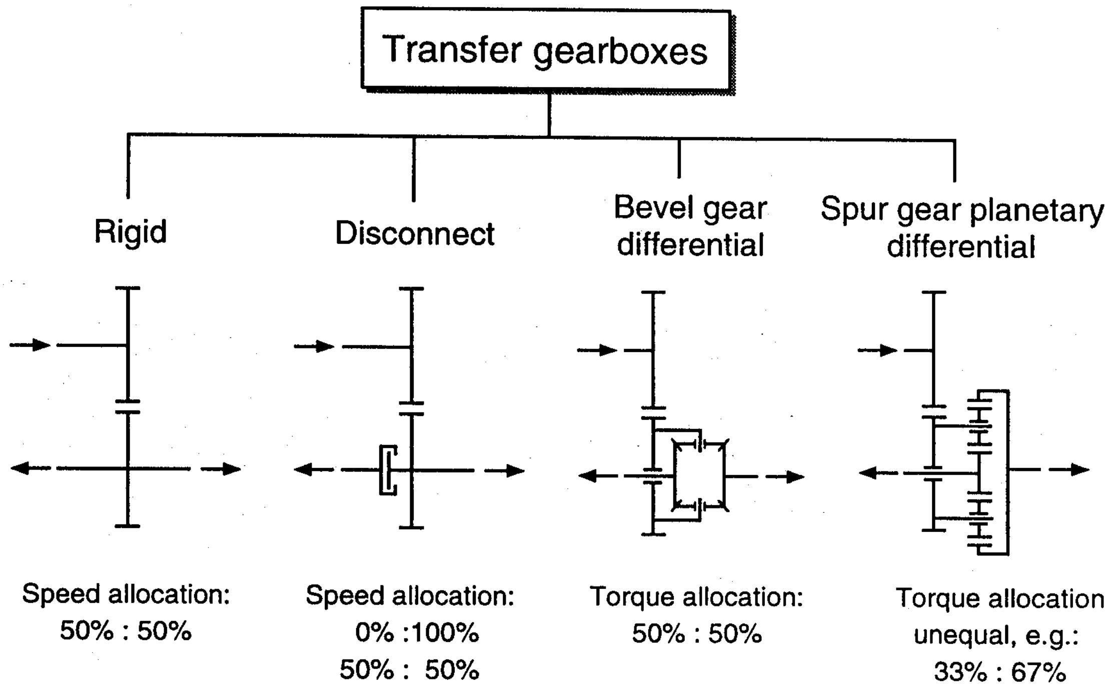


Figure 6.50. Transfer gearboxes categorised by their construction

For passenger cars and commercial vehicles with permanent all-wheel drive, the only viable option is a transfer box with differential. The differential makes it possible to equalise the speed and the forces between the power axles. With a bevel gear differential, the torque is split equally between the front and rear axles. In straight differentials the split is unequal.

All-terrain passenger cars and commercial vehicles are also fitted with transfer boxes including a range-change unit. In the transfer box there is a choice between off-road and on-road modes. The high torque multiplication of the off-road gear thus only comes into effect after the main gearbox. For heavy commercial vehicles there are also main gearboxes with integral transfer box.

Existing transfer box designs are shown and discussed in Section 12.7. The differential gear units, which are closely related to transfer boxes, are discussed in Section 6.10, as are differential locks and locking differentials. Section 12.6 examines the various existing designs in greater detail.

Power Take-Offs

In commercial vehicles there is often a need for power to supply auxiliary units. The power flow can either be switched entirely to a power take-off, or can be split into a vehicle drive branch and a power take-off branch.

Power take-offs are commonly used for pumping water or mud or for driving hydraulic pumps, winches, fire-fighting ladders, crane superstructures, or sweepers. Power take-offs can be divided into two groups:

- clutch-controlled power take-offs,
- engine-controlled power take-offs.

Clutch-Controlled Power Take-Offs

In clutch-controlled power take-offs the power split to the auxiliary unit is located after the master clutch. Power flows to the auxiliary unit only when the master clutch is engaged. They are for example coupled to the countershaft of the main gearbox by means of a dog clutch. Clutch-controlled auxiliary units can be operated with the vehicle stationary (transmission in neutral), or with the vehicle moving. Since they are connected to the countershaft, they constitute an additional load on the transmission, and especially the synchronisers. The power take-off can be in the form of an axial extension of the countershaft (Figure 6.51), variant 1, or an additional gear stage at another point in the transmission housing (Figure 6.51), variant 2.

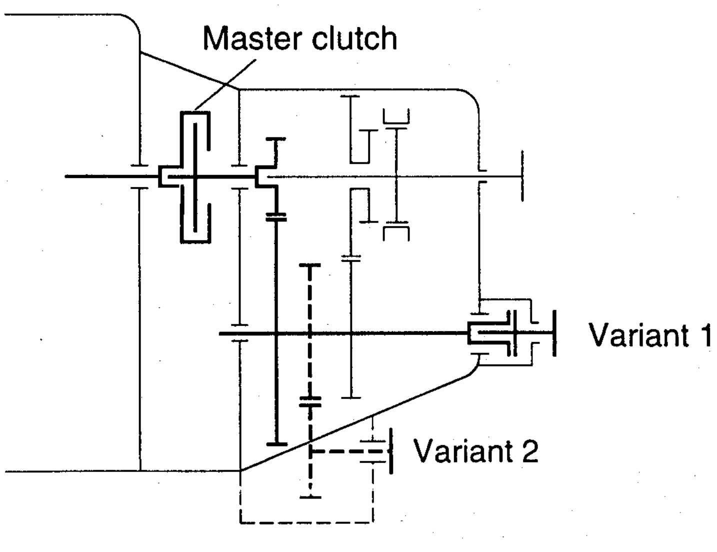


Figure 6.51. Clutch-controlled power take-off

Engine-Controlled Power Take-Offs

Engine-controlled power take-offs are located on the engine side of the clutch, viewed in the direction of power flow (Figure 6.52). This is achieved using a hollow shaft through which the main transmission drive-shaft passes. The power take-off is thus independent of frictional engagement of the drive clutch. Since the power flow does not go through the main gearbox, a much greater power flow can be achieved with an engine-controlled power take-off than with a clutch-controlled one. This design can function equally with the vehicle stationary or moving.

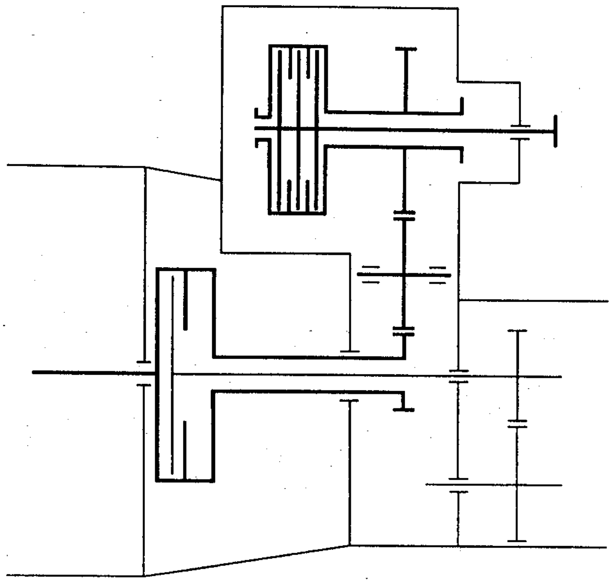


Figure 6.52. Engine-controlled power take-off

6.9 Final Drives: Formats, Performance Limits, Transmission Ratios

There are different ways of designing the final drive of a vehicle. The alternatives involve the position of the engine relative to the direction of travel, the position of the engine relative to the transmission, and the ratio allocation between the transmission, transfer box, axle gearbox and hub gearbox.

The final drive contains the following assemblies, depending on the design:

- axle gearbox, differential carrier,
- hub gearbox, wheel hub,
- differential gear unit, locking differential,
- drive shafts, propeller shafts,
- housing,
- steering knuckles,
- axle beam with suspension tubes.

Because of the major differences in the final drive designs, a distinction is made below between passenger cars and commercial vehicles.

"Typical Designs of Final Drives" are given in Section 12.5, illustrating the main final drive variants currently in mass production. Section 6.10 discusses differential gear units in detail. Examples of differential gear unit designs are given in Section 12.6.

6.9.1 Final Drive Systems for Passenger Cars

The basic axle gearbox designs shown in Figure 6.53 can be derived from the numerous possible configurations of assemblies in the power train:

- spur gear final drive,
- bevel gear final drive of helical bevel or hypoid design,
- worm gear final drive.

Other options for the final transmission are belt drives (DAF VARIOMATIC) and chain drives (motorcycles).

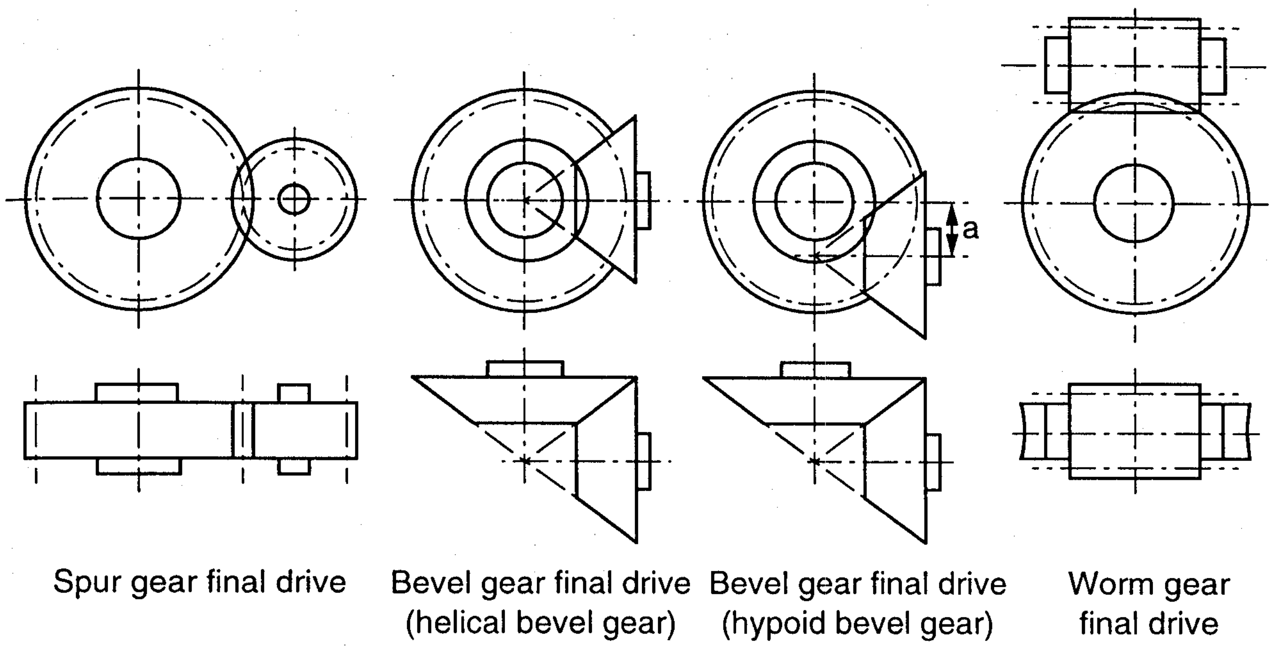


Figure 6.53. Diagrammatic view of the formats for passenger car axle gearboxes

Spur Gear Final Drives

Spur gear final drives are now common because of the popularity of vehicles with transverse front-mounted engines. The axle gearbox is driven either directly by the output shaft of the transmission, or by idler gears. It is normally favourable for the differential cage drive if the engine and transmission are mounted side by side, with the disadvantage of having drive shafts of unequal length to the wheels. The reasons for their popularity are the compactness and low production cost of spur gears, normally helical cut. There are also maintenance advantages in combining transmission and final drive, since in most cases the lubrication system is the same.

Bevel Gear Final Drive

In power trains where the engine is longitudinally mounted, and in all-wheel drives, the power flow to the wheels has to be turned through 90°. A bevel gear final drive is one of a wide variety of means of achieving this. The final drive can be integrated in the transmission housing (transaxle design), or designed as an independent assembly, as in vehicles with standard drive. In the case of bevel gear final drives a distinction can also be made between helical bevel drive and hypoid drive, according to the engagement of bevel gear and crown gear (Figure 6.53). In passenger cars, hypoid drives are usually used, in which the bevel drive pinion engages below the axial centre of the crown gear.

This offset makes the diameter of the bevel drive pinion larger, and the crown gear can be smaller for the same load than in helical bevel drives in which the axes intersect.

The sliding friction between the tooth flanks, which contributes substantially to reducing noise, creates very high surface pressure forces which demand pressure-resistant oil (hypoid oil) to lubricate the final drive (see also Section 11.2 "Transmission Lubricants"). The offset also enables the propeller shaft to be mounted lower, reducing the size of the transmission tunnel.

Worm Gear Final Drive

There are now no more axle gearboxes with worm gear final drive in production. This type of drive was used in some Peugeot models in the 1970s. They are now rarely used, mainly because of the difficulty and expense of manufacturing the worm and the worm gearwheel. But the worm gear final drive does have significant advantages. It offers large multiplications in a compact space. The worm can be located below or above the worm gearwheel. The former case means a low centre of gravity and low-mounted propeller shaft, eliminating the obtrusive propeller shaft tunnel. Mounting the worm above the gearwheel gives the vehicle good ground clearance, a particular advantage for off-road vehicles. If worms are used in multi-axle drives, then a continuous propeller shaft coupled to the axles can be used. Worm gear drives are superior to all other types of drive as regards quiet running, because meshing is sliding, and there is always a film of oil between the frictionally engaged tooth flanks. Since there are large axial forces with worm gear drives, the worm bearing requires careful consideration.

Transmission Ratios

The ratio of the power train i_A in passenger cars is derived from the transmission ratio i_G and the final ratio i_E (see also Section 4.1). In most passenger car transmissions the transmission ratio i_G of the largest gear is fixed at $i_G \approx 0.7–1.0$. The ratio of the power train i_A is fixed by selecting the final ratio i_E to match the power, the desired maximum final speed, etc. For individual passenger car final drives the following ratios are typical:

- spur gear final drive $i_E \approx 3.0–5.5$,
- bevel gear final drive $i_E \approx 2.5–5.0$,
- worm gear final drive $i_E \geq 5.0$.

The small ratios occur in powerful passenger cars and sports cars, whilst the large ratios occur in low-powered small passenger cars and all-wheel drive vehicles. The specification of the final drives relates to the input torque available, i.e. the output torque of the transmission.

Comparison of Different Designs

Table 6.13 provides a comparison of the main basic designs of passenger car final drives: spur gear, bevel gear and worm gear final drive. The comparison criteria are quiet running, manufacturing cost, arrangement of bearings, lubrication, efficiency, service life, load capacity and space requirements.

6.9.2 Final Drive Systems for Commercial Vehicles

The ratio of the final drive i_E of a commercial vehicle is determined by the ratio of the centre transmission i_M and the hub transmission i_N.

In commercial vehicles the centre transmission can be of single-stage or multi-stage design. It accommodates the drive bevel gears or the worm drive, the differential gear unit and, in the case of multi-stage drives, spur gear-sets or planetary-sets and the drive through to the next axle. As mentioned above, the centre transmission can be subdivided into single-stage and multi-stage axle gearboxes. Some designs of centre transmissions and hub gearboxes are given in Section 12.5.2.

Table 6.13. Un-weighted evaluation of final drive formats

Final drive / Feature	Spur gear	Bevel gear: Helical bevel drive	Bevel gear: Hypoid bevel drive	Worm gear
Quiet running	0	0	+	++
Manufacturing cost	++	+	0	– –
Bearings	++	0	0	0
Lubrication	++	++	0	0
Efficiency	++	++	++	+
Service life	++	+	+	++
Load capacity	0	+	+	++
Space requirements	+	0	+	+
Total	**11+**	**7+**	**6+**	**8+**
++ very good; + good; 0 satisfactory; – bad; – – very bad				

Single-Stage Centre Transmissions

Depending on the type of drive, the single-stage centre transmission (Figure 6.54) is divided into

- ❍ bevel gear drive, Figure 6.54a,
- ❍ double bevel gear drive (split bevel drive), Figure 6.54b, and
- ❍ worm gear drive, Figure 6.54c.

The spur gear final drive is used only in commercial vehicles of up to 4.0 t gross weight rating, and will not be discussed further here.

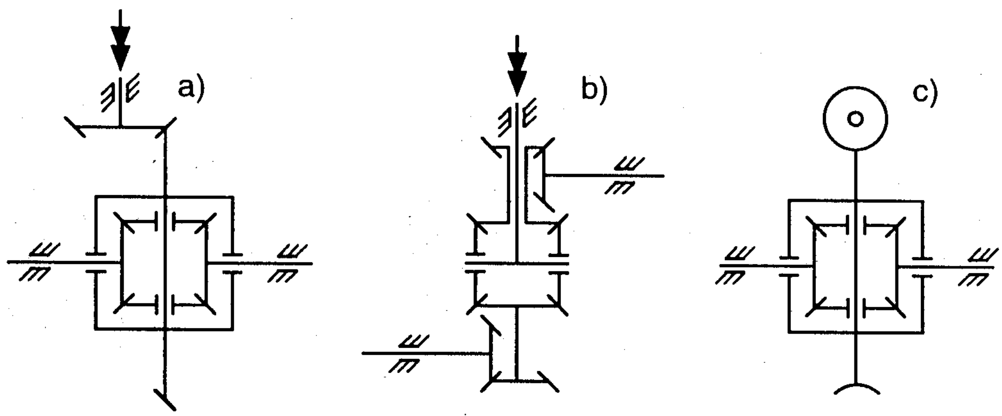


Figure 6.54. Diagrammatic view of the single-stage centre transmissions. **a)** Bevel gear drive; **b)** Double bevel gear drive; **c)** Worm gear drive

Multi-Stage Centre Transmissions

In the case of multi-stage centre transmissions (Figure 6.55) there are several designs:

- ❍ countershaft, front mounted, Figure 6.55a1,
- ❍ countershaft, top mounted, Figure 6.55a2,
- ❍ two-speed with spur wheel countershaft, Figure 6.55c,
- ❍ two-speed with planetary gear, Figure 6.55d.

The term "top mounted" refers to the fact that the propeller shaft sits higher than the drive shafts to the wheel hubs. In this design it is easy to arrange a drive through to a second driven axle.

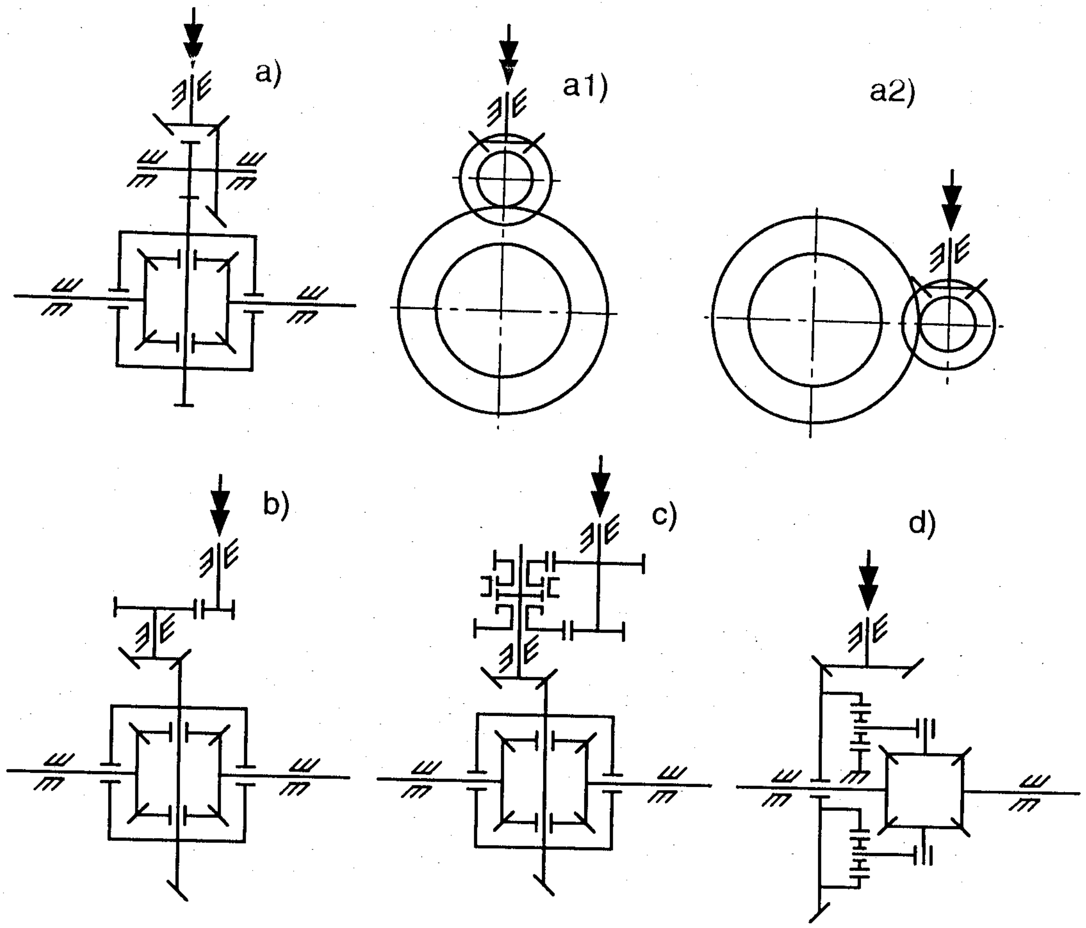


Figure 6.55. Diagrammatic view of multi-stage centre transmissions. **a)** Countershaft; **a1)** Front-mounted; **a2)** Top-mounted; **b)** Countershaft; **c)** Two-speed with spur wheel countershaft; **d)** Two-speed with planetary gear (top: engaged; bottom: not engaged)

If the propeller shaft and the drive shafts are at the same height, the drive of the centre transmission is thus directly from the front, hence the term "front mounted".

Hub Gearboxes

The necessary multiplication i_E can be distributed either just in the centre transmission, or between the centre transmission and the hub gearbox. By increasing the torque directly at the hub gearboxes, the centre transmission and drive shafts to the wheel hubs can be smaller. The hub gearboxes can include the following designs:

- without wheel hub gearbox, Figure 6.56a,
- with external toothed spur gear countershaft with drive above, below or at the same height as the wheel axis, Figure 6.56b,
- with internal toothed spur gear countershaft, Figure 6.56c, d,
- spur gear planetary drive, Figure 6.56e, f,
- with bevel gear planetary gear (fixed at $i_E = 2$), Figure 6.56g.

Figure 6.56 shows a schematic view of possible hub gearboxes. In combination with the single-stage centre gearbox, which is small because of transmission ratio, an external toothed spur gear countershaft with drive above the wheel axis (Figure 6.56b) creates a gantry axle. It is mainly used for off-road vehicles that require a lot of ground clearance beneath the axles. Mounting the drive below the wheel axis is favourable for low-frame vehicles.

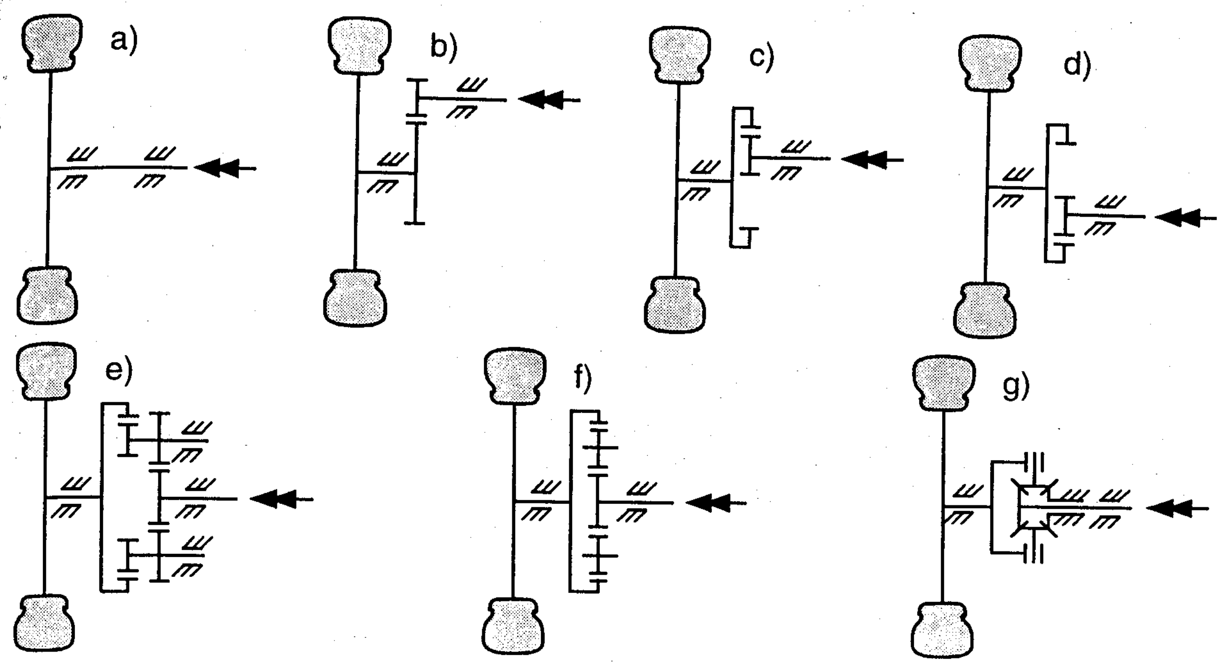


Figure 6.56. Diagrammatic view of the hub gearboxes. **a)** Without wheel hub drive; **b)** Spur gear countershaft with external toothing; **c)** Spur gear countershaft with internal toothing, above the wheel axis; **d)** Spur gear countershaft with internal toothing, below the wheel axis; **e)** Spur gear planetary drive, double planetary drive; **f)** Spur gear planetary drive; **g)** Bevel gear planetary drive

6.10 Differential Gears, Differential Locks and Locking Differentials

Single-axle drive is the minimum for passenger cars and commercial vehicles, for reasons of directional control and traction. The engine power must therefore be distributed to a left and a right driving wheel, in the simplest case by means of a non-split wheel drive shaft as shown in Figure 6.57. But when cornering, the outer wheel covers a greater distance than the inner wheel, which with a rigid drive causes tyre abrasion and high wear and stress to the power train resulting from distortion.

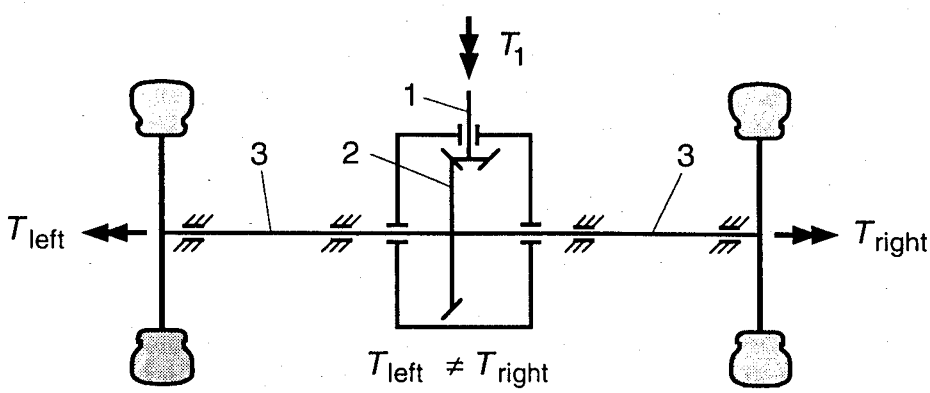


Figure 6.57. Diagrammatic view of a rigid, non-compensating differential gear. *1* Drive; *2* Bevel gear; *3* Axle shaft

6.10.1 Principles of Differential Gears

A transmission is therefore needed which, unlike a rigid direct drive without a split output shaft, permits free *speed and force compensation*. This gear unit must provide a *transverse torque split* 50% : 50% to the left and right driving wheel. *Transverse* relates to the direction of travel of the vehicle (see Section 6.8).

Differential gear units of this type are also necessary between the axles of vehicles with more than one powered axle. They are only non-essential when travelling very slowly or on loose surfaces, or when there is a sufficiently small gap between the axles concerned. *Interaxle compensation* can be represented with asymmetrical torque distribution, depending on the traction potential of the axles and the handling required. The usual figures for all-wheel drive passenger cars are ratios of 50% : 50% or 33% : 67% for the front and rear axles respectively.

In automotive engineering the interaxle differential for restricting the power to different power axles is generally referred to as a transfer box (see Section 6.1.3 "Four-Wheel Drive Passenger Cars", 6.8 "Transfer Boxes" and 12.7 "Typical Designs of Four-Wheel Drives".) The interwheel differentials for splitting the power to the drive wheels of an axle are usually integrated in the final drive of the power train. (In normal usage, the term "differential" applies not only to the differential unit proper, but also to its drive and associated transmission, and any wheel reduction gears.)

According to [6.8], differential drives with two or more degrees of freedom F can be used for the requirements described. These are generally simple or compound planetary gears for overlaying speeds and power. Any epicyclic system with three freely moveable, coaxial shafts and which is positively actuated by two input or output movements could be used as a differential for vehicles.

In [6.22] ALTMANN describes ways of depicting differential gears by means of gear-wheel pairings, of which the following types are now common:

- bevel gear differentials,
- spur gear planetary (straight differentials),
- worm gear differentials.

Straight differentials are usually used as interaxle differentials because of the possibility of asymmetrical torque distribution, and bevel gear differentials are standard for inter-wheel compensation. The worm gear differential (TORSEN) is used in both types. Some selected examples are discussed in Section 12.6. The various possible ways of driving the differential units are explored in Section 6.9 "Final Drives".

In contrast to Figure 6.57, Figure 6.58 shows no rigid through drive, but split axle shafts with an interposed bevel gear differential. The torque T_1 introduced through the drive *1*, for example a helical or hypoid bevel drive, is transmitted via the differential cage *2* to the compensating bevel gears *3*, which act like a balance beam to ensure a torque equilibrium $T_{\text{left}} = T_{\text{right}}$ between the left and right output sides. As long as there is no slip at the drive wheels, the following applies for the rotational speeds:

Rotational speed of the outer wheel when cornering, or the one with less grip:
$n_a = n + \Delta n$
Rotational speed of the inner wheel when cornering, or wheel with more grip:
$n_i = n - \Delta n$

where n is the input speed of the crown gear and Δn is the speed difference between the output speed of the outer wheel when cornering and the input speed of the differential. When travelling in a straight line, the differential cage *2*, the axle bevel gears *4*, the axle shafts *6* torsionally locked to the axle bevel gears, and the differential bevel gears *3* inside the cage rotate as a block.

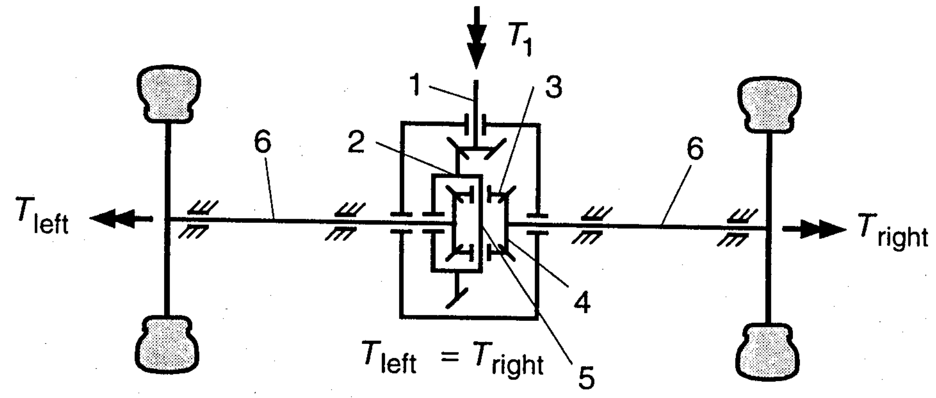


Figure 6.58. Principle of a rear-axle gear unit with bevel gear differential. *1* Drive; *2* Differential cage; *3* Differential bevel gears; *4* Axle bevel gears; *5* Differential shaft; *6* Axle shafts

There is no relative movement between the differential shaft *5* and the differential bevel gears mounted on it. When cornering, one axle shaft has to rotate faster than the opposite one; axle bevel gears and differential bevel gears move on rolling contact. This enables speed compensation between the wheels.

6.10.2 The Need for Locking

The design of conventional differential gears described in Section 6.10.1 has two important advantages for automotive engineering:

- The rotational speeds of the drive wheels can be adjusted independently of each other according to the different distances travelled by the left and right wheels.
- The drive torque is symmetrically distributed to both drive wheels, without any yawing moment.

These two advantages are however offset by a serious disadvantage. When the frictional potential of the two drive wheels are different, the propulsive forces transmitted to the road surface for both drive wheels depends on the smaller of the two. This comparison relates in this case to interwheel compensation in the axle gearbox, but applies analogously to interaxle compensation between different power axles. This means that a wheel standing on ice will spin and the other wheel standing on asphalt does not receive more torque than the one that is spinning. The vehicle can therefore not move off. In order to overcome this disadvantage of conventional differential gears, the compensating action has to be inhibited. This can be effected in different ways:

- By means of a *differential lock*. It can be activated manually or automatically by mechanical, magnetic, pneumatic or hydraulic means, and blocks any compensating action 100% by locking the differential unit. This makes the axle rigid again, with all the consequent advantages and disadvantages. The use of such a traction control system is thus appropriate where there is inadequate traction for one wheel or one axle, and should preferably automatic and temporary.
- By using self-locking differentials, also known as limited-slip or *locking differentials*. These are differentials with a compensating action that is deliberately tight and restricted. This enables them to transmit torque to one wheel or axle even when the other wheel or axle is spinning because of poor grip. This means

losing the advantage of power transmission without yawing moment. The free adaptation of both wheel speeds to the different distances travelled by the two tracks is restricted. The axle shafts are more stressed because of the torque redistribution. Locking differentials are divided into load or torque controlled, and speed or slip controlled. The former lock the differential action as a function of the torque applied, the latter as a function of the speed difference of two out of three of the differential gear shafts.

- Using externally activated *differential brakes*. These systems are usually electronically controlled and hydraulically activated, and the degree of locking can be varied within a wide range, often from 0 to 100%, as a function of the driving conditions, normally being unlocked or only slightly locked. The advantage of such systems is that the control can inhibit the compensating function to match the driving conditions. This largely avoids negative effects on handling in situations in which a locking differential produces an unwanted locking effect.
- By combinations of the above solutions.

6.10.3 The Interlock Value

The interlock value S is a key design variable, representing the degree of inhibition of the compensating action. The interlock value for the degree of inhibition of the interwheel compensation S_{trans} is defined as follows:

$$S_{\text{trans}} = \frac{\text{Locking torque } T_B}{\text{Propulsive torque } T} = \frac{T_{\text{right}} - T_{\text{left}}}{T_{\text{right}} + T_{\text{left}}}. \tag{6.7}$$

The interlock value for the degree of inhibition of the interaxle compensation S_{lengths} is derived analogously thus:

$$S_{\text{lengths}} = \frac{\text{Locking torque } T_B}{\text{Propulsive torque } T} = \frac{T_{\text{front}} - T_{\text{rear}}}{T_{\text{front}} + T_{\text{rear}}}. \tag{6.8}$$

By definition the interlock value S is in the range 0 to 1, and is often expressed as 0% to 100%. An interlock value of 0% describes a loss-free, non-locking differential gear; a figure of 100% represents a rigid direct drive.

In front-wheel drive passenger cars the interlock values must be kept low (maximum 17%), because of undesirable effects on the steering. The interlock values of locking differentials in rear-wheel drive passenger cars are between 25% and 50%, and in commercial vehicles up to 75%. If a locking differential with fluid clutch is used, the interlock value can reach 100% with large differences in rotational speed and high thermal stress, the so-called "hump effect".

The way the interlock works is that with a locking differential where S = 50%, a maximum of 75% of the drive torque can be fed to the wheel with the better grip, with at least 25% going to the wheel which is tending to spin. The difference between these two figures is S = 50%; the interlock value is as it were the "distribution figure" related to the total torque transmitted, i.e. $T_{\text{left}} + T_{\text{right}}$. In other words, the higher the interlock value, the more torque is channelled through the differential brake as braking torque or locking torque T_B rather than distributed by the differential. The interlock value is thus also a measure of the power distribution between the differential and differential brake.

An example is set out below to illustrate the limited use of locking differentials as a traction control system. The example relates to a vehicle standing on a surface which is slippery on one side where $\mu_{left} < \mu_{right}$, equipped with a locking differential with an interlock value $S_{trans} = 0.3$. A maximum of $T_{left} = 25$ Nm can be transmitted to the road surface with the left wheel. It follows from Equation 6.7 that:

$$T_{right} = T_{left} \frac{1 + S_{trans}}{1 - S_{trans}} . \tag{6.9}$$

Thus a torque of $T_{right} \approx 46.4$ Nm can be transmitted to the right wheel regardless of the engine torque available. The total torque transmitted amounts to only $T \approx 71.4$ Nm. This calculation shows the limited potential of locking differentials, since depending on the driving resistance (gradient, etc.) this torque could be insufficient for propulsion.

Interlock values of slip-related and load-related locking differentials have to be regarded differently: a purely load-related self-locking differential has a fixed, unchanging interlock value. This means that whatever the amount of input torque, the percentage determined by the nominal interlock value is always "diverted".

A purely slip-dependent locking differential produces a braking torque independent of the input torque, as a function of the speed difference arising. This gives rise to higher transient interlock values with small input torque, and smaller transient interlock values with large input torque. The effects of such locking differentials on performance and traction can thus only be controlled by the braking torque profile, which depends on the difference in rotational speed.

There are many designs and principles of operation for locking differentials and self-locking differentials. Examples of some of these designs are given in Section 12.6. The main types currently in use are:

- load-dependent self-locking differential with multiplate clutches,
- load-dependent self-locking differential with worm gears,
- slip-dependent self-locking differential with fluid clutch,
- slip-dependent self-locking differential with hydrostatic locking forces,
- electronically controlled (automatic) locking differentials with pressurised multiplate clutches,
- cam self-locking differentials,
- load-dependent self-locking differential with conical friction plates.

6.10.4 Alternatives to Self-Locking Differentials

Self-locking differentials can always only represent a compromise between improving traction and directional control on the one hand, and disadvantages in terms of steering responsiveness and possible distortions of the power train on the other hand. The main purpose in developing further systems will therefore be to only lock a differential when absolutely necessary.

For this purpose Volvo uses a differential that has a similar design to a self-locking differential with multiplate clutches. In this case however there is neither preload nor load-dependent pressure. Instead, a centrifugal governor presses against the disc sets when a predetermined speed difference is exceeded, monitoring the speed difference of the two wheels. Above 40 km/h another centrifugal governor again serves to open the interlock. This gives two differential settings: 100% locked or completely open. In the interest of predictability for the driver the lock must not be activated at high speeds; for this reason the Volvo system should be considered merely as a traction control system for moving off.

A completely different but effective and economical approach is used by VW with its so-called electronic differential lock, which despite its name does not act on the differential. The system uses the signals from the wheel sensors of the antilock braking system to detect drive wheel slip on one side. When a wheel starts to spin it is braked by the service brake, forcing more torque to the other wheel through the standard differential. If both wheels spin, the electronic differential lock does not intervene. This system promises a distinct improvement in traction when moving off and cornering without impairing directional control, especially on surfaces with marked variations in friction at different wheels. A similar system is installed in the Mercedes M-Class.

If the traction potential of all drive wheels is exceeded despite 100% locking, the only further measure that can be taken, apart from braking the spinning wheels is to reduce the drive torque (engine torque), and possibly shift into a higher gear. These additional possibilities are taken into account in electronic traction control systems. The driver can no longer influence the slip of the drive wheels with the accelerator pedal. This means a significant safety improvement, since it prevents vehicle instability caused by spinning drive wheels.

7 Design of Gearwheel Transmissions for Vehicles

Gearwheel calculation: Global standard – much empiricism and some theory

The declared aim of this book is to present a complete picture of the development process for vehicle transmissions. Chapters 3 to 5 showed how the ratios are selected – the fundamental design decision. In Chapter 6 some basic design concepts were introduced. Chapters 7 to 11 consider the design and construction of important components.

This does not involve the use of sophisticated calculations such as the German standard DIN 3990 gearwheel calculation, but an attempt is made to present the fundamentals of calculation methodology and calculation procedures. The aim is to equip the design engineer to quickly design important gear components "by hand". Such an approach is required for example for feasibility studies, where a quick draft is needed. For this purpose "flow charts" are displayed at suitable points throughout the following chapters, with algorithms for manual calculation.

By far the greatest proportion of vehicle transmissions are gearwheel transmissions. They still deliver the highest power-to-weight ratio for converting speed and torque. The transmission flow is normally between parallel shafts, using spur-toothed and helical-cut spur gears. The questions of *centre distance*, *transmission mass* (largely determined by *face width*), *service life* and *noise* have already been examined in Section 2.4 "Fundamental Performance Features of Vehicle Transmissions". Formulae are given below relating to these points, including measures for reducing transmission noise.

7.1 Gearwheel Performance Limits

The starting point for gearwheel design calculations is their performance limits, i.e. causes of failure. The performance limit of a gear pair is basically determined by four different types of damage:

- tooth failure,
- pitting,
- gear scuffing (hot scuffing),
- wear limit (cold scuffing).

These limits determine the load capacity of the gearwheels (Figure 7.1). The major factors affecting the performance limits indicated above are:

- operating conditions (type of load, tooth forces and additional forces, circumferential speed, temperature),
- selection of materials,
- gear geometry,
- manufacturing accuracy,
- surface treatment and surface roughness,
- selection of lubricant (chemical and physical characteristics).

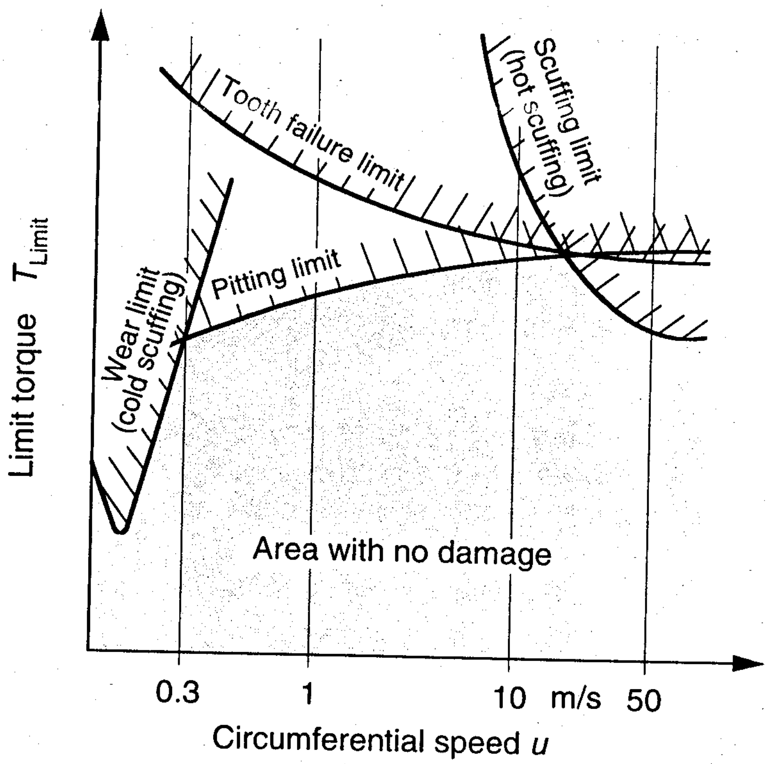


Figure 7.1. Bearing capacity limits (maximum permitted torque) of gearwheels

7.1.1 Causes and Types of Damage

Tooth Failure

Tooth failure is where the whole tooth or part of a tooth breaks off. A distinction is made between overload failure and vibration fatigue failure (fatigue fracture). Overload failure is the result of a brief, drastic overload of the gear pair as shown in Figure 7.2.

A gearwheel is normally subject to pulsating load. Idler gears are an exception, being exposed to alternating load, although at a low level. The maximum bending stress occurs at the tooth root. If the level of stress is frequently or occasionally in excess of the vibrational resistance of the gearwheel, this can lead to vibration fatigue failure or fatigue failure.

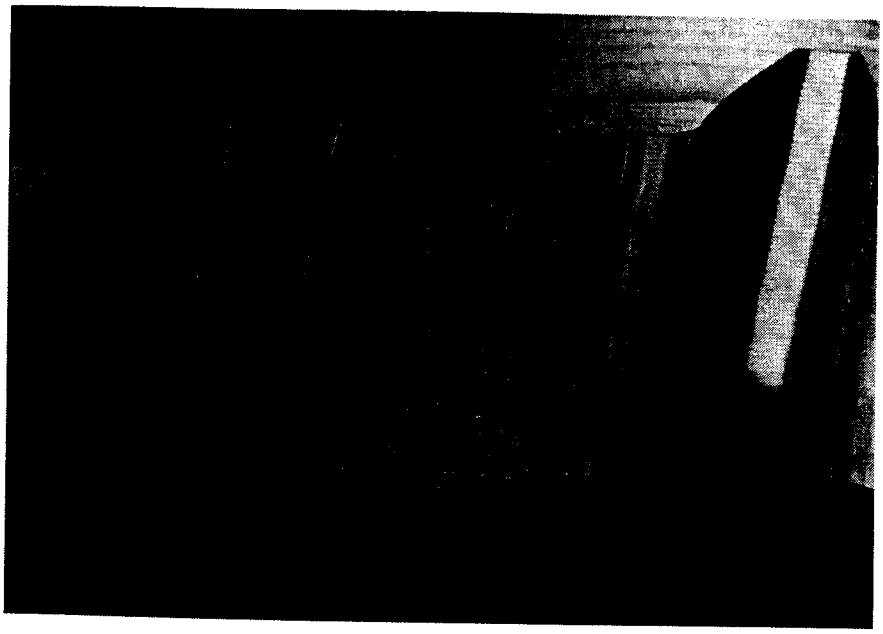


Figure 7.2. Overload failure of a helical cut spur gear [7.1]

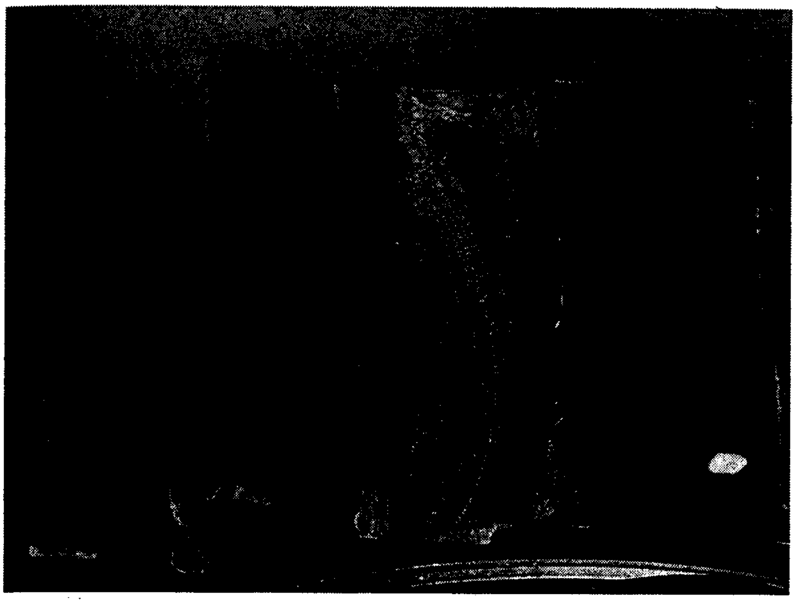


Figure 7.3. Vibration fatigue failure of a straight spur gear [7.1]

The vibrational resistance of the gearwheel is to a large extent determined by the tooth root design, surface roughness, surface bonding in the tooth fillet, and heat treatment. Figure 7.3 shows vibration fatigue failure in a straight spur gear.

Pitting

Damage to the gearwheel by pitting is indicated by the appearance of pin holes and extended flank spalling, mostly below the pitch circle. It is a symptom of material fatigue at the tooth flanks. Depending on the assumption made, the causes can be surface cracks resulting from slip, or shear stresses in the area below the tooth flank surface.

Hertzian stress is used to assess pitting load capacity, and is the basis for calculating surface stress. It is an important characteristic value for tooth flank meshing stress. But it is no more the sole cause of pitting than is the corresponding shear stress occurring below the surface [7.2].

Pitting only occurs in lubricated transmissions. Resistance to pitting is affected by hardness, oil viscosity, oil temperature, specific sliding, flank profile defects, surface roughness and circumferential speed. An area of spalling with pittings of different sizes is shown in Figure 7.4.

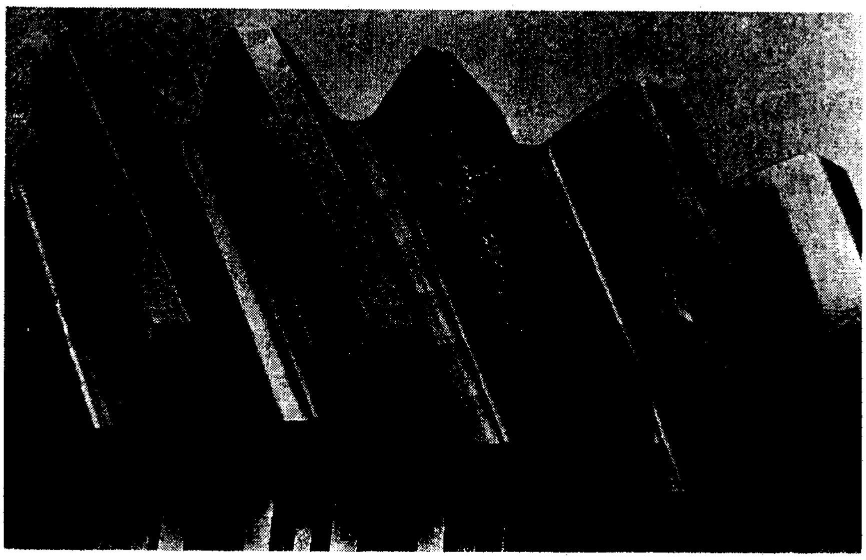


Figure 7.4. Pitting with tip fracture due to flank surface fatigue [7.1]

Gear Scuffing

Two different types of failure can occur when tooth flank lubrication fails, depending on the circumferential speed – cold scuffing and hot scuffing. Cold scuffing occurs mostly at low circumferential speeds below 5 m/s on heat-treated gearwheels of coarse toothing quality. It is caused purely by wear, and seldom occurs in vehicle transmissions.

Hot scuffing arises when the lubricant film breaks down because of high temperatures or excessive stress. This leads to metal-to-metal contact, local welding, and flaking of the tooth flanks. This gives rise to damage such as that shown in Figure 7.5. This is due to both physical and especially chemical processes. The physical phenomena are explained by elasto-hydrodynamic lubrication theory. The chemical processes occur in extremely thin layers and under high pressures, and are very complex [7.3]. A distinction must be made between two different types of lubricant film in gear lubrication: *elasto-hydrodynamic lubricant film* and *chemical protective film* resulting from the products of chemical reaction of the gearwheel material and additives. On this subject see also Section 11.2 "Lubrication of Gearboxes, Gearbox Lubricants".

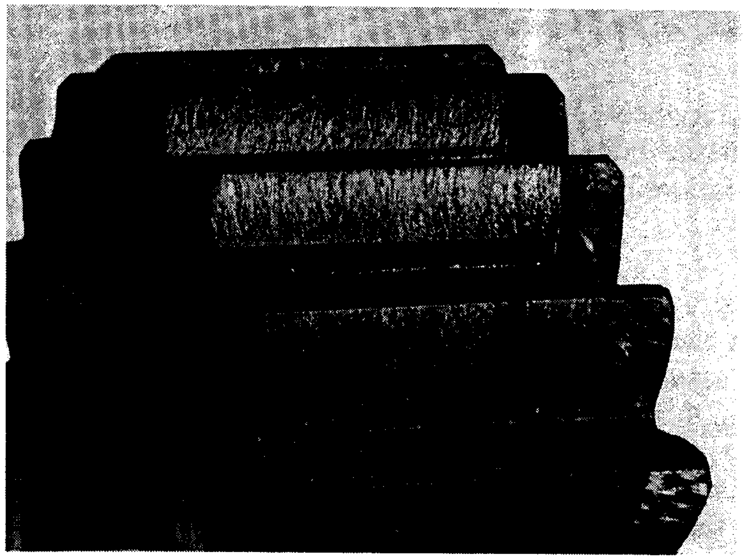


Figure 7.5. Scuffing across the whole contact pattern of a straight spur gear [7.1]

Two types of scuffing can be distinguished:

- *Scoring*
  Individual scoring or clusters of scoring appear in the sliding direction of the tooth flanks, varying from minor to serious. Typical of doped oils and circumferential speeds < 30 m/s.
- *Scuffing*
  This occurs as individual fine lines (scuffing lines), as clusters (heavy scuffing) or as areas across the full face width (scuffing zones). The main feature of the scored areas is a matt appearance. Typical for undoped and doped oils at circumferential speeds > 30 m/s.

The scuffing process is critically affected by the gearwheels heating up, the critical temperature being the "tooth flank constant temperature", that is the temperature which the tooth flank is constantly exposed to even when not engaged. Gearwheels frequently run in the mixed friction range, but the proportion of hydrodynamic lubrication along the contact path is high. With worn gearwheels and circumferential speeds > 4 m/s it is more than 60% even with high stress levels, and often even 80–95% [7.4] (see also Figure 11.3).

The scuffing process is started by a breakdown of the protective chemical film on the tooth flank. The strength of this protective film depends on the temperature of the tooth flank. Stressing of the protective chemical film is determined by Hertzian stress.

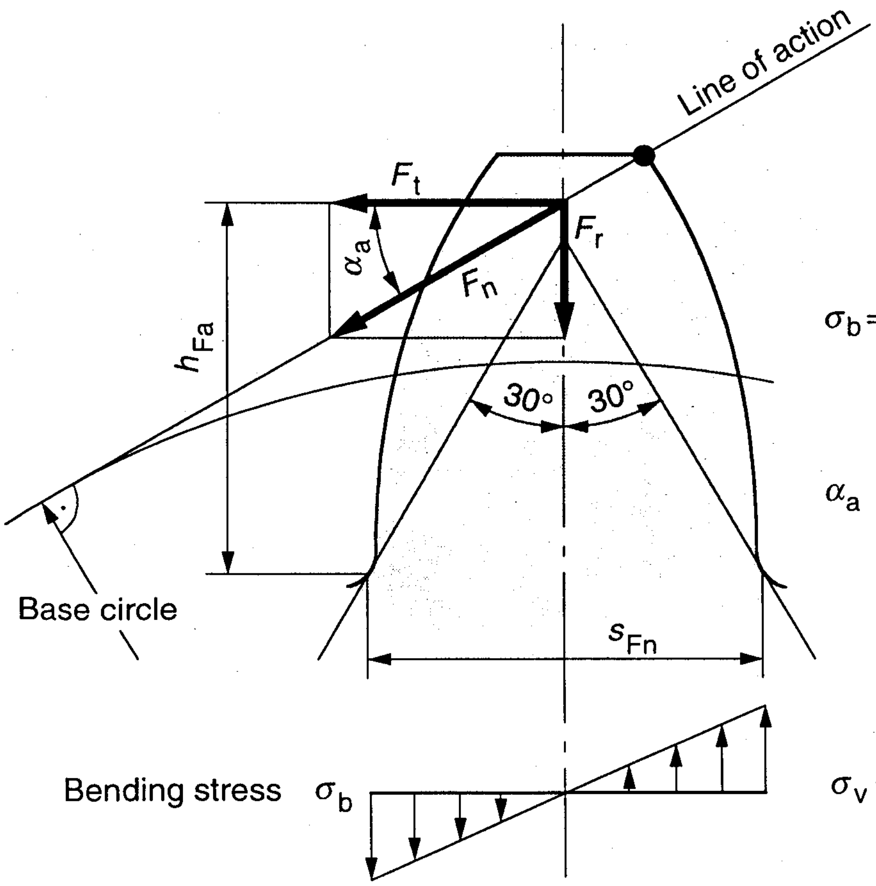
A thicker film of lubricant produced by higher lubricant viscosity can prevent scuffing. EP (extreme pressure) additives in oil to improve bearing capacity are of particular significance.

Gearwheels for vehicle transmissions are now almost without exception designed so that the "pitting" performance limit is critical. The design loads are now well established, so module and face width can be precisely selected so as to eliminate tooth failure, which is particularly serious since it causes immediately transmission failure. Gear scuffing is prevented by using a suitably doped gear oil.

Gearwheels are in principle case hardened. Exceptions are ring gears and planetary gear-sets in planetary gear units. Some of these are carbonitrided. The gearwheels are generally shaved and honed, for price reasons; ground gearwheels are used in low-noise gear units. Corrections in the form of profile bearing and transverse crowning are now state of the art.

The outline calculations for tooth failure, pitting and scuffing to German standard DIN 3990 [7.2] are set out below. A procedure for approximating centre distance and face width is then presented in Sections 7.2 and 7.3, based on the pitting load capacity calculation.

A component or assembly is generally calculated in three steps. The first step is to carry out an initial calculation determining all the principal dimensions. Correctly selecting the safety factor is of great significance here. When the dimensions have been determined, the strength for a particular type of load can be calculated. The final step is an operational integrity calculation taking into account the actual load profile encountered.



$$\sigma_b = \frac{F_t \; 6 \; h_{Fa} \cos \alpha_a}{b \; s_{Fn}^2 \cos \alpha} \qquad (7.1)$$

α_a Angle of application of force relative to tip edge

$\sigma_v \approx \sigma_b$

Figure 7.6. Bending stress at the tooth root with force acting on the tip

7.1.2 Calculating the "Tooth Failure" Performance Limit

To calculate the "tooth failure" performance limit, the tooth root load capacity must be checked. The tooth is most at risk when the perpendicular force F_n along the line of action (with its components F_r and F_t) acts at the tooth tip (Figure 7.6) (see also Figure 8.7 "Forces acting on the tooth flanks"). The impinging forces cause compression stress, bending stress and shear stress in the tooth. It has been shown that bending stress is the only stress that is critical for calculation purposes ($\sigma_V \approx \sigma_b$). The tooth cross-section to which the bending stress relates is the product of the face width b and the root thickness chord s_{Fn}. The root thickness chord s_{Fn} is determined by two tangents at the tooth root fillet at an angle of less than 30°. The existing root bending stress σ_F is determined in German standard DIN 3990 [7.2] from a nominal value multiplied by various parameters

$$\sigma_F = \frac{F_t}{b\, m_n} Y_{Fa}\, Y_{Sa}\, Y_\varepsilon\, Y_\beta\, K_A\, K_V\, K_{F\beta}\, K_{F\alpha}\ . \tag{7.2}$$

The terms in Equation (7.2) are as follows:

- F_t nominal circumferential force at the pitch circle in N,
- b face width in mm,
- m_n standard module in mm,
- Y_{Fa} form factor to DIN 3990, Part 3, Page 13,
- Y_{Sa} stress correction value (notch configuration value) DIN 3990, Part 3, Page 2,
- Y_ε contact ratio to DIN 3990, Part 3, Page 38,
- Y_β helical overlap to DIN 3990, Part 3, Page 39,
- K_A application factor,
- K_V dynamic load factor to DIN 3990, Part 1, Page 16, 17,
- $K_{F\beta}$ longitudinal factor to DIN 3990, Part 1, Page 19,
- K_{Fa} transverse factor to DIN 3990, Part 1, Page 45.

The tooth root strength σ_{FG} is determined in accordance with German standard DIN 3990 as

$$\sigma_{FG} = \sigma_{F,\,lim}\, Y_{ST}\, Y_{NT}\, Y_{\delta,\,relT}\, Y_X\ , \tag{7.3}$$

where:

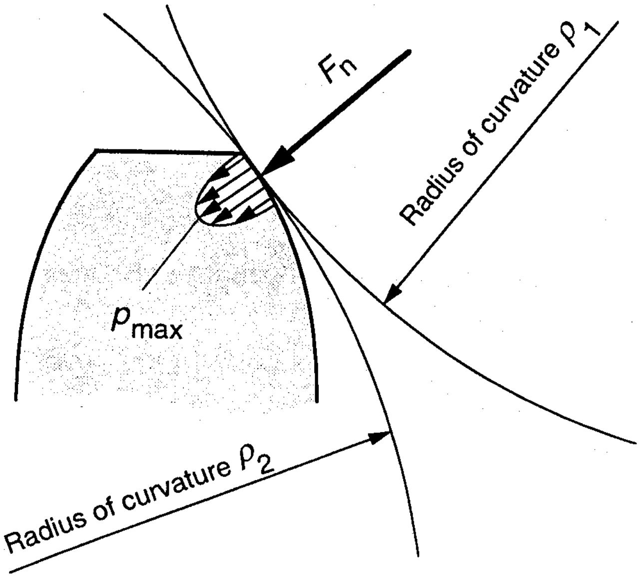
- $\sigma_{F,\,lim}$ fatigue strength value to DIN 3990, Part 5, Pages 4–10,
- Y_{ST} stress correction factor to DIN 3990, Part 3, Page 4,
- Y_{NT} service life factor to DIN 3990, Part 3, Page 40,
- $Y_{\delta,\,relT}$ relative support figure (= f(Y_{Sa})) to DIN 3990, Part 3, Page 44,
- Y_X tooth root size factor to DIN 3990, Part 3, Page 50.

The existing root bending stress σ_F is compared with the tooth root strength σ_{FG}. The quotient of tooth root strength and existing root bending stress forms the safety factor S_F

$$S_F = \frac{\sigma_{FG}}{\sigma_F}\ . \tag{7.4}$$

7.1.3 Calculating the "Pitting" Performance Limit

To calculate the "pitting" performance limit, the tooth flank load capacity must be checked. This is based on the equations developed by Hertz for the compression of two cylindrical rollers (Figure 7.7).



Hertzian stress:

$$p_{max} = \sqrt{\frac{F_n \; E}{5.72 \; \rho \; b}} \; , \qquad (7.5)$$

where $E_1 = E_2 = E$ or

$$E = \frac{2 \; E_1 \; E_2}{E_1 + E_2} \quad \text{and}$$

substitute radius of curvature

$$\rho = \frac{\rho_1 \; \rho_2}{\rho_1 + \rho_2} \; .$$

Figure 7.7. Stresses at the tooth flank

If two rollers in contact along their common contour lines are subjected to the normal force F_n, they undergo flattening at the contact line. The distribution of contact pressure is unequal, peaking in the centre of the flattened surface. The Hertzian equation of roller pressing applies only for purely elastic deformation with rollers at rest. Equation 7.5 gives only an approximation of the actual compression relations at the tooth.

Resistance to pitting is derived in accordance with German standard DIN 3990 [7.2] as the quotient of the tolerable surface stress σ_{HG} and the existing Hertzian stress σ_H. Both values are in turn derived from a nominal value and the corresponding parameters

$$\sigma_H = Z_{B/D} \; Z_H \; Z_E \; Z_\varepsilon \; Z_\beta \sqrt{\frac{F_t \; (u+1)}{d_1 \; b \; u}} \sqrt{K_A \; K_V \; K_{H\beta} \; K_{H\alpha}} \; , \qquad (7.6)$$

where the terms in Equation 7.6 have the following meanings:

$Z_{B/D}$	pinion contact factor Z_B, wheel contact factor Z_D DIN 3390, Part 2, Page 8,
Z_H	zone factor to DIN 3990, Part 2, Page 6,
Z_E	elasticity factor to DIN 3990, Part 2, Page 8,
Z_ε	contact ratio to DIN 3990, Part 2, Page 8,
Z_β	helical overlap to DIN 3990, Part 2, Page 10,
F_t	nominal circumferential force in N/mm^2,
b	contact face width in mm,
d_1	pitch circle diameter of the pinion in mm,
u	gear ratio z_2/z_1; $\mid z_2/z_1 \mid \geq 1$,
K_A	application factor to DIN 3990, Part 1, Page 55,
K_V	dynamic factor to DIN 3990, Part 1, Pages 16–17,
$K_{H\beta}$	longitudinal load distribution factor for Hertzian stress to DIN 3990, Part 1, Page 19,
$K_{H\alpha}$	transverse factor to DIN 3990, Part 1, Page 45.

The pitting boundary strength σ_{HG} is derived from

$$\sigma_{HG} = \sigma_{H,\,lim} \; Z_{NT} \; Z_L \; Z_R \; Z_V \; Z_W \; Z_X \; , \qquad (7.7)$$

where the expressions in Equation 7.7 have the following meanings:

$\sigma_{H,\,lim}$	fatigue strength value to DIN 3990, Part 5, Pages 4–9,
Z_{NT}	service life factor to DIN 3990, Part 2, Pages 11–12,
Z_L	lubricant factor to DIN 3990, Part 2, Page 13,
Z_R	roughness factor to DIN 3990, Part 2, Page 15,
Z_V	velocity factor to DIN 3990, Part 2, Page 14,
Z_W	material mating factor to DIN 3990, Part 2, Page 16,
Z_X	size factor for surface stress to DIN 3990, Part 2, Page 1.

The numerical safety factor for surface stress (against pitting) is determined accordingly from Equations 7.6 and 7.7 as

$$S_H = \frac{\sigma_{HG}}{\sigma_H} \,. \tag{7.8}$$

7.1.4 Calculating the "Gear Scuffing" Performance Limit

Based on the hypothesis that the lubricant film is broken down by high surface temperatures caused by high stresses and high rubbing speeds, two methods of calculation are proposed in German standard DIN 3990:

- *The Integral Temperature Method*
  This gives a weighted average of surface temperature along the contact path.
- *The Flash Temperature Method*
  This describes varying contact temperature along the contact path.

The calculation is specified in German standard DIN 3990.

7.2 Estimating Centre Distance

Centre distance is the crucial parameter in automotive transmissions. It is important to obtain an approximation of this value at the start when designing gearwheel transmissions.

Pitting is the critical performance constraint. In order to derive an equation for calculating the centre distance a, it is therefore necessary to start with Hertzian stress at the pitch circle σ_H, Equation 7.6 (German standard DIN 3990)

$$\sigma_H = Z_{B/D}\,\sigma_{H0}\sqrt{K_A\,K_V\,K_{H\beta}\,K_{H\alpha}} \,. \tag{7.9}$$

With the nominal contact pressure σ_{H0}

$$\sigma_{H0} = Z_H\,Z_E\,Z_\varepsilon\,Z_\beta\sqrt{\frac{F_t\,(u+1)}{d_1\,b\,u}} \,, \tag{7.10}$$

the torque to be transmitted at the pinion shaft T_1

$$T_1 = \frac{F_t\,d_1}{2} \tag{7.11}$$

and the face width diameter relationship b/d_1 the following results:

$$\sigma_H = Z_{B/D}\, Z_H\, Z_E\, Z_\varepsilon\, Z_\beta \sqrt{\frac{2\, T_1\,(u+1)}{d_1^3\, \dfrac{b}{d_1}\, u}} \sqrt{K_A\, K_V\, K_{H\beta}\, K_{H\alpha}} \;. \tag{7.12}$$

If the diameter d_1 in Equation 7.12 is replaced by

$$d_1 = \frac{2\,a}{1+u}\;, \tag{7.13}$$

the surface stress σ_H is replaced by the permissible stress $\sigma_{H,\,\mathrm{perm}}$

$$\sigma_{H,\,\mathrm{perm}} = \frac{\sigma_{H,\,\mathrm{lim}}\, Z_{NT}\, Z_L\, Z_R\, Z_V\, Z_W\, Z_X}{S_H}\;. \tag{7.14}$$

Now this can be solved for the centre distance a, giving

$$a = \sqrt[3]{\frac{T_1\,(u+1)^4}{4\,\dfrac{b}{d_1}\,u}}\; \sqrt[3]{\frac{\left(Z_{B/D}\, Z_H\, Z_E\, Z_\varepsilon\, Z_\beta\, S_H\right)^2}{\left(\sigma_{H,\,\mathrm{lim}}\, Z_{NT}\, Z_L\, Z_R\, Z_V\, Z_W\, Z_X\right)^2}}\; \sqrt[3]{K_A\, K_V\, K_{H\beta}\, K_{H\alpha}}\;. \tag{7.15}$$

The estimate is made *for the gear with the highest torque multiplication*. For the face width diameter ratio b/d_1 an established practical value is to be used. In order to minimise uneven face wear, a different ratio is selected for each gear. An evaluation of existing vehicle transmissions in respect of the ratio b/d_1 (average) is shown in Figure 7.8.

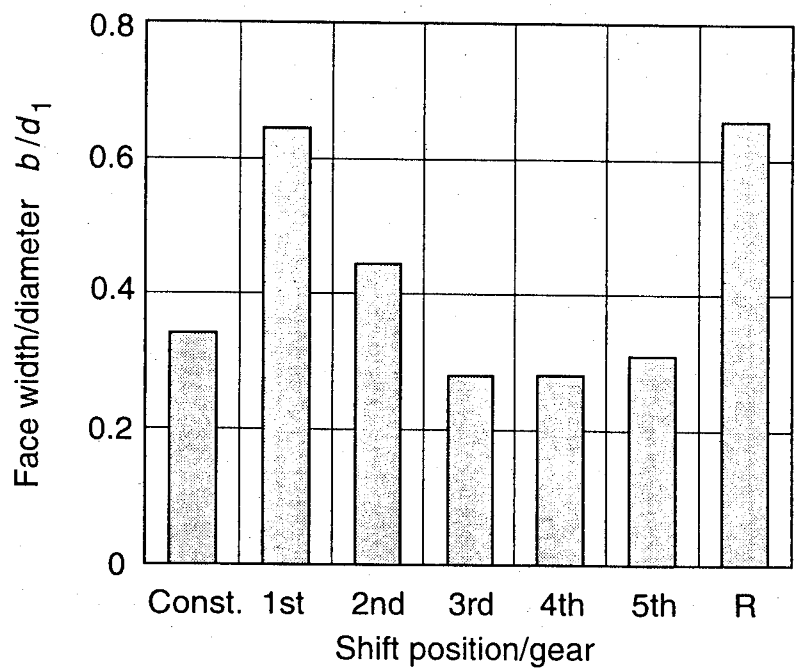


Figure 7.8. Face width diameter relationship b/d_1 of existing passenger car and commercial vehicle transmissions

For the individual factors in Equation 7.15, the following values should be used to arrive at an estimate:

b/d_1 = 0.65,
K_A = 0.65 for passenger cars to DIN 3990, Part 41, Page 28,
K_A = 0.85 for trucks to DIN 3990, Part 41, Page 28,
K_V, $K_{H\beta}$, $K_{H\alpha} = 1$,
Z_H = 2.25 (for $a_n \approx 20°$, $\beta \approx 15°$, $(x_1+x_2)/(z_1+z_2) \approx 0.015$),
$Z_{B/D}$ = 1,
Z_E = $\sqrt{0.175\,E} = 189.8\sqrt{\text{N/mm}^2}$ for steel/steel,
Z_ε = 0.95,
Z_β = 0.95,
Z_{NT}, Z_L, Z_R, Z_V, Z_W, $Z_X = 1$,
$\sigma_{H,\,lim}$ = 1800 N/mm² DIN 3990, Part 41, Page 29 (material 16 MnCr5),
S_H = 1.2.

Calculating the centre distance with *first gear* engaged, this gives the following *approximation equation* (centre distance a in mm; torque T_1 in Nmm)

$$a = K_a \sqrt[3]{\frac{T_1\,(u+1)^4}{u}}\,, \tag{7.16}$$

K_a = 0.255 for passenger cars,
K_a = 0.278 for trucks,
T_1 torque at the shaft on which the pinion of the first gear is mounted. E.g. for a two-stage countershaft transmission $T_1 = i_{CG}\,T_G$, where i_{CG} is a constant ratio: T_G transmission input torque,
u gear ratio of the first gear pair, $|u| \geq 1$.

Figure 7.9 shows a comparison of the theoretical centre distance calculated using Equation 7.16, with the centre distance of actual transmissions.

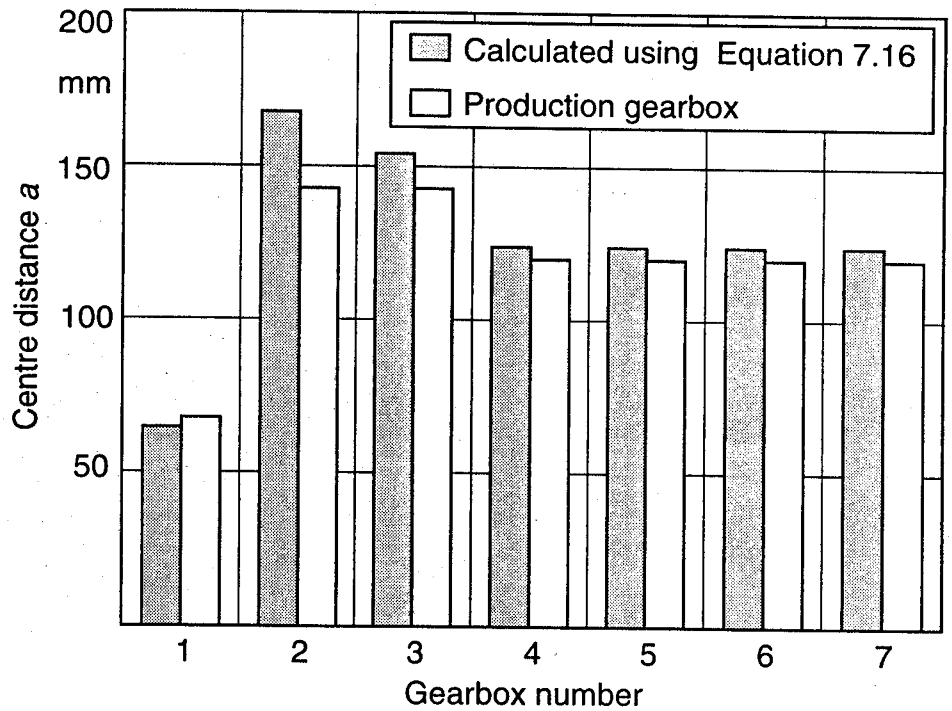


Figure 7.9. Comparison of theoretical and actual gearbox centre distances

7.3 Estimating Face Widths

Having estimated the centre distance it is possible to derive the pinion diameter d_1 from Equation 7.13, and given the transmission ratio, the wheel diameter d_2. The face width $b_{1,\,1st}$ of the pinion *of first gear* is calculated from the preselected face width/diameter ratio of 0.65 as

$$b_{1,\,1st} = 0.65\, d_{1,\,1st}\ . \tag{7.17}$$

For the remaining gears $n = 2, ..., z$ the required face width $b_{1,\,n}$ of the pinion is derived by equating Equation 7.12 with Equation 7.14:

$$b_{1,\,n} = \left(Z_{B/D} Z_H Z_E Z_\varepsilon Z_\beta\right)^2 \frac{2\,T_1\,(u_n+1)\,S_H^2\,K_A\,K_V\,K_{H\beta}\,K_{H\alpha}}{d_{1,\,n}^2\,u_n\,\sigma_{H,\,lim}^2\,\left(Z_{NT}\,Z_L\,Z_R\,Z_V\,Z_W\,Z_X\right)^2}\ . \tag{7.18}$$

With the factors determined as shown in Section 7.2 for the estimate, it is possible to arrive at the following simplified equation (T_1 in Nmm; $d_{1,\,n}$ in mm; $b_{1,\,n}$ in mm)

$$b_{1,\,n} = 427800\,K_A\,\frac{T_1\,(u_n+1)}{d_{1,\,n}^2\,u_n\,\sigma_{H,\,lim}^2}\ , \tag{7.19}$$

assuming the fatigue strength value $\sigma_{H,\,lim}$ as shown in Table 7.1, and using $K_A = 0.65$ (passenger car) or $K_A = 0.85$ (truck) as the application factor.

Table 7.1. Low cycle fatigue life $\sigma_{H,\,lim}$, face width constant K_b

Gear	1st gear	2nd gear		Other gears/constants	
$\sigma_{H,\,lim}$ (N/mm²)	1800	1600		1500	
Face width constant K_b		Passenger car	Truck	Passenger car	Truck
	–	0.108622	0.142044	0.123588	0.161614

The pinion face widths $b_{1,\,n}$ for the *second and subsequent gears* can be calculated from Equation 7.19 and the values in Table 7.1. using the following *unit equation* (T_1 in Nmm; $d_{1,\,n}$ in mm; $b_{1,\,n}$ in mm)

$$b_{1,\,n} = K_b\,\frac{T_1\,(u_n+1)}{d_{1,\,n}^2\,u_n} \quad \text{for the gear pairs of gears } n = 2 \text{ to } z\ . \tag{7.20}$$

K_b from Table 7.1,
T_1 torque at the shaft on which the pinion is mounted;
e.g. in the case of a two-stage coaxial countershaft transmission:
case 1: constant: $T_1 = T_G$;
case 2: pinion on countershaft: $T_1 = i_{CG}\,T_G$;
case 3: pinion on main shaft ($i_{gear} < 1$): $T_1 = i_{CG}\,i_P\,T_G$;
i_{CG} constant gear; i_P gear pair; T_G transmission input torque,
u_n gear ratio of the gear pair of the n-th gear, $|u| \geq 1$.

7.4 Operational Integrity and Service Life

Reliability, durability and weight all have to be taken into account when developing a modern vehicle transmission.

The performance of a component under loads that change over time is summarised in the term "vibrational resistance" [7.5]. Reliability is historically regarded as an element of vibrational resistance. Stress profiles are stochastic and completely irregular. The aim of the *operational integrity* design concept is to design components reliably for typical operational stresses for a particular service life. In aeronautical and automotive engineering this design approach is state of the art. It is aimed at adequate component reliability with low weight and low costs.

The service life of components is determined predominantly by failure due to wear and fatigue (see also Figure 16.5 "Bathtub curve"). The topics of service life and reliability are extremely closely interrelated, so see also Chapter 16 "Reliability and Testing of Vehicle Transmissions".

Since the different gears in a transmission are engaged in the power flow for very different proportions of time (Figure 7.10), and are subjected to different stresses, individual gears are only designed for a finite life.

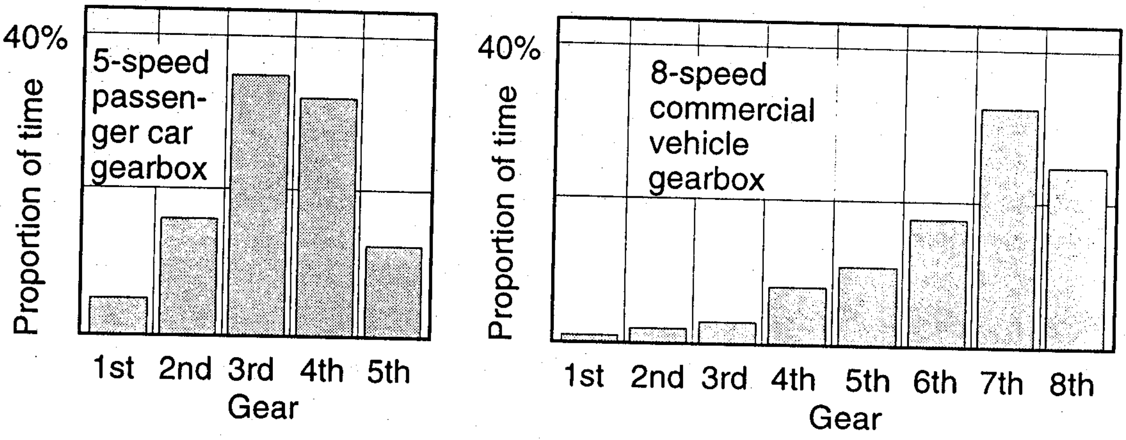


Figure 7.10. Proportion of time the various gears of a 5-speed passenger car transmission and an 8-speed commercial vehicle transmission are engaged when travelling on a mountainous highway (example)

To minimise protracted and costly test rig and road trials, the service life of individual components should preferably be determined by calculation. The information required for this purpose is shown in Figure 7.11. But not all transmission components are amenable to service life calculation, or rather *service life estimation.* (See also Section 16.2.2 "Qualitative Reliability Analysis"). No service life calculation is possible for the socalled "B" components in the "A, B, C" analysis. It is necessary to rely on empirical values and results of trials. B components include synchronisers and seals (see Figure 16.16). "A" components such as gearwheels, shafts, bearings, etc., can be subject to service life calculation. For low-risk "C" components such as circlips and screw plugs, service life calculations are neither possible nor necessary.

There are basically two different ways of determining the service life of transmission components based on a stress profile: either by numerical calculation or on a test bed (Figure 7.12). Numerical service life calculation using enumeration and damage accumulation hypotheses must be verified at least on a sample basis by service life measurement on the test bed. This is done by simulating the load profile under computer control on a transmission test bed. The results are used as the basis for determining relative service life.

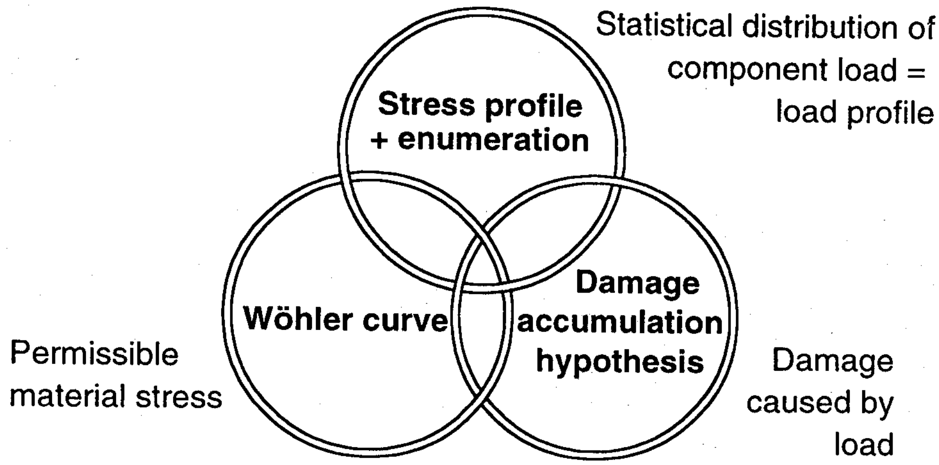


Figure 7.11. Components required to calculate operational integrity

7.4.1 The Wöhler Curve

The Wöhler curve provides data on the resilience of a component. Wöhler curves should ideally be derived from real components. Cost and time constraints normally dictate the use of test pieces subjected to alternating compression – tension tests, with constant stress amplitude σ_i until failure (single-stage test). If the stress amplitudes are plotted against the tolerable number of oscillations N_i, this results in the typical curve which characterises the fatigue strength for finite life and fatigue resistance ranges (Figure 7.13).

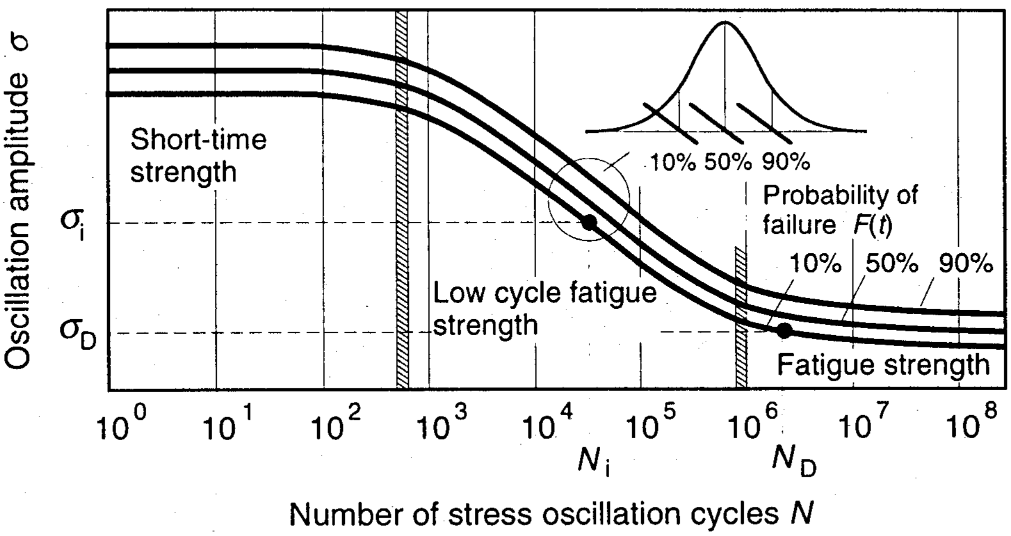


Figure 7.13. Wöhler curve of permissible material stress

The resultant numbers of stress cycles before failure are random values, i.e. they are distributed around an average. The most common curves are Wöhler curves for 10% failure probability. But 1% and 50% curves are also frequently encountered, where determining the 1% Wöhler curve is not straightforward. The B_{10} service life of a component can be estimated with the Wöhler curve for 10% failure probability. That is the service life at which on average 10% of this component has already failed in operation. The B_{10} service life only describes a point of the time-dependent failure behaviour of a component.

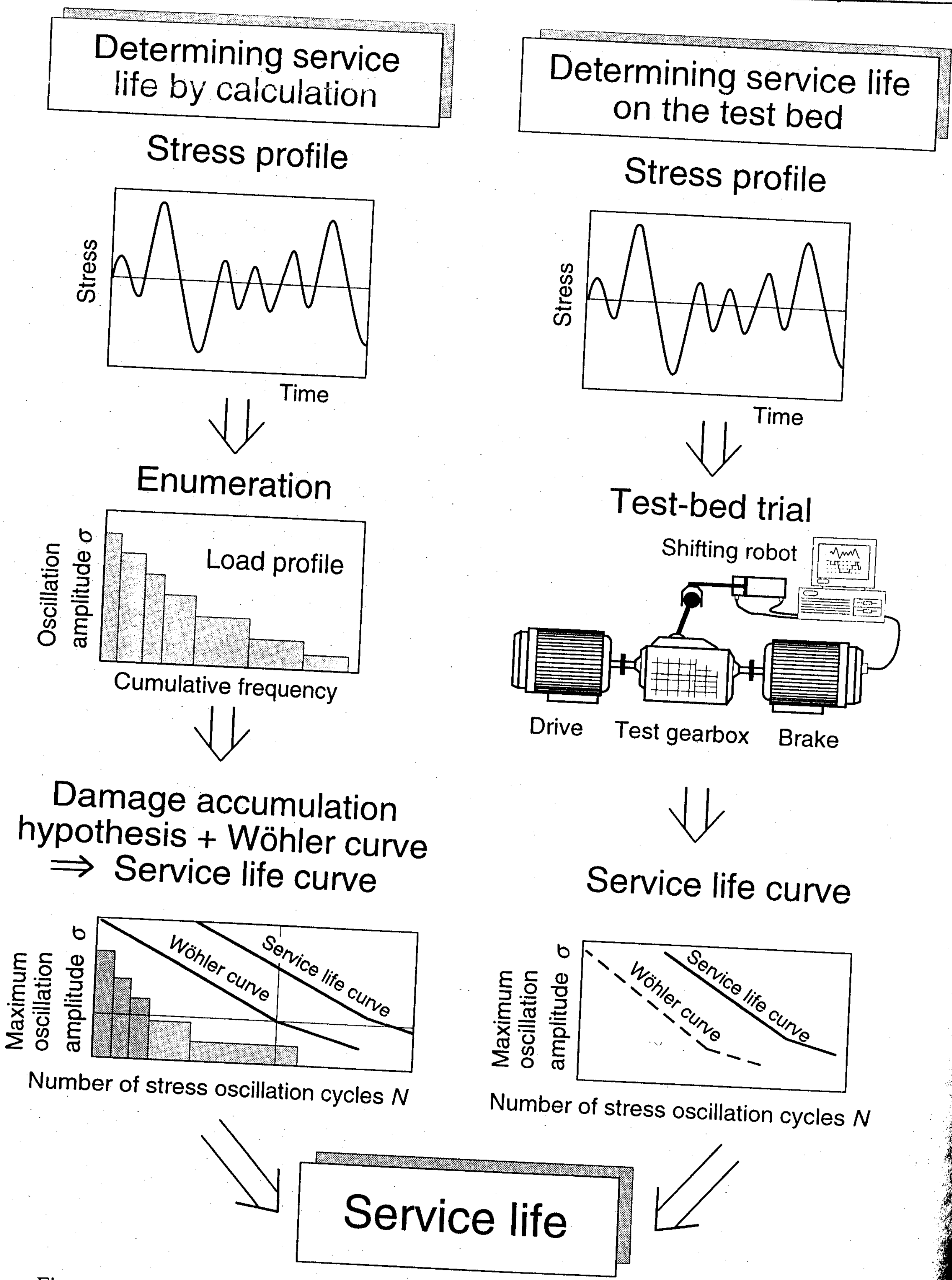


Figure 7.12. Various ways of determining component service life. In contrast to the Wöhler curve, which is determined by the single-stage test, the service life curve is based on a load profile. The service life curve can be determined experimentally in operational integrity tests by simulating the random stress pattern [7.6]. It can also be derived from the Wöhler curve using a damage accumulation hypothesis

The low cycle fatigue zone of the Wöhler curve can be described as a straight line in the log log co-ordinate system with the following Equation

$$N_\mathrm{i} = N_\mathrm{D}\left(\frac{\sigma_\mathrm{i}}{\sigma_\mathrm{D}}\right)^{-k} . \tag{7.21}$$

The exponent k determines the gradient for the low cycle fatigue zone. The exponent k for gearwheels depends on the failure under consideration and the surface hardening process, and has values ranging from $k = 4$ to $k = 16$ (Figure 7.14). As the exponent rises, the characteristic graph becomes flatter. Even small differences in stress have a great influence on service life, which leads to a wide range of component service life in practical use.

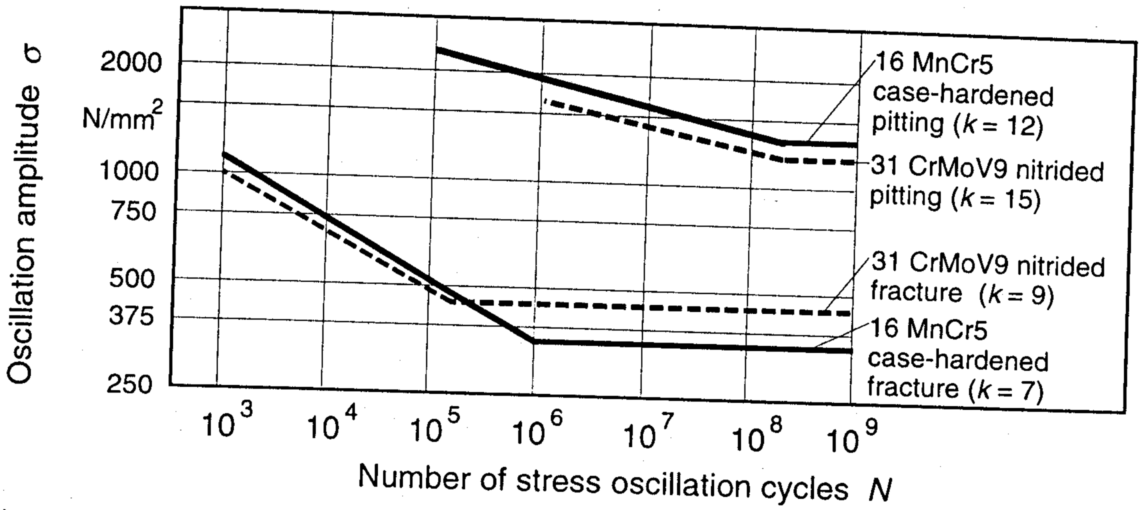


Figure 7.14. Wöhler curves (10% probability of failure) for various materials and failures (examples)

7.4.2 Load Profile and Enumeration

In addition to the data on resilience of transmission parts, the anticipated operational stresses must also be evaluated mathematically to determine service life. (Definition: *stress* within a component, e.g. tension σ; *loading* from outside on the component, e.g. torque T.)

To determine operational loadings, a load/time or a load/distance chart is needed. The loadings can be determined by

1/ road trials on defined stretches of road,
2/ numerical driving simulation.

To be able to process the resultant load profiles, the volume of data has to be reduced. Since the principal factors involved in calculating service life are the size and frequency of loads, the sequence and frequency of the occurrence of loads are ignored. Data reduction on this basis is achieved by means of enumeration leading to a load profile.

In practice, use is made of both single-parameter counting procedures in which only amplitudes are counted, and also two-parameter counting procedures in which pairs of values are counted. An example of a single-parameter counting procedure is shown in Figure 7.15, the class continuity procedure to German standard DIN 45667, whereby the whole load amplitude range is divided into classes, and every instance of class boundaries being exceeded in a positive direction is counted and totalled. This evaluation gives an overview of the maxima and minima measured, but not the average values and amplitudes of individual oscillations.

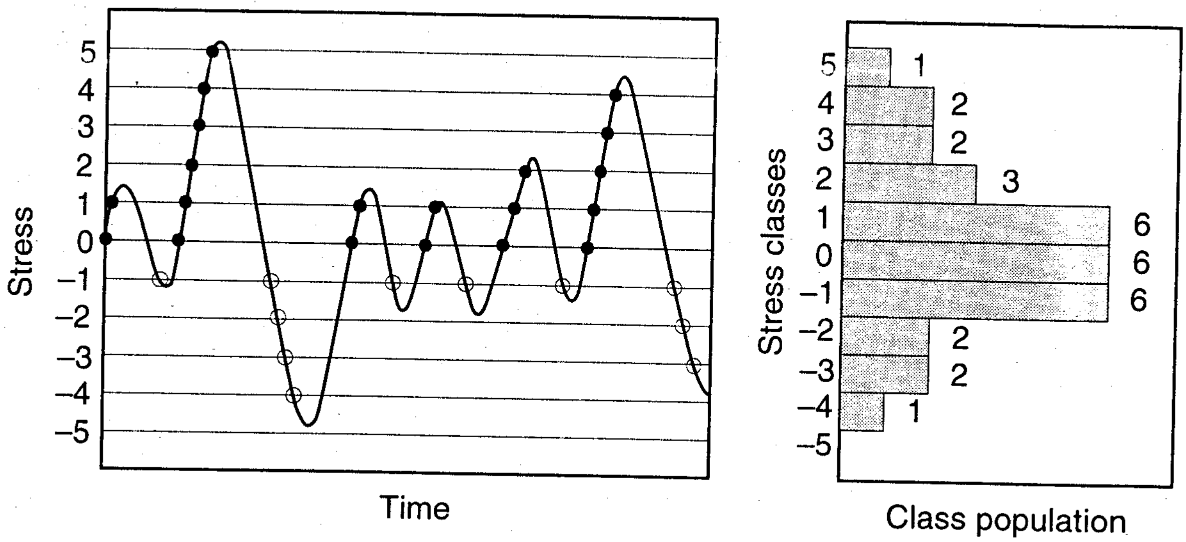


Figure 7.15. Example of single-parameter enumeration: class continuity procedure to German standard DIN 45667

The actual damage to a component can be better described by a load profile, using a two-parameter counting procedure, e.g. evaluation using the Rainflow enumeration [7.7]. The results of this enumeration are stored in a matrix, allowing conclusions to be drawn as to amplitudes of individual oscillations and also their average values.

1/ Road Trials

The resources required for direct measurement of components is reduced by measuring the load function at just one point in the power flow. The values are then transferred by computation to the remaining components. This method is frequently used to measure torque and speed at the transmission input, and at the gear engaged.

The test routes are characterised by their height profiles, their gradient and road speed distributions, and the level of stress involved in road trials provides an "accelerated testing effect". Customer surveys are also used to decide the types of road and loadings used in these trials. The influencing variables impinging on a transmission load profile can be divided into three groups.

1. Vehicle: Engine performance map, loading, transmission ratio.

2. Driver: Driving style, characterised by frequency of shifting, gearshift engine speed and accelerating and braking behaviour.

3. Road type: Distribution of total kilometres over motorways, highways, trunk roads, urban and local traffic, mountain roads, typical gradients and maximum speeds (see also Table 2.9).

2/ Determining Load Profiles by Numerical Driving Simulation

Since transmission stress depends on many different stochastic variables, substantial resources are required for statistically validated road trials. Road tests have to be repeated whenever design changes are made to the power train. One way of limiting the number of costly road trials is numerical driving simulation (see also Chapter 15 "Computer-Aided Transmission Development, Driving Simulation"). Figure 7.16 shows the load profile for third gear in a mid-range passenger car on a mountainous highway as an example of numerical simulation.

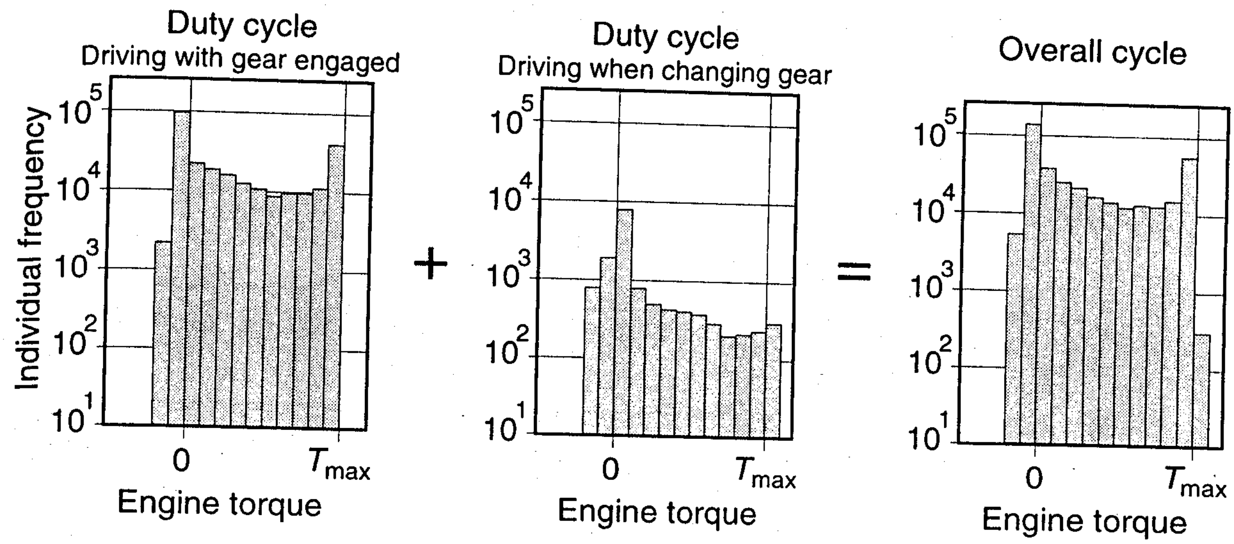


Figure 7.16. Load profile for 3rd gear of a mid-range passenger car on a mountainous highway

7.4.3 Damage Accumulation Hypothesis

The damage accumulation hypothesis establishes the relationship between a load profile and the damage to a component caused by the load profile, taking into account the resilience described by the Wöhler curve.

Dynamic stresses above a certain level produce an effect in the material which is generally called "damage". MINER [7.8] therefore proceeds on the assumption that a component absorbs work during the fatigue process, and regards the relation between actual work absorbed and the maximum possible work absorbed as a measure of actual damage. Thus the ratio of the number of load cycles n to the ultimate number of cycles N is equal to the ratio of actual work absorbed w to the maximum possible work absorbed W, and is called partial damage

$$\frac{w}{W} = \frac{n}{N} \,. \tag{7.22}$$

Assuming equal absorbable energy of fracture W at all stress cycles occurring above the fatigue limit allows partial damage by stress cycles of variable size to be summed. The non-quantifiable work variable w is replaced by the number of stress reversals n. The individual frequencies n_i of oscillation amplitudes σ_i classified by enumeration into j classes, are related to the tolerable number of load cycles N_i at this amplitude, given by the Wöhler curve (Figure 7.17). The resultant elements of partial damage are summarised as total damage.

The boundary condition of admissible strain arises when the work absorbed equals the maximum possible work absorbed, i.e. when the *damage total* equals 1, the component fails

$$\frac{w_1}{W_1} + \frac{w_2}{W_2} + \ldots + \frac{w_j}{W_j} = \frac{n_1}{N_1} + \frac{n_2}{N_2} + \ldots + \frac{n_j}{N_j} = \sum_{i=1}^{j} \frac{n_i}{N_i} = 1 \,. \tag{7.23}$$

This relatively simple method of calculation is preferable for determining the service life of anti-friction bearings.

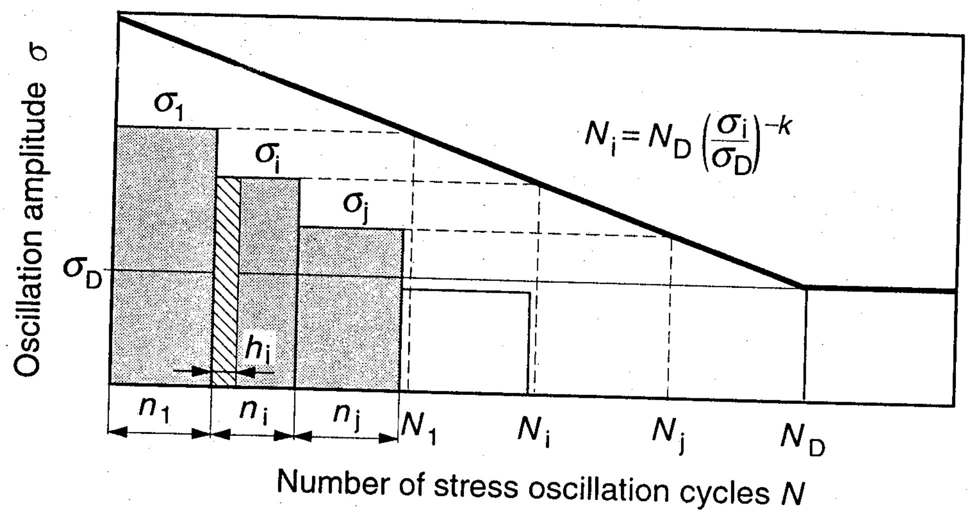


Figure 7.17. The Palmgren-Miner damage accumulation hypothesis

Using the PALMGREN-MINER method of calculating service life [7.8], [7.9], the service life in load cycles N is derived from the total of load cycles n_i in the various classes. To expand the load profile h_i into a larger duty cycle, simply multiply by the number of passes z to represent passing z times along the test track on which the load cycle is determined, whereby Equation 7.23 becomes:

$$\sum_{i=1}^{j} n_i = N \ , \tag{7.24}$$

$$z \sum_{i=1}^{j} h_i = N \ . \tag{7.25}$$

The acceptable number of vibration cycles N_i allocated to the individual frequencies n_i is determined using the Wöhler curve equation (7.21). Substituting Equation 7.21 and Equation 7.25 in Equation 7.23 gives:

$$1 = \sum_{i=1}^{j} \frac{z\, h_i}{N_D \left(\frac{\sigma_i}{\sigma_D} \right)^{-k}} \ . \tag{7.26}$$

Solving for z:

$$z = \frac{1}{\sum_{i=1}^{j} \frac{h_i}{N_D} \left(\frac{\sigma_i}{\sigma_D} \right)^{+k}} \ , \tag{7.27}$$

where z is the number of possible load cycles corresponding to the load profile used, which the component withstands before failure. See also the following sample calculation [7.10]. By substituting Equation 7.27 in Equation 7.25, the component service life in load cycles is finally derived

$$N = N_D \frac{\sum_{i=1}^{j} h_i}{\sum_{i=1}^{j} h_i \left(\frac{\sigma_i}{\sigma_D}\right)^{+k}} \,. \tag{7.28}$$

The MINER-HAIBACH Damage Accumulation Hypothesis

It has been shown in practice that even stresses below the fatigue limit must not be ignored where there is previous partial damage. The MINER-HAIBACH damage accumulation hypothesis assumes that stresses in the fatigue strength range also cause damage. The assumption is made that the Wöhler curve with the gradient exponent k continues with a smaller gradient $(2k-1)$ after falling below the fatigue strength amplitude σ_D (Figure 7.18).

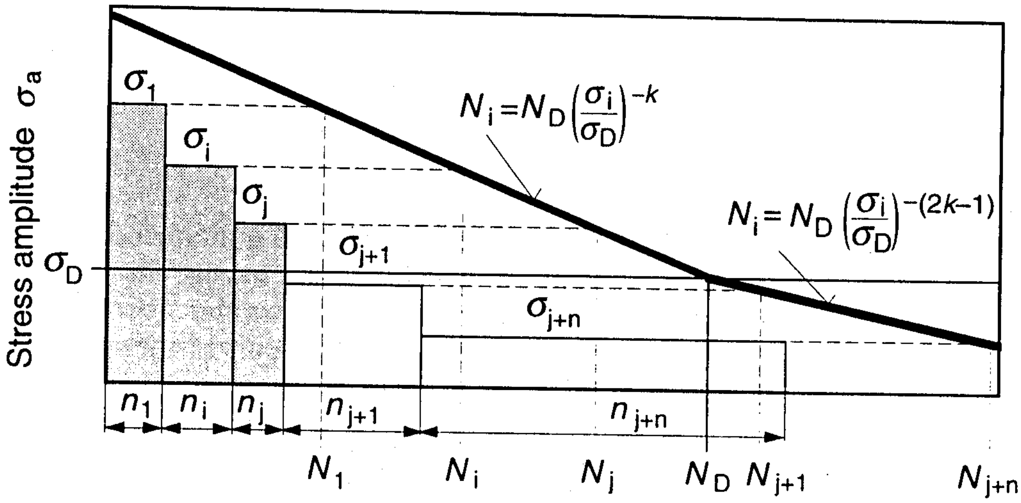


Figure 7.18. The MINER-HAIBACH damage accumulation hypothesis. Wöhler curve modified after HAIBACH

For stress amplitudes above the fatigue limit $\sigma_i \geq \sigma_D$, Equation 7.21 applies where $i = 1, ..., j$; below the fatigue limit $\sigma_i < \sigma_D$ the following applies:

$$N_i = N_D \left(\frac{\sigma_i}{\sigma_D}\right)^{-(2k-1)} \qquad \text{for } i = j+1, ..., j+n \,. \tag{7.29}$$

The MINER-HAIBACH service life calculation is carried out as above, but taking into account the stresses below the fatigue limit

$$N = N_D \frac{\sum_{i=1}^{j+n} h_i}{\sum_{i=1}^{j} h_i \left(\frac{\sigma_i}{\sigma_D}\right)^{+k} + \sum_{i=j+1}^{j+n} h_i \left(\frac{\sigma_i}{\sigma_D}\right)^{+(2k-1)}} \,. \tag{7.30}$$

Relative Service Life Prediction

The practice of predicting service life by calculation is frequently unreliable. This is partly due to a marked dispersion of the damage total on failure. Prediction can normally be made more accurate by combining computational methods with experimental analyses.

When designing components in automotive and aeronautical engineering the *Relative Miner Rule* is frequently used. The service life L is calculated from the damage total belonging to a load cycle, as follows

$$L = \frac{1}{Damage\ total\ (Cycle)}\ Cycle\ (\text{distance or time})\ . \tag{7.31}$$

The result for the relative Miner rule is thus

$$\frac{L_{\text{calculated}}}{L_{\text{actual}}} = \frac{\text{Actual damage total}}{\text{Calculated damage total}} = \frac{\left[\sum_{i=1}^{j} \frac{n_i}{N_i}\right]_{\text{actual}}}{\left[\sum_{i=1}^{j} \frac{n_i}{N_i}\right]_{\text{calculated}}}\ . \tag{7.32}$$

In the Relative Miner Rule, practical experiments matched to anticipated use are used to determine the damage total at the end of service life. This means that experience and analogies of the previous model or of a transmission with similar operating conditions can be used. Erroneous assumptions relating to the Wöhler curves then have much less effect on the result than when determining service life purely by calculation. Uncertainties nevertheless remain as to how far the load profile on which the calculation was based corresponds to the load profile in the trial.

Whereas the relative Miner rule makes use of serviceability tests, the *Modified Relative Miner Rule* makes use of operational experience [7.5]. This involves determining by how much the actual service life achieved in operation L_{actual} has deviated from the numerically predicted service life $L_{\text{calculated}}$. The ratio then represents the factor by which the service life predicted by calculation for a similar component (a new design, for example) must be corrected (Figure 7.19).

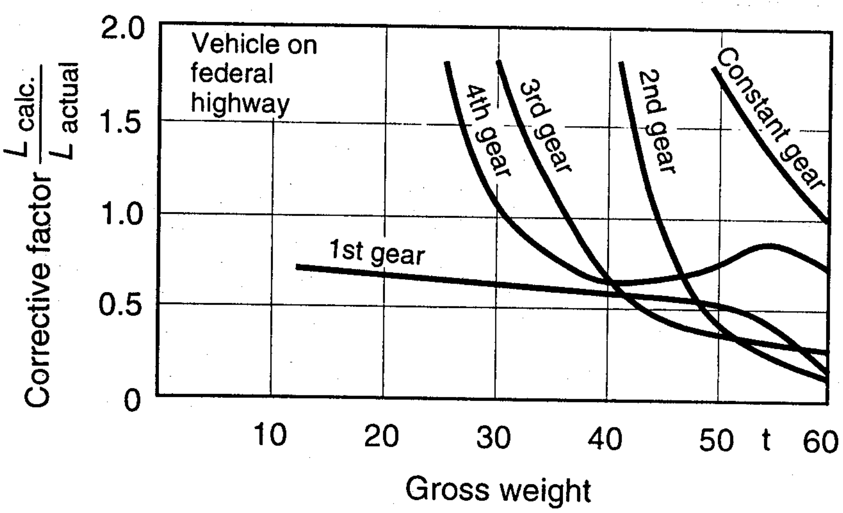


Figure 7.19. Assessment of the relative service life of four gears of a commercial vehicle transmission with different payloads

Example of Service Life Calculation for a Gear Pair in a Vehicle Transmission

The record of the transmission input torque $T_G = f(t)$ is available for a 6-speed commercial vehicle transmission (Figure 7.20), from which the number of revolutions U_G in the various classes of the transmission input torque can be determined (see Table 7.2). On the 1 km reference route, the 5th gear under observation was only used once. It accounted for 21.5% of the distance travelled.

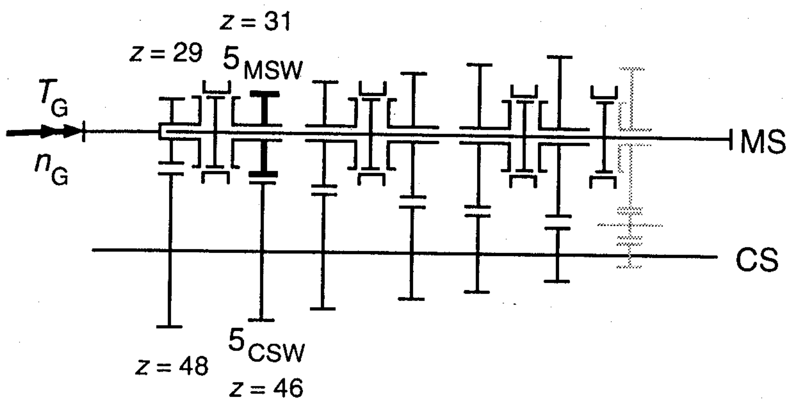


Figure 7.20. Diagrammatic view of a two-stage 6-speed countershaft type commercial vehicle transmission.
MS Main shaft
CS Countershaft
MSW Main shaft wheel
CSW Countershaft wheel

1 The wheel duty cycle of the pinion of 5th gear is calculated from the gearbox input duty cycle, ignoring the dynamic behaviour of the transmission.

1.1 Converting the transmission input duty cycle into the load profile of the wheel 5_{MSW}. Gear ratios: constant ratio: $u_{CG} = \frac{48}{29}$, gear pair 5: $u_5 = \frac{46}{31}$.
This gives the torque at the countershaft: $T_{CS} = u_{CG}\, T_G$.

1.2 For a reference torque $T_{ref} = 1400$ Nm at the countershaft, the contact pressure at gear pair 5 has been calculated for the driving flank in accordance with German standard DIN 3990: $\sigma_{H,\,ref} = 978$ N/mm². Conversion to any required torque:

$$\sigma_H \sim \sqrt{T} \qquad \text{and thus} \qquad \sigma_H = \sigma_{H,\,ref} \sqrt{\frac{T_{CS}}{T_{ref}}} \,.$$

1.3 Converting the transmission input speeds into revolutions of the wheel 5_{MSW}, i.e. into load cycles

$$U_{5,\,MSW} = \frac{U_G\, u_5}{u_{CG}} \,.$$

Table 7.2. Initial values (shaded grey) and results of the calculation

Class i	T_G (Nm)	T_{CS} (Nm)	σ_H (N/mm²)	U_G	$U_{5,\,MSW}$	N_i Fig. 7.21	$\frac{U_{5,\,MSW}}{N_i}$
4	1500 – 2000	3310	1504	4	3.6	$6.3 \cdot 10^6$	$5.71 \cdot 10^{-7}$
3	1000 – 1500	2482	1302	21	18.8	$3.2 \cdot 10^7$	$5.88 \cdot 10^{-7}$
2	500 – 1000	1655	1063	251	225.0	$3.3 \cdot 10^8$	$6.82 \cdot 10^{-7}$
1	0 – 500	828	752	67	60.1	$1.0 \cdot 10^{12}$	$6.60 \cdot 10^{-11}$

2 The service life of the driving flank of pinion 5_{MSW} of 5th gear as regards pitting is calculated using the PALMGREN-MINER damage accumulation hypothesis, Equation (7.23)

$$\sum_{i=1}^{j} \frac{n_i}{N_i} = \sum_{i=1}^{j} \frac{z\, U_{5,\,\mathrm{MSWi}}}{N_i} = 1 \ ,$$

whereby

$$z = \frac{1}{\sum_{i=1}^{j} \frac{U_{5,\,\mathrm{MSWi}}}{N_i}} = \frac{1}{\mathit{Damage\ total\ (Cycle)}} \ .$$

2.1 The acceptable load cycles are derived from the Wöhler diagram modified after HAIBACH, Figure 7.21, see Table 7.2.

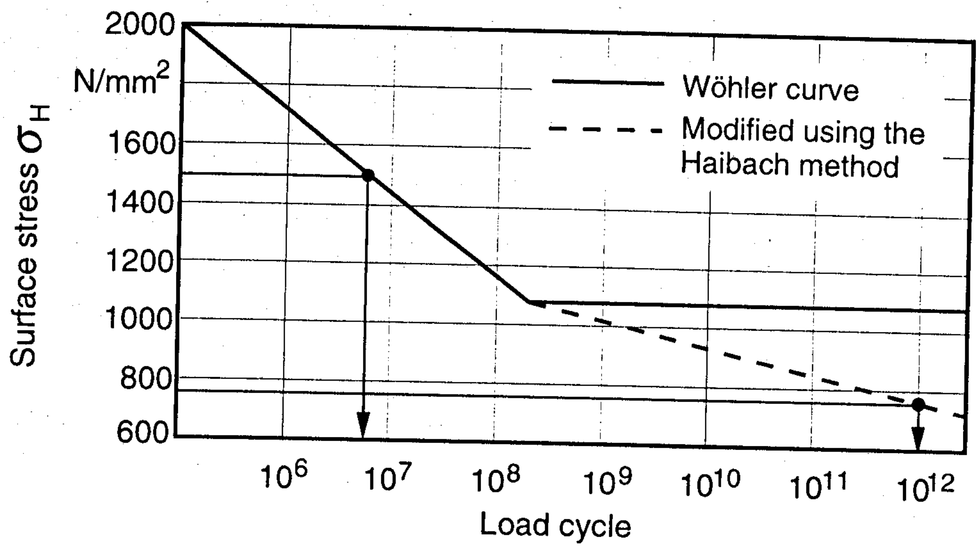


Figure 7.21. Wöhler curve for pitting, 10% probability of failure. (Ordinate simplified, not logarithmic)

2.2 Calculating the number of load cycles by solving the PALMGREN-MINER equation. The number of load cycles is derived from the damage factors calculated in Table 7.2 as:

$$z = \frac{1}{5.71\cdot 10^{-7} + 5.88\cdot 10^{-7} + 6.82\cdot 10^{-7} + 6.60\cdot 10^{-11}} = 543240 \ .$$

2.3 The mileage life of 5th gear is derived from Equation (7.31)

$$L_{\text{5th gear}} = z \ s_{\text{ref}} \ \mathit{Proportion}_{\text{ 5th gear}} \ .$$

For a reference route s_{ref} of 1 km and a proportion of 21.5% in 5th gear, the resultant mileage life is $L_{\text{5th gear}}$ = 116 800 km. The mileage lives of the gear pairs needed in the reference route must be known to determine the service life of the transmission. The transmission mileage life is then equal to the shortest mileage life.

7.5 Developing Low-Noise Transmissions

Noise reduction is an important objective when developing vehicles. Vehicle noise not only impairs the wellbeing and health of driver and passengers, but also subjects the population at large to traffic noise. Together with the engine, bodywork and running gear, a vehicle's transmission is a major source of noise. Table 7.3 shows the increasingly strict legal limits on noise emissions for various types of motor vehicle.

Table 7.3. European vehicle noise emissions limits in dB(A) for accelerated passage to ISO 362 [7.11]

Type of vehicle		Noise emission limits in dB(A)			
		1970	1977	1994	1995/96
Passenger cars		82	80	77	74
Buses	< 150 kW	89	82	80	78
	> 150 kW	91	85	83	80
Buses,	< 2 t	84	81	78	76
Trucks	> 2 t ≤ 3.5 t	84	81	79	77
Trucks	> 3.5 t < 75 kW	89	86	81	77
	> 7.5 t ≤ 150 kW	89	86	83	78
	≥ 150 kW	91	88	84	80

As noise reduction measures relating to other sources within the vehicle have proved successful, so there has been increasing pressure to reduce transmission noise. It is generally not so much the absolute level of noise, as its particular character that distinguishes transmission noise from the other sources of noise in the vehicle. Some transmission noise phenomena do not in themselves constitute too great a source of noise pollution since they are only audible under certain operating conditions, like rattling and clattering. But they often lead to complaints, since the customer (wrongly) believes the vehicle to be defective. Non-specialists often mistake transmission noise for engine noise. When seeking to reduce transmission noise, it is not sufficient just to make improvements to the transmission itself. As with all vehicle noise problems, the bodywork and other components involved in transmitting and radiating sound have to be taken into account. See also Section 5.5 "Dynamic Behaviour of the Power Train, Comfort".

7.5.1 Transmission Noise and Its Causes

Sources of transmission noise can be divided into four categories (Table 7.4). The noise phenomena are considered below in the order of their significance as sources of noise in modern vehicle transmissions.

Table 7.4. Transmission noise and its causes

Transmission noise	Cause
1/ Howling/Squealing	Vibration of gearwheels under load: – Meshing impact – Parametrically excited vibration – Rolling contact noise
2/ Rattling/Clattering	Vibration of loose parts, caused by torsional vibration of the power train: – Idler gears – Synchroniser rings
3/ Engagement noise	Scraping and grating of the selector teeth when the synchromesh is not functioning properly
4/ Bearing noise	Running noise of the rolling bearings; especially when they are worn

1/ Howling/Squealing

The rolling contact noise of gear pairs under load – transmitting power – can be described as *howling, squealing, grinding and singing* [7.12]. This running noise has several causes:

- *Meshing impact*
  Meshing impact is the consequence of pitch errors or variations from the Law of Gears due to deformation of gears under load (Figure 7.22). The profile corrections made to avoid these meshing impacts are however only effective for a certain load range.

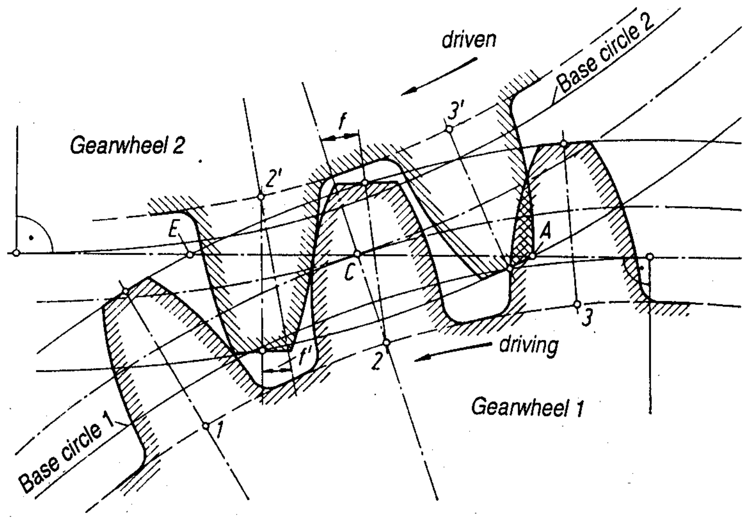


Figure 7.22. Meshing impact resulting from deformed teeth [7.12]

- *Parametrically excited vibration*
  Parametrically excited vibration arises from tooth rigidity changing with the meshing position (Figure 7.23). The amount of this vibration depends on gearing geometry and speed. If the excitation frequency (speed and number of teeth) is

close to a natural frequency of the gear pair, the resonance creates particularly large oscillation amplitudes, and hence particularly loud noises. These vibrations arise even with perfect gear teeth.

- *Rolling contact noise*
  Rolling contact noise due to the "washboard effect" arising from inadequate surface quality. Even where production tolerances are adhered to, noise can occur if vibration is generated by certain tooth flank surface structures arising in production.

Reverse gear in many vehicles still uses straight-cut gearwheels, which are not optimised in terms of noise generation; rolling contact noise is particularly noticeable in this case.

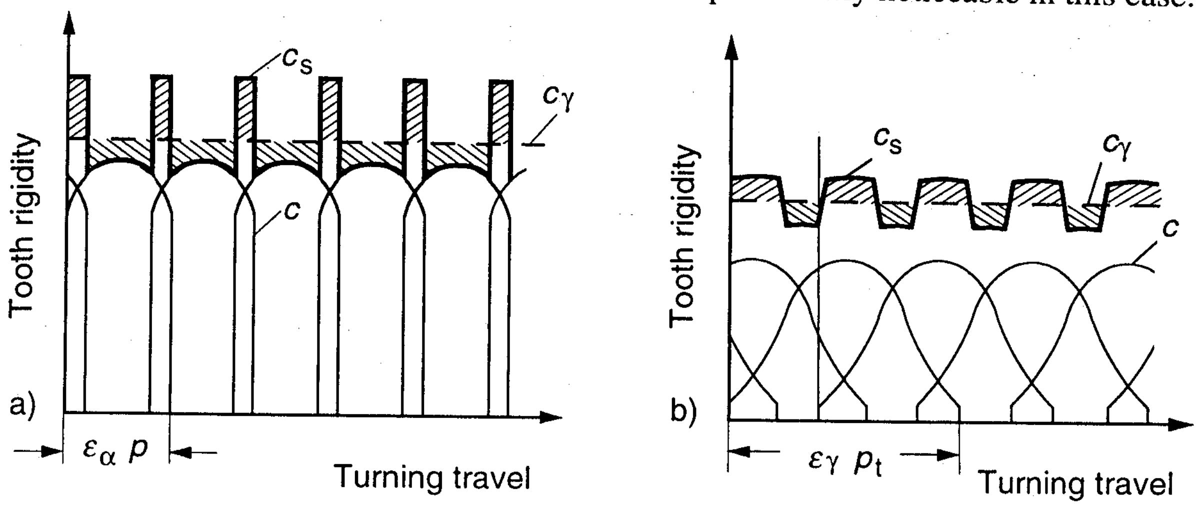


Figure 7.23. Overall tooth rigidity pattern c_s, meshing rigidity c_γ (average value of c_s over time). **a)** Spur gearing; **b)** Helical gearing [7.12]

2/ Rattling/Clattering

Gearwheels and gearshift components vibrating within permitted functional and production limits when not under load produce rattling and clattering noises [7.13–7.15], caused by torsional vibration in transmission shafts. If the amplitude of the torsional vibration exceeds a certain value, the idler gears that are not currently engaged lift away from their driving flank, and vibrate backwards and forwards within their tooth clearance. This amplitude depends on the moment of inertia of the idler gear and the drag torque, which has a delaying effect on this part. Torsional vibration can also be induced in synchroniser rings and sliding sleeves within their clearance. The real cause of this noise is the impacts when the loose parts encounter the clearance limits. If they occur in the transmission's neutral position, the phenomenon is called gear rattle at idling speed, and if they occur in motion, it is called traction or thrust rattle. Figure 7.24 shows the possible vibration elements based on the example of a 5-speed transmission.

Torsional vibration in the transmission shafts, which gives rise to these noises, is caused by irregularities in engine speed due to the finite number of cylinders. (See also Figure 5.14 "Internal combustion engine in neutral".) Almost all measures to improve fuel consumption and emissions standards in internal combustion engines cause irregular running, which is why noise from loose parts has been the subject of increasing attention recently. But they occur only under certain operating conditions.

Gear-rattle at idling speed is particularly pronounced with diesel engines. Gearboxes rattle at low speeds under power. Beyond a certain degree of irregularity, thrust rattle also occurs. The flanks of the gearwheels and the engaging gears of the current gear and of the axle gearbox can also lift with very small traction or thrust loads, causing rattling noise.

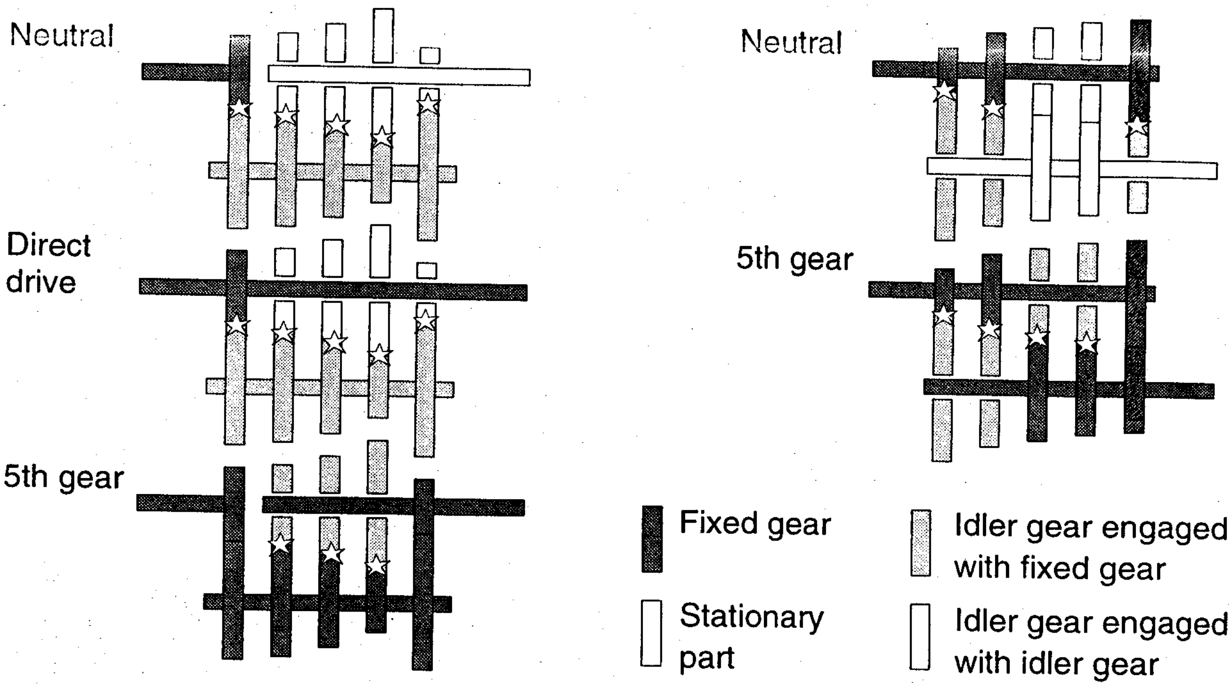


Figure 7.24. Sources of rattle and clatter in coaxial and non-coaxial gearboxes

A decisive factor in minimising rattle and clatter is power-train tuning [7.16–7.18], especially correct design of torsion dampers in the clutch plate, dual-mass flywheels and dampers, to keep the torsional vibration amplitudes of the power train within certain limits in all operating situations. In the case of automatic transmissions there are normally no large excitation amplitudes, since the torque converter damps the irregularity of the engine's rotation very well. But if the converter is locked up to save fuel, this can also cause rattling.

3/ Engagement Noise

If the synchroniser is not functioning correctly, engagement noise will be audible when shifting [7.19]. Torsional vibration, especially from torsional backlash in the power train, can cause the sliding sleeve to mesh in the engaging gears of the idler gear prematurely, before speeds have been synchronised. This then causes grating or scraping noises. Whether such noise arises depends on the driver's gear-changing technique. Engagement noise can be regarded as a purely aesthetic problem.

4/ Bearing Noise

Bearing noise is normally barely perceptible. Noise only arises with damaged rolling bearings, increasing rapidly as the damage increases. This noise phenomenon can be regarded as beneficial, since it brings a potentially critical failure to the driver's attention. Classical transmission noise – noise caused by gear pairs transmitting power – can be reduced with modern technology to the point that it can barely be heard inside the vehicle. The increased manufacturing cost associated with increased expectations is however a problem. Rattling and clattering are kept to a tolerable level by measures to unlink the torsional vibration of the engine from the transmission. But here again economic constraints are soon encountered.

7.5.2 How Noise Reaches the Ear

The main source of noise is the flanks of the gearwheels. The level and character of this noise is influenced in many different ways by the path it then takes to reach the ear. The airborne noise generated within the gearbox is very largely contained, and produces no significant level of noise outside. The vibration of the gearwheel bodies is transmitted to the shafts of the gearbox as structure-borne noise – directly in the case of fixed gears, and through the respective bearings in the case of idler gears. The input and output shafts transmit the vibrations outside the gearbox. But most of this vibration is transmitted to the housing through the shaft bearings. The bending vibrations of the shafts in particular excite the housing. If the excitation frequency is close to a natural frequency of the housing, the vibrations are further amplified, and the noise level is high. Part of the noise from the gearbox housing is radiated as airborne noise, and the rest is transmitted as structure-borne noise to the body via the gearbox suspension (Figure 7.25).

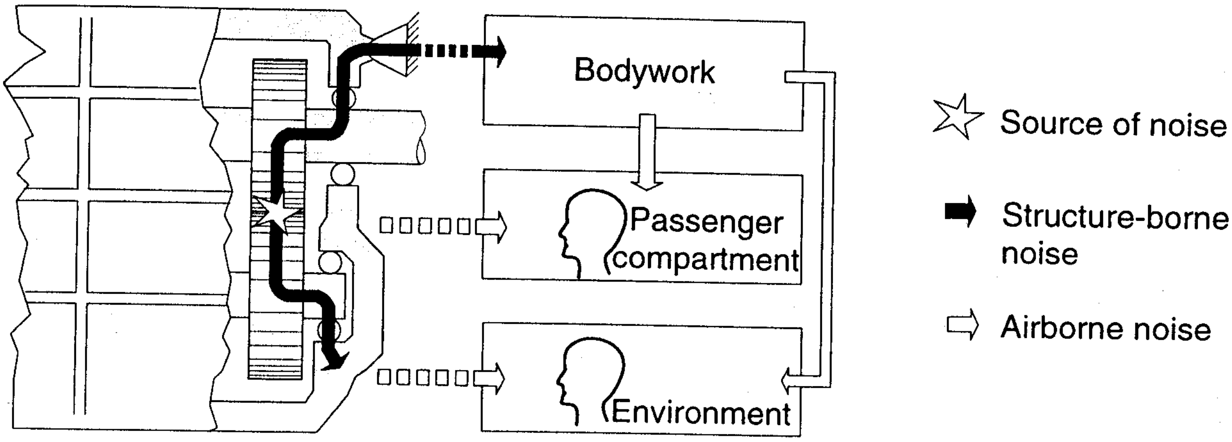


Figure 7.25. Passage of airborne and structure-borne noise from the gearbox

The noise radiated by the gearbox is further influenced by the insulating characteristics of the body, or by a special sound reduction shell. The dynamic behaviour of the gearbox suspension and the bodywork as a whole also have a significant effect on the intensity with which structure-borne noise is transmitted or damped.

When developing new transmissions or improving existing vehicles, not only the transmission itself has to be taken into account, but also the various routes the noise takes and the ways of controlling this.

7.5.3 Assessment Criteria

The subjective perception of this noise by the driver, passengers and people outside the vehicle is crucial in assessing transmission noise.

When a vehicle is under development, improvement measures are frequently evaluated by trained drivers subjectively rating particular noise phenomena [7.20]. One rating scale frequently used is the "ATZ Evaluation". These test drivers have to repeatedly compare their ratings to ensure reasonably uniform and consistent evaluation. Subjective assessment is however not sufficient to accurately compare small differences in noise level, particularly after a lapse of time. It is therefore necessary to measure noise emissions objectively in order to evaluate and accurately compare different transmissions at various stages of development. Noise measurement is also necessary to verify compliance with legal requirements.

Human hearing is sensitive not only to the energy level of the source, but also very much to the frequency distribution of the noise. The frequency sensitivity of hearing has been comprehensively studied for pure sounds (sinusoidal sound pressure profile over time). The "A evaluation filter" is based on the results of these investigations. The lower frequency range is given a lower weighting since hearing is less sensitive to lower frequencies (Figure 7.26).

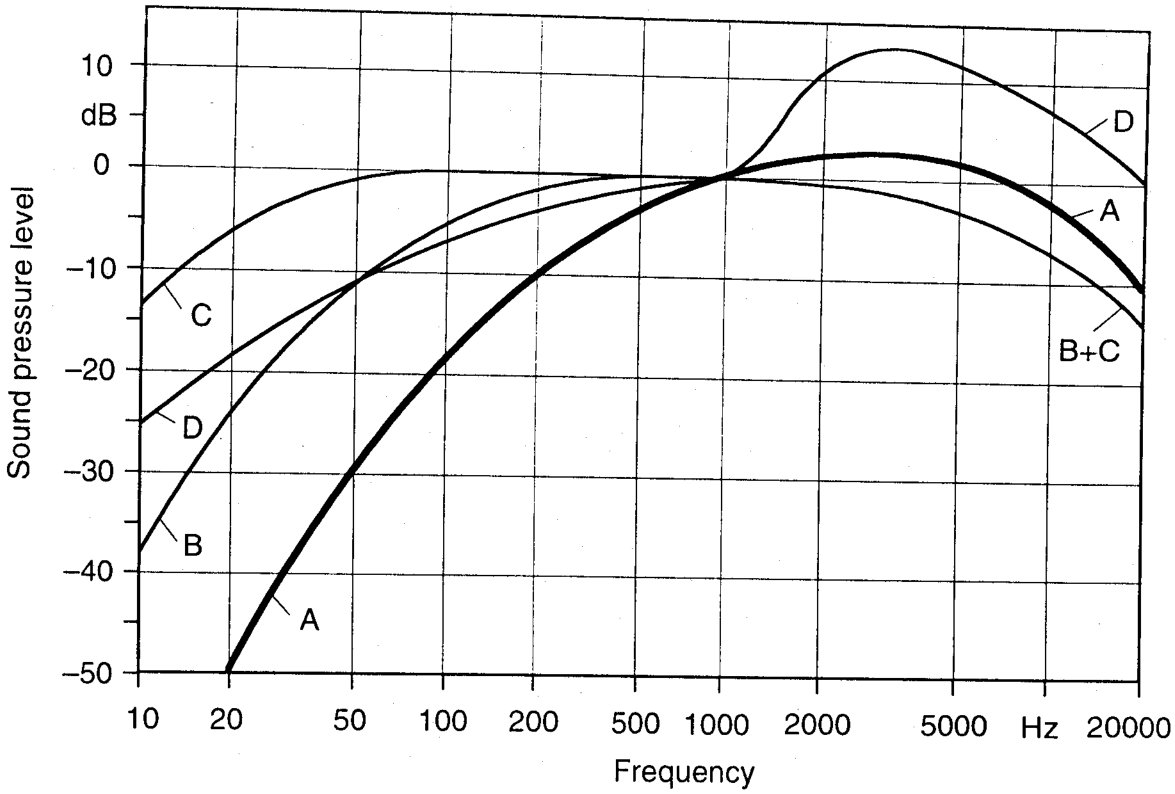


Figure 7.26. Frequency evaluation for sound pressure level. Curves for evaluation filters A, B, C and D

This evaluation function has been agreed internationally, and is very easy to use, particularly since it is incorporated in all standard sound ranging equipment. But unlike pure sound, transmission noise contains a mixture of many frequencies. The A evaluation is therefore not necessarily applicable. More complex methods of evaluation are needed, which, although known, are not widely used because of their complexity in use. But if similar noises (i.e. noises with a similar spectral distribution) are measured under the same conditions, the A evaluation levels can be meaningfully compared.

The microphone of a sound-ranging device outputs a signal which is proportional to the sound pressure profile (alternating part of the air pressure). Either the whole frequency range of human hearing is filtered out of this signal (20 Hz to 20 kHz), or a narrower band is filtered out and the root mean square value derived. The root mean square value is finally output logarithmically in decibel units, since there is a very large range from the hearing threshold to the threshold of feeling, and the quantitative sense of hearing is roughly logarithmic.

The decibel unit is so configured that 1 dB roughly corresponds to the difference threshold of human hearing. When adding up the levels of several sound sources, it is necessary to add the sound intensities of the individual sources, which are proportional to the acoustic power. The total of two levels of equal volume is thus 3 dB more than that of the individual levels. This means that when two levels with a difference of more than 6 dB are added, the quieter one has practically no effect, since the total is less than 1 dB more than the louder source alone.

Airborne sound is often measured in special chambers. A distinction is made between anechoic chambers and echo chambers, which are used to suit the particular measuring task. Anechoic chambers are used to simulate an ideal free field, echo chambers to simulate a completely diffuse sound field.

Structure-borne sound is measured as well as airborne sound. Acceleration sensors are usually used for this purpose, since they are simple to use and to attach to the desired point on the gearbox. There are hardly any impinging variables in this type of measurement, since only the state of motion at a particular point is measured. The root mean square value is normally derived from the signal of the acceleration sensor, as when measuring airborne noise.

Frequency analysis is an important tool for analysing transmission noise, especially when allocating a noise component to a particular transmission component. The gear pairs responsible for particular peaks in the noise spectrum can easily be identified from the speeds of rotation and number of teeth. If the amplitude frequency curves recorded at various fixed speeds are plotted sequentially over the whole speed range of the transmission in the form of a waterfall diagram, natural vibrations can easily be distinguished from speed-related vibration. This characteristic is clearly apparent in Figure 7.27.

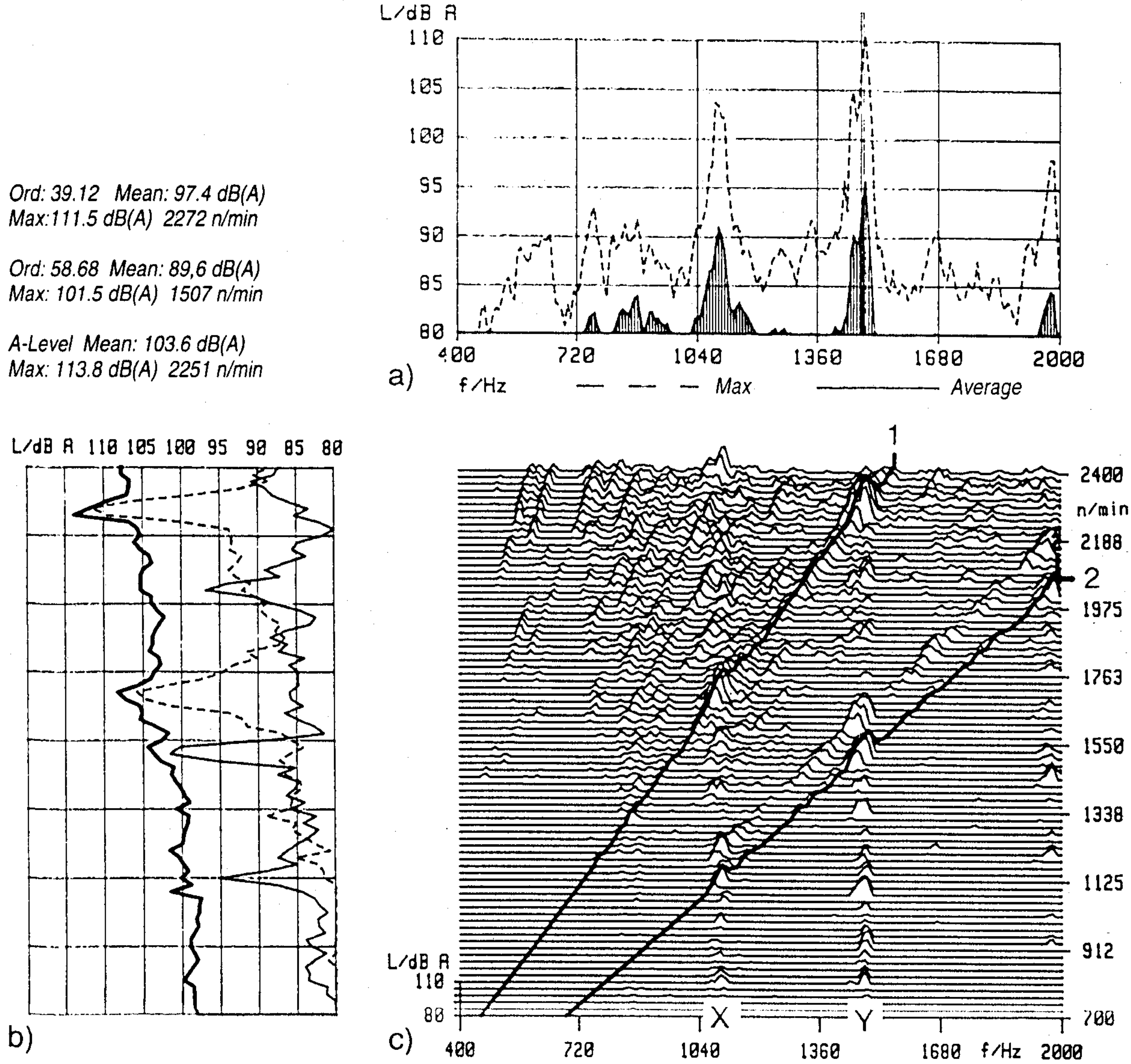


Figure 7.27. Frequency analysis of noise from a truck transmission. **a)** Maximum and average exposure as a function of frequency; **b)** Exposure as a function of speed for lines 1, 2 and the A-level; **c)** "Waterfall diagram"

It is clear that for example at frequencies of 1096 Hz (Figure 7.27c: *X*) and 1490 Hz (Figure 7.27c: *Y*) vibrations occur that are unrelated to rotational speed, whereas the vibration patterns marked (1) and (2) are speed-related. Figure 7.27b shows the noise level of the three salient lines (1, 2, and A-level) as a function of speed, and Figure 7.27a shows the maximum or average noise level as a function of frequency.

7.5.4 Countermeasures

Noise reduction has to be built into the development of a new transmission right from the planning and drafting stage, because it is extremely difficult to effectively reduce the noise emissions of an existing unit. A distinction can be made between active measures to reduce the generation of noise, and passive measures affecting its propagation (Table 7.5). Active measures thus only ever affect a particular type of vibration or noise. Passive measures reduce the transmission noise level overall. Active measures to reduce the vibrations of power-transmitting gear pairs affect the geometry and production quality of the toothing. A large transverse contact ratio and overlap ratio (high-contact gearing and helical gearing) reduces the irregularity of the resultant tooth rigidity and moderates the meshing impact.

The meshing faults due to load and consequent deformation are reduced firstly by profile corrections (tip relief, transverse crowning), and secondly by making gear bodies, shafts, bearings and housings as rigid as possible to prevent excessive deformation. The production quality of the toothing is the main factor in rolling contact noise. The speed the transmission runs at has a major effect on the noise emitted by the gearwheels under load, but the load itself has little effect. But neither of these operating parameters can usually be changed.

The extent of loose part vibration is affected by three parameters in the transmission itself. The backlash and the moments of inertia should be as small as possible, and the drag torque on the individual loose parts should be as large as possible. Since there are many functional constraints, there is little potential for noise reduction here. Rattling and clattering can be significantly reduced, or even eliminated, if the amplitude of the torsional vibrations of the transmission shafts are reduced accordingly. If engine irregularity is de-coupled, this creates great potential for noise reduction.

It is not yet possible to offer a universal prescription as to how to reduce engagement noise; each transmission requires individual matching and tuning. Apart from synchronisation components themselves, the dynamic behaviour of the whole power train during gear shifting must be considered. Passive measures in the gearbox relate in particular to the transmission of structure-borne sound through the shafts from the gear teeth to the housing. A soft bearing support should be used as low-pass filter for the high-frequency vibrations, without there being any unacceptable deformation under load. The design of the gearbox housing is of particular importance. Noise-intensive forms of natural vibration in the transmission walls must be avoided. The housing material also has a major effect on sound reflection. The light alloys in most common use today have much poorer attenuation characteristics than cast iron.

The transmission of structure-borne sound to the bodywork must be minimised by tuning the transmission mountings. But here too there are conflicts with other functional demands on the power-train mounting in the vehicle.
Airborne noise radiation can be nearly completely prevented by encasing the gearbox, but there are penalties in terms of weight, heat dissipation and price. Verifying the effectiveness of noise reduction measures is a very resource-intensive process. For example every time the housing is modified, a new casting mould has to be made.

Table 7.5. Active and passive measures for reducing transmission noise

<table>
<tr><th colspan="2">Active measures/reduction of noise generation</th></tr>
<tr><th>Internal</th><th>External</th></tr>
<tr><th colspan="2">Vibration of gearwheels under load</th></tr>
<tr><td>Toothing geometry:
– Helical gears ($\varepsilon_\alpha + \varepsilon_\beta > 2.5$)
– High contact gearing ($\varepsilon_\alpha > 2$; not 2; 2.5)
– Profile correction (tip relief)
– No integral gear ratios
Toothing quality:
– Tolerances (IT 7 to IT 4)
– Machining processes grinding, shaving, honing (surface)</td><td>Operating status (cannot generally be changed):
– Speed
– Torque</td></tr>
<tr><th colspan="2">Loose part vibration</th></tr>
<tr><td>– Backlash constriction
– Arrangement of idler gears
– Synchroniser type
– Drag torque increase
– Direct measures:
Bracing idler gears or synchroniser rings (e.g. mechanical or magnetic)</td><td>Reduce transmission shaft torsional vibration:
– De-couple the engine: torsion dampers in clutch plate or two-mass flywheel
– Reduction of resonance rise by dampers</td></tr>
<tr><th colspan="2">Passive measures/Reduction of sound propagation</th></tr>
<tr><td colspan="2">Reduce structure-borne sound within the transmission:
– Bearings and bearing support
Reduce housing vibration:
– Housing design (ribbing)
– Housing material
Reduce transmission of structure-borne sound to bodywork:
– Transmission and engine suspension
Encasing the transmission</td></tr>
</table>

It would therefore be desirable to determine the generation and propagation of transmission noise by calculation, which could significantly reduce the time and expense required to perfect the design. Research is currently being conducted into this problem. To calculate excitation, the vibrations of gearwheels and shafts are modelled as multiple-component systems. Even very complex models can be created using digital computing techniques. The main problem is determining the parameters, especially the attenuation values. It has so far proved possible only to estimate housing vibration, since the structures are very complex. The method generally used is the finite element method (FEM).

8 Specification and Design of Shafts

The size of a gearbox is largely determined by the diameters of the transmission shafts

The specification and design of transmission shafts is of special significance in the layout of vehicle transmissions. Shaft diameters are a key factor in determining the centre distance of a gearbox, and thus its size. Strength *and* resistance to distortion must therefore be carefully considered during the design process.

8.1 Typical Problems in Vehicle Transmissions

8.1.1 Configuration of Shafts in Vehicle Transmissions

Figure 8.1a shows the shaft configuration (drive shaft, countershaft and main shaft) of a two-stage countershaft transmission for standard drive. Figure 8.1b shows the shaft configuration for a single-stage countershaft transmission intended for front transverse mounting.

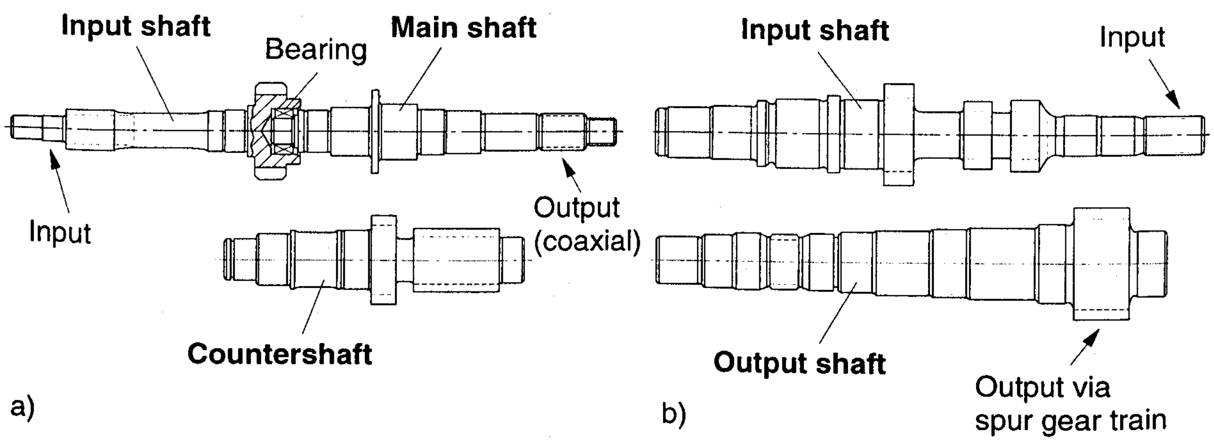


Figure 8.1. Characteristic shaft configurations in vehicle transmissions. **a)** Two-stage coaxial countershaft transmission ("3-shaft transmission"); **b)** Single-stage countershaft transmission for front transverse mounting

8.1.2 Designing for Stress and Strength

The process of designing transmission shafts for operational strength is described in Section 7.4 "Operational Integrity and Service Life". The following factors have to be borne in mind when determining the load profiles for transmission shaft design, the "design duty cycles":

- Stresses when moving off and changing gear are determined largely by the driver, and are consequently subject to wide variation; but they have a crucial effect on the service life of the transmission shaft, as do driver errors such as when the foot slips off the clutch pedal causing a violent engaging jolt.
- The load profile derived from a loading pattern depends on the classification procedure used (see also Section 7.4.2. "Load Profile and Enumeration"). This means that, for example, load profiles determined for gearwheels using the class continuity procedure cannot simply be used to determine the serviceability design of transmission shafts. The information on the amplitude and mean amplitude of the individual oscillations is lost. Dual-parameter enumerations such as the Rainflow enumeration are therefore preferable for the serviceability design of shafts; they take into account the amplitude and mean amplitude of each oscillation.

There are three approaches to designing transmission shafts:

1. *Preliminary specification of the shaft diameters (Section 8.3.7)*
   This involves an initial estimate of the shaft diameters required for a given loading.
2. *Design for fatigue strength (Section 8.3.8)*
   The design is based on the maximum anticipated loading; the transmission must be capable of sustaining that load long term. The maximum engine torque T_{max} is used in the calculation. A transmission that is completely durable under continuous maximum load is generally over-engineered.
3. *Specification of operational integrity (Section 8.3.9)*
   Transmission shafts are designed for a finite service life based on established load profiles. For this purpose it is necessary to establish design duty cycles; in the case of toothed gear transmissions the load class in each gear must be taken into account, and also the proportion of mileage covered in the various gears.

The shaft configuration typical of vehicle transmissions is particularly unfavourable from the point of view of strength. Large distances between bearings cause large bending moments, and there are many notches as a result of shaft clamps, grooves, collars, bearing seats, etc. (Figure 8.2).

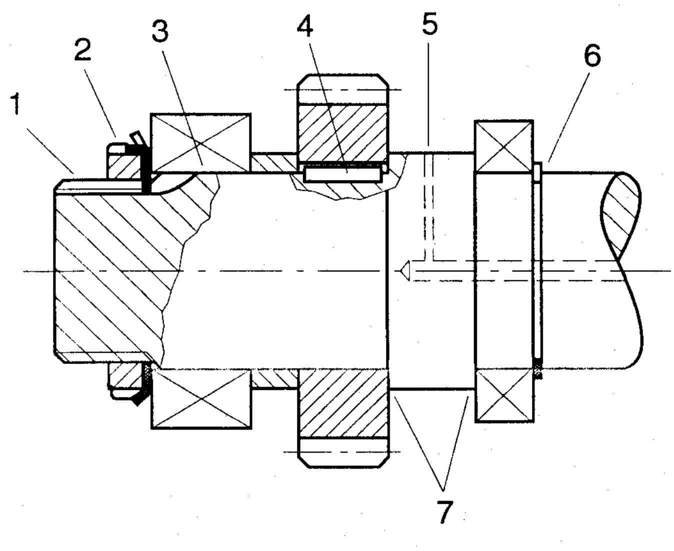


Figure 8.2. Typical notches on transmission shafts:
1 Thread
2 Slot for locking plate
3 Bearing seat
4 Feather keyway
5 Transverse drill hole
6 Slot for circlip
7 Shoulders

8.1.3 Deflection

Vehicle transmissions have long shafts with long distances between bearings, and are usually subjected to asymmetrical loads. This results in large deflections f and large bending angles φ (Figure 8.3).

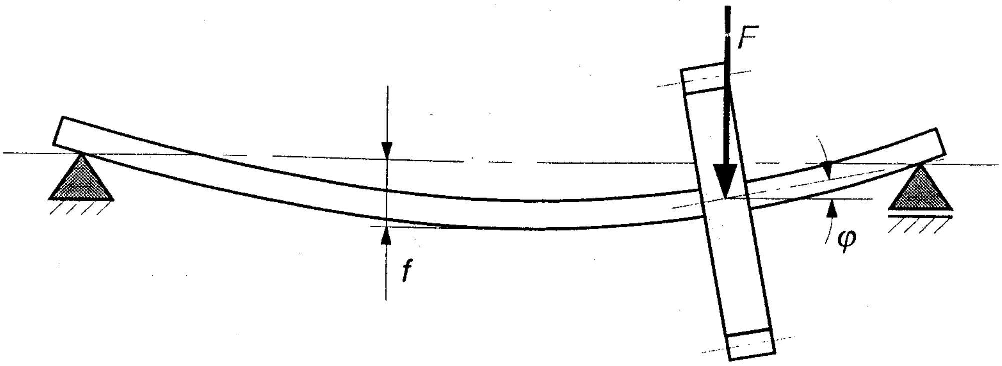


Figure 8.3. Deflection f and bending angle φ in shafts with large distances between bearings, and asymmetrical loading

The resultant inclination of the teeth causes a one-sided contact pattern, i.e. the active width of the driving face is reduced, increasing stress on the teeth (Figure 8.4).

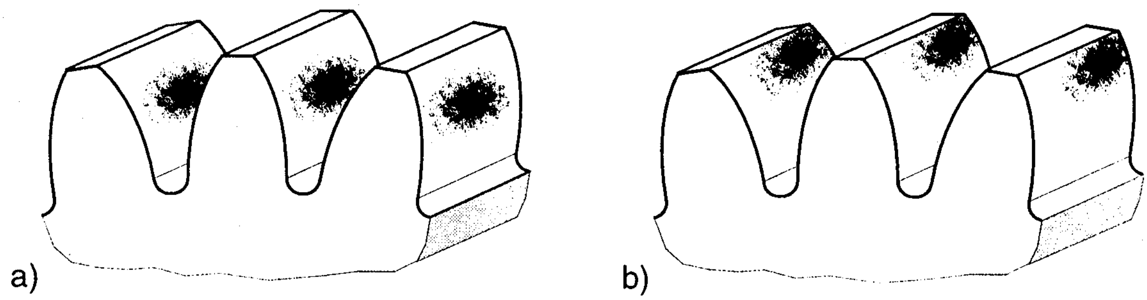


Figure 8.4. Contact pattern: **a)** Uniform contact pattern; **b)** One-sided contact pattern

To avoid this, the shaft deflection and the strength calculations must be checked very accurately, preferably taking into account the deformation of the housing and bearings.

8.1.4 Vibration Problems

Power-train vibration presents particular challenges in transmission design. Although the bracing effect of bearings and hubs and also the effects of idler gears and synchromesh can hardly be measured, it is essential to analyse the vibration behaviour of the transmission, since the stress peaks and deformations caused by dynamic effects can be substantial.

The dynamic behaviour of a transmission must always be considered in combination with the power train as a whole. Vibration analysis can be carried out on test beds and also by computer simulation. There are numerous mathematical models available [8.1, 8.2]. They are based on multiple-component systems with linear or non-linear linking. One significant factor with these models is the "critical speeds". Unbalanced rotating masses generate forces that cause vibration. Transmission shafts with rotating gearwheels, synchromesh, etc. have several critical whirling speeds that result in bending vibration resonance. High critical speeds are preferred, as they give quiet running and a long service life.

A distinction is made between:

- *Torsional vibration*
  Low-frequency and high-frequency vibration principally caused by irregularities in the power flow from internal combustion engines (see also Figure 5.14).
- *Bending vibration*
  Higher-frequency transmission shaft vibrations. This can for example be caused by tooth meshing (see also Figure 7.23 "Overall tooth rigidity pattern").

8.2 General Design Guidelines

The problems specific to vehicle transmissions described in the previous section lead to three main requirements for transmission shaft design:

1/ Avoid notches!

2/ Reduce bending moments!

3/ Increase critical speeds!

To satisfy these requirements, the following design principles should be followed:

- Reduce the distance between bearings by means of compact overall structural design.
- Locate heavily stressed gearwheels close to bearings to reduce deflection and bending torque, and to achieve high critical whirling speeds.
- Keep diameter transitions between shafts below the ratio $D/d \approx 1.4$. The transitions should preferably be of conical design or with a large radius of curvature rather than with shaft shoulders.
- Specify splined connections or oil press connections rather than feather key connections.
- Smooth rectangular ring-grooves by using relief notches or by rounding off the inside edges of the groove (Figure 8.5a).
- Locate circlips only at the end of the shaft if at all possible. Use distance sleeves for axial restraint at the middle of the shaft.
- Reduce the notching effect at the shaft chamfers (Figure 8.5b): *1* Locate relief notches at the transition by means of rounded axial grooves; *2* Use large rounding radius; *3* Use radial stress relief edges; *4* Use additional notches in the transition zone.
- Shafts with a mounted hub must be made thicker at the wheel seat; specify a large transition radius and reduce the thickness of the hub towards the rim (Figure 8.5c).
- Transverse drill holes should be smoothed by relief notches at the mouth of the drill hole, by increasing the shaft diameter and using larger transition radii, and by repressing the edges of the drill holes with a smooth thrust piece (Figure 8.5d).

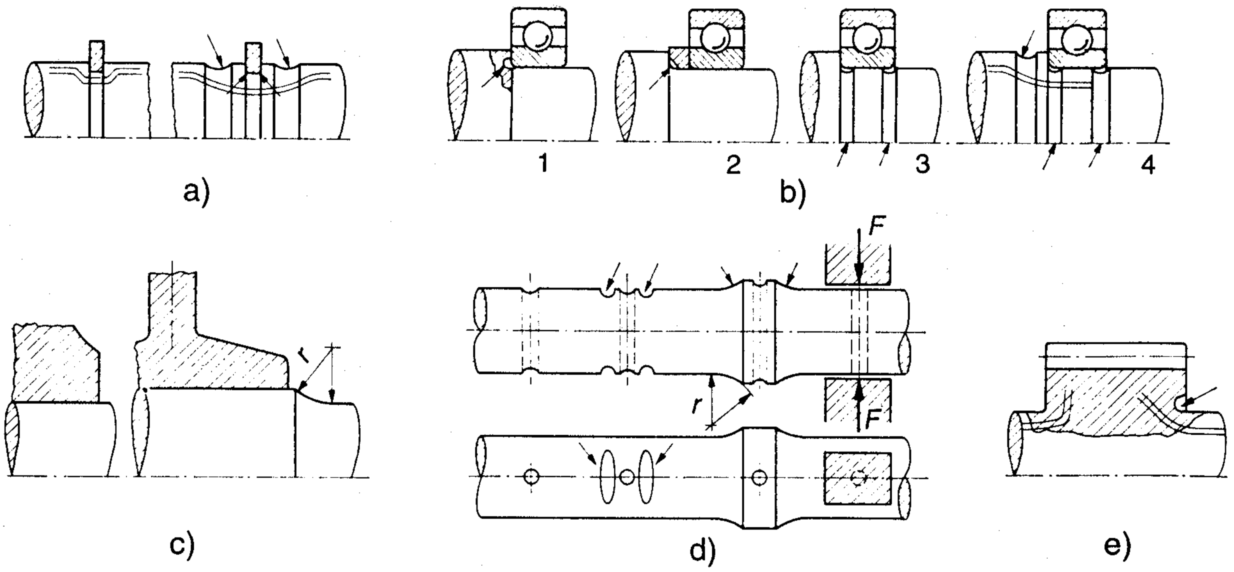


Figure 8.5. Several design approaches to reducing the notching effect (NIEMANN, [8.3])

- Gradual power diversion using relief notches (Figure 8.5e).
- Balance shafts precisely in order to minimise centrifugal forces and associated bending vibrations.
- Reduce the moment of inertia of components mounted on the shaft in order to reduce deflections and increase critical speeds.

8.3 Transmission Drive-Shaft Strength Design

The following model strength calculation is based on the transmission drive shaft shown in Figure 8.1b. The resultant load ratios vary depending on the gear engaged, so the following calculation has to be carried out to determine the most highly stressed point and the greatest deflection *for each gear* and also for the load conditions under *power and overrun*.

The calculation shown is restricted to statically defined cases, i.e. to two-bearing transmission shafts. Shafts secured at multiple points cannot be analysed statistically, and require very extensive computation which is beyond the scope of manual calculation in the case of graduated diameters (see also Section 8.4).

8.3.1 Loading

When designing a transmission, the designer initially works on the basis of the maximum engine torque T_{max}. (Engaging jolts can give rise to transmission input torque values more than twice as high). If the calculation relates to some other shaft than a transmission drive shaft, the effective torque in the shaft must be determined from the ratio of the respective gears.

Applied external forces, such as tooth forces and bearing forces, are treated as point loads. The forces are determined in the co-ordinate system of the power train, the x axis corresponding to the shaft axis. This gives rise to the system of forces at the drive-shaft shown in Figure 8.6.

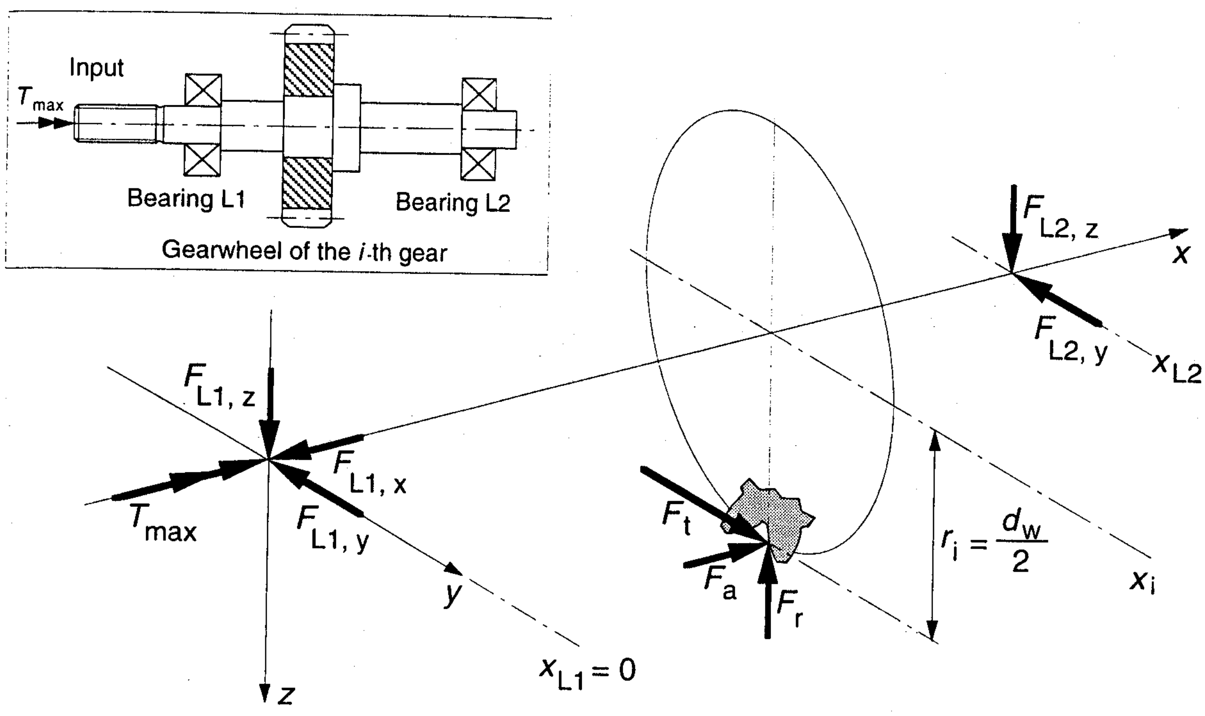
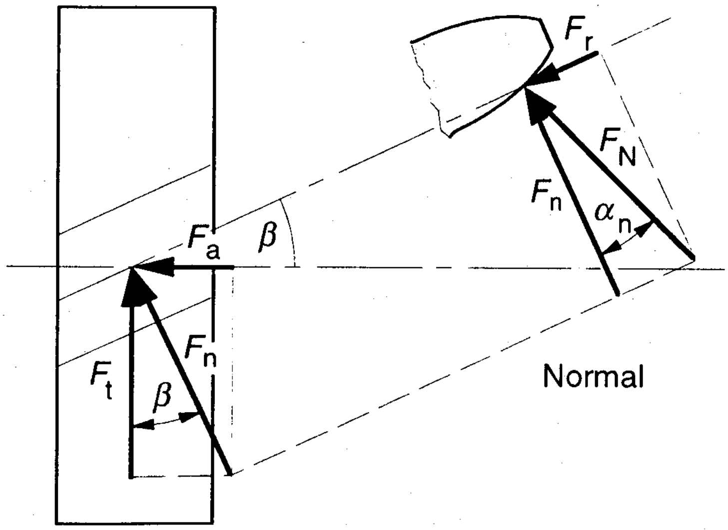


Figure 8.6. Loading of the transmission input shaft of the theoretical model (single-stage countershaft transmission)

The first step is to determine the forces at the tooth flanks, which depend on the gear currently selected and on the type of gear teeth. In the case of helical gearwheels there are axial forces as well as tangential and radial forces. Figure 8.7 shows the system of forces at the tooth flanks of a helical spur gearwheel.



$$F_t = \frac{2\,T_{max}}{d_w} \tag{8.1}$$

$$F_r = F_t\,\frac{\tan\alpha_n}{\cos\beta} \tag{8.2}$$

$$F_a = F_t\,\tan\beta \tag{8.3}$$

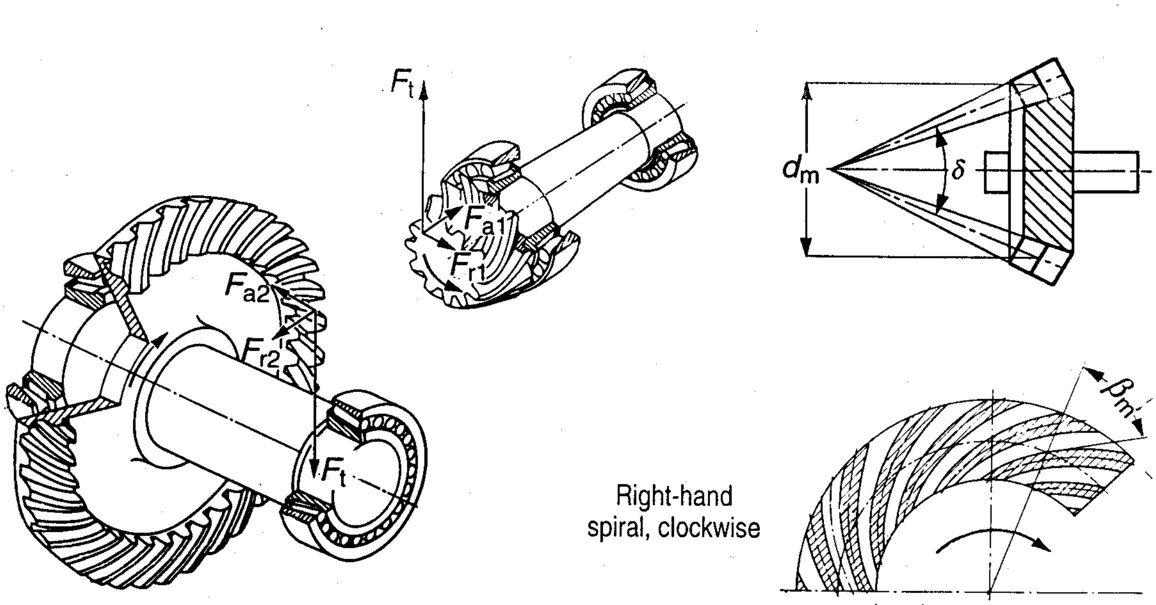
Figure 8.7. Forces acting at the tooth flanks of a helical cut spur gear

The tangential, radial and axial forces are calculated using the equations listed in Figure 8.7. In the case of spur gears, the radius to the point of application of the force depends on the pitch circle diameter d_W

$$r_i = \frac{d_W}{2}\ . \tag{8.4}$$

The equations for the forces F_t, F_r, F_a and for r_i for straight and helical bevel gears are summarised in Table 8.1.

Table 8.1. Forces acting on straight-cut and helical bevel gears



Spiral gear/ direction of rotation driving gear	Axial force	Radial force
Right-hand spiral, clockwise or Left-hand spiral, anti-clockwise	**Driving gear** $F_a = \dfrac{F_t}{\cos\beta_m}(\tan\alpha_n \sin\delta + \sin\beta_m \cos\delta)$	**Driving gear** $F_r = \dfrac{F_t}{\cos\beta_m}(\tan\alpha_n \cos\delta - \sin\beta_m \sin\delta)$
	Driven gear $F_a = \dfrac{F_t}{\cos\beta_m}(\tan\alpha_n \sin\delta - \sin\beta_m \cos\delta)$	**Driven gear** $F_r = \dfrac{F_t}{\cos\beta_m}(\tan\alpha_n \cos\delta + \sin\beta_m \sin\delta)$
Right-hand spiral, anti-clockwise or Left-hand spiral, clockwise	**Driving gear** $F_a = \dfrac{F_t}{\cos\beta_m}(\tan\alpha_n \sin\delta - \sin\beta_m \cos\delta)$	**Driving gear** $F_r = \dfrac{F_t}{\cos\beta_m}(\tan\alpha_n \cos\delta + \sin\beta_m \sin\delta)$
	Driven gear $F_a = \dfrac{F_t}{\cos\beta_m}(\tan\alpha_n \sin\delta + \sin\beta_m \cos\delta)$	**Driven gear** $F_r = \dfrac{F_t}{\cos\beta_m}(\tan\alpha_n \cos\delta - \sin\beta_m \sin\delta)$

where	d_m	average pitch circle diameter (to centre face width)	β_m	helix angle (spiral angle) at the pitch cone in tooth centre
	$r_i = \dfrac{d_m}{2}$	radius to the point of application of the force	δ	pitch cone angle of the examined gearwheel
	$F_t = \dfrac{2T_{max}}{d_m}$	circumferential force	α_n	meshing angle normal

If there is spatial loading, i.e. F_t and F_r are not located on the specified co-ordinate planes, they must be resolved into their y and z components (Figure 8.8.).

$$F_y = F_t \cos \xi + F_r \sin \xi \,, \tag{8.5a}$$

$$F_z = F_t \sin \xi - F_r \cos \xi \,. \tag{8.5b}$$

When balancing torque levels, it must be taken into account that the effective lever arms are reduced in this case (Figure 8.8).

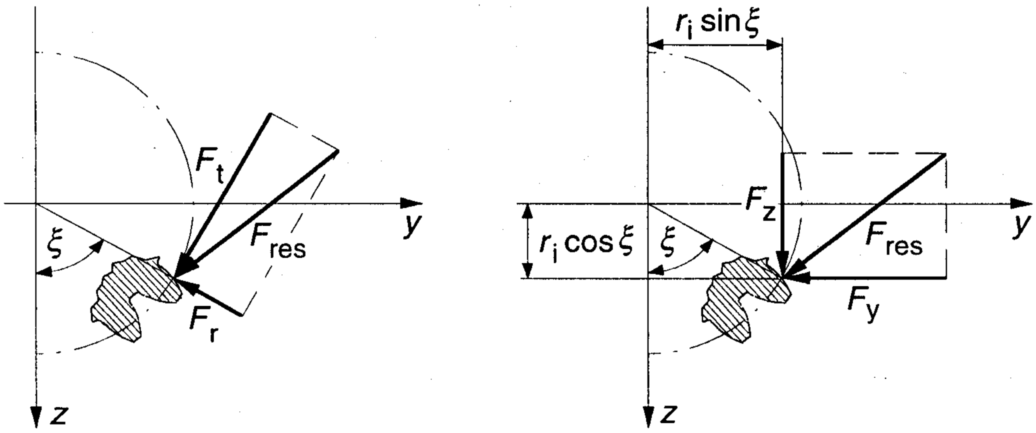


Figure 8.8. Conversion of tangential and radial forces in y and z components in the case of spatial loading

8.3.2 Bearing Reactions

The bearing forces F_{L1} and F_{L2} are calculated from the external forces. To determine the bearing reactions it is necessary to balance the forces ($\Sigma F = 0$) in the x, y and z directions and also to balance the moments ($\Sigma M = 0$) about the y and z axes relative to the origin of the co-ordinates.

In addition to the axial bearing forces, the resultant radial bearing forces are also of significance when calculating the bearings (Section 11.1.2)

$$F_{L1} = \sqrt{F_{L1,y}^2 + F_{L1,z}^2} \,, \tag{8.6a}$$

$$F_{L2} = \sqrt{F_{L2,y}^2 + F_{L2,z}^2} \,. \tag{8.6b}$$

8.3.3 Spatial Beam Deflection

In the general case of spatial beam deflection it is important to determine accurately the sign of the intersection reactions at the beam intersection bank. Figure 8.9 shows the method of determining the sign used for calculating transmission shafts.

In the xz plane (Figure 8.9a) all the intersection reactions (transverse load $F_{Q,\,z}$ and bending moment $M_{b,\,y}$) act in the direction of the positive co-ordinate axes, i.e. $M_{b,\,y}$ rotates about the y axis in a positive sense.

In the xy plane on the other hand (Figure 8.9b) only the transverse load $F_{Q,\,y}$ acts in the positive co-ordinate direction. The bending moment $M_{b,\,z}$ rotates about the z axis in a negative sense, but retains the same orientation to $F_{Q,\,y}$ as $M_{b,\,y}$ to $F_{Q,\,z}$ in the xz plane. But it is precisely this orientation that is critical for the relation between the transverse load and the bending moment.

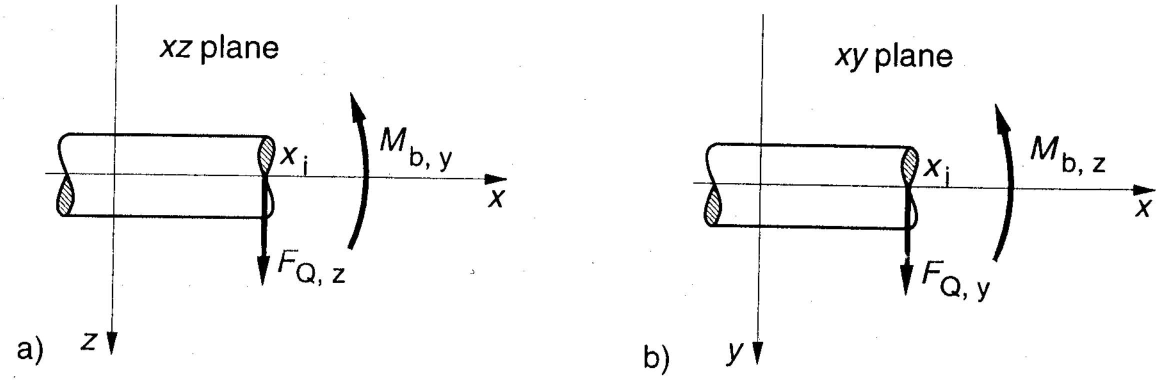


Figure 8.9. Determining the sign for the intersection reactions at the positive beam intersection bank at the point x_i. **a)** xz plane; **b)** xy plane

In engineering mechanics it is normal practice to use a closed right-hand system for the intersection reactions. Specifically the bending moment $M_{b,z}$ in the xy plane (Figure 8.9b) would be defined as positive against the direction shown.

There is thus no longer a corresponding relationship between the xz and xy planes, i.e. separate derivations must be carried out for both planes, some with changing signs. That is in a sense the price that has to be paid for a coherent scientifically exact representation of a spatial system.

But determining the sign in this way has the decisive advantage that the xz and xy planes can be treated in exactly the same way as regards the relation between transverse load and bending moment. But the following should be borne in mind:

- A positive sign in the power or torque profile calculated means positive in relation to the sign as previously determined.
- Equations relating the transverse loads, bending moment, deflections or directions of stresses are only valid if the sign as previously determined is taken into account. This is particularly important when performing comparisons with other literature or with the results of calculation programs.

Another problem with establishing the power and torque profiles subsequently is the application of individual forces and moments. This is why there are jumps at the points x_i in the power and torque profiles. These discontinuities can be overcome by introducing the "Heavyside overlap function" by means of which non-analytical functions can be represented as "closed".

$$\{x - x_i\} = \begin{cases} 0 & \text{for} \quad x \le x_i \\ (x - x_i) & \text{for} \quad x > x_i \end{cases} . \tag{8.7a}$$

In particular, a unit jump function may be defined as

$$\{x - x_i\}^0 = \begin{cases} 0 & \text{for} \quad x \le x_i \\ 1 & \text{for} \quad x > x_i \end{cases} . \tag{8.7b}$$

8.3.4 Power and Torque Profiles

In order to calculate the stresses in the transmission shaft, the bending moments and torsional moments must be known at every point on the shaft.

To determine the *bending moment profile* $M_b(x)$, the transverse load profile $F_Q(x)$ must first be calculated. The bending moment profile can then be calculated by integrating the transverse load profile as

$$M_b = \int F_Q(x)\,dx + C\ . \tag{8.8}$$

Equation 8.8 applies both in the xz plane and in the xy plane because of the sign determination in Figure 8.9. The parameters used to find the integration constant C are calculated from the sum of all individual points to the left of the point x_i. The action of individual moments must be taken into account, such as the *pitching moment* created by the axial force of a helical gearwheel (Figure 8.10). This changes the sum of all the individual moments, and there is a discontinuity in the bending moment profile at point x_i. This must be described using the Heavyside overlap function Equation 8.7.

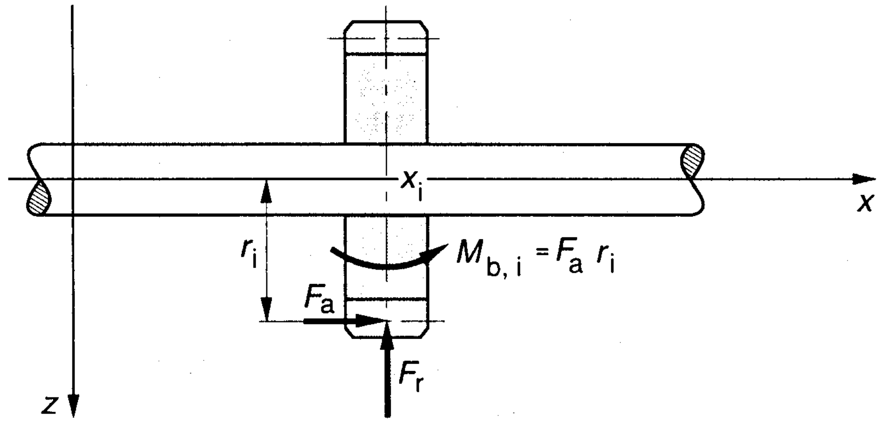


Figure 8.10. Generation of pitching moment with axial forces by helical gearing

In the xz plane the transverse load profile is given by

$$F_{Q,z}(x) = -F_{L1,z} + F_r\,\{x - x_i\}^0\ . \tag{8.9}$$

The bending moment profile can now be calculated from the transverse load profile by integration. It should be noted that there is a discontinuity of $F_a\ r_i$ in the bending moment profile at point x_i resulting from the axial force present

$$M_{b,y}(x) = -F_{L1,z}\,x + F_r\,\{x - x_i\}^1 - F_a\,r_i\,\{x - x_i\}^0\ . \tag{8.10}$$

A similar procedure applied to the xy plane results in

$$F_{Q,y}(x) = F_{L1,y} - F_t\,\{x - x_i\}^0 \tag{8.11}$$

giving

$$M_{b,z}(x) = F_{L1,y}\,x - F_t\,\{x - x_i\}^1\ . \tag{8.12}$$

Only the magnitude of the resultant bending moment is of significance in the subsequent stress calculation

$$M_b(x) = \sqrt{M_{b,y}^2(x) + M_{b,z}^2(x)}\ . \tag{8.13}$$

The *torsional moment profile* $M_t(x)$ in the transmission shaft is derived from the torque equilibrium around the x axis as

$$M_t(x) = -T_{max} + F_t\, r_i \left\{ x - x_i \right\}^0 . \tag{8.14}$$

The contact paths and torque profiles can also be represented graphically (Figure 8.11). This shows clearly how the bending moment profile (Figure 8.11b) is derived from the transverse load profile (Figure 8.11a) by integration. The torsional moment is constant between the power input and power output points, and has magnitude T_{max}, (Figure 8.11c).

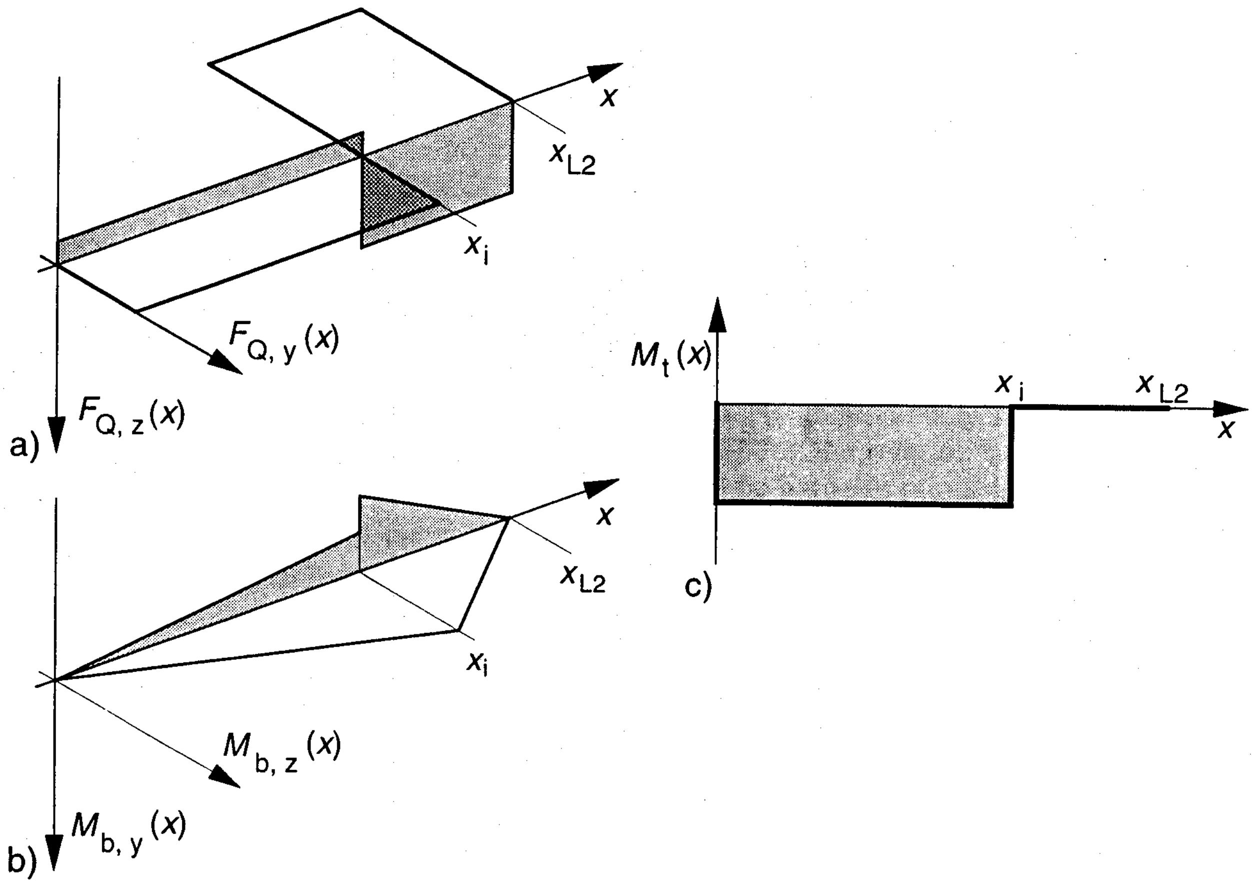


Figure 8.11. Representation of the qualitative contact paths and torque profiles of the sample calculation shown in Figure 8.6: **a)** Transversal force profile $F_Q(x)$; **b)** Bending moment profile $M_b(x)$; **c)** Torsional moment profile $M_t(x)$

8.3.5 Critical Cross-Section

The greatest reference stress acts in the critical, most highly stressed, cross-section. It must therefore be shown that the shaft can bear the resultant stress at this point with the required safety margin. For a successful design it is very important to check this critical cross-section. If the location of the critical cross-section cannot be precisely predicted, several cross-sections will have to be analysed.

The following criteria can be used to determine the location of critical cross-sections:

- peaks in the bending moment and torsional moment profiles (Fig. 8.11 point x_i),
- small shaft diameters,
- notches.

The critical factor is often the peak stresses at the notches! Thus the critical cross-section does not necessarily have to coincide with the peak in the torque profiles.

8.3.6 Stresses

In calculating the effective stresses in transmission shafts it is normal practice to ignore the direct stress caused by axial forces and the shear stress caused by transverse loads. The only significant stresses are those caused by bending moment and torsional moment.

Nominal Stresses

The maximum stress value σ_{max} occurs in the notch root. Depending on the notch acuity, this value exceeds the stress σ_n that would prevail in an un-notched shaft with the same root cross-section. σ_n is the symbol for nominal stress. The following points must be borne in mind when calculating the nominal bending stress and torsional stress:

- The magnitude of the resultant bending moment M_b from Equation 8.13 determines the nominal bending stress, since transmission shafts rotate and have a circular cross-section.
- If the critical cross-section is located at a discontinuity in the torque profile, then the moment with the greatest magnitude must be used in calculations.

Nominal bending stress:
$$\sigma_{b,n} = \frac{M_b}{W_b} , \tag{8.15a}$$

Nominal torsional stress:
$$\tau_{t,n} = \frac{M_t}{W_t} . \tag{8.15b}$$

Notch Stresses

The increased stress in the notch root causes the nominal stresses to rise by the form factor α_k under static stress, and by the fatigue notch factor β_k under dynamic stress. Thus

under static stress
$$\sigma_{b,max} = \alpha_{k,b}\ \sigma_{b,n} , \tag{8.16a}$$
$$\tau_{t,max} = \alpha_{k,t}\ \tau_{t,n} , \tag{8.16b}$$

under dynamic stress
$$\sigma_{b,max} = \beta_{k,b}\ \sigma_{b,n} , \tag{8.16c}$$
$$\tau_{t,max} = \beta_{k,t}\ \tau_{t,n} . \tag{8.16d}$$

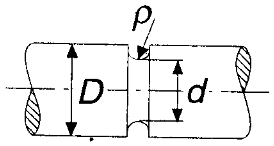
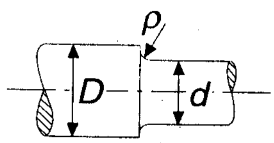
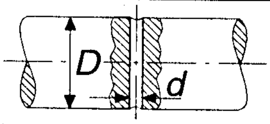
A transmission shaft will of course always be subject to dynamic stress, but if the β_k values are difficult or impossible to determine, the calculation can also be performed using the static form factor, α_k, justifiably erring on the side of safety. Under dynamic stress the form factor α_k, which applies under static stress, does not fully impact on the stress increase, thus:

$$1 \le \beta_k \le \alpha_k , \tag{8.17}$$

where $\beta_k = 1$ insensitivity to notches and
$\beta_k = \alpha_k$ full notch sensitivity.

The static form factors α_k can be determined from the shaft geometry, and are set out in Table 8.2 for various classes of stress. There are various methods of calculating the dynamic fatigue notch factor β_k [8.4–8.8]. Standard values for β_k are also given in [8.3] and [8.9].

Table 8.2. Determining the static form factor α_k from the shaft geometry

Load type / Notch type	Bending $\alpha_{k,b}$					Torsion $\alpha_{k,t}$				
	ρ/d \ D/d	1.01	1.03	1.15	∞	ρ/d \ D/d	1.01	1.05	1.20	∞
	0.05	1.60	1.90	2.40	2.60	0.05	1.30	1.55	1.75	1.85
	0.10	1.40	1.60	1.87	2.00	0.10	1.20	1.35	1.46	1.52
	0.20	1.27	1.40	1.55	1.60	0.20	1.14	1.22	1.29	1.32
	0.30	1.20	1.30	1.35	1.40	0.30	1.12	1.17	1.20	1.22
	ρ/d \ D/d	1.01	1.10	2.00	6.00	ρ/d \ D/d	1.09	1.20	2.00	6.00
	0.05	1.55	1.89	2.19	2.40	0.05	1.26	1.55	1.75	1.80
	0.10	1.36	1.58	1.73	1.88	0.10	1.17	1.32	1.45	1.51
	0.20	1.20	1.37	1.44	1.50	0.20	1.11	1.16	1.25	1.31
	0.30	1.13	1.27	1.30	1.33	0.30	1.08	1.11	1.17	1.22
	d/D	0.01	0.10	0.20	0.40	d/D	0.01	0.10	0.20	0.40
		2.70	2.35	2.02	1.80		1.90	1.65	1.50	1.40

The fatigue notch factor β_k can be determined as a function of the form factor α_k, the tensile strength R_m and the type of stress using the RÜHL method [8.6]. Figure 8.12 shows on the left the dependence of $1/\beta_k$ on α_k as revealed by experiment. This diagram only applies to steel with $R_m = 550$ N/mm², so $1/\beta_k$ for steels with different R_m must be adjusted by a factor taken from the right hand side of Figure 8.12. If the diagram gives $\beta_k > \alpha_k$ then, as a consequence of Equation 8.17, let $\beta_k = \alpha_k$.

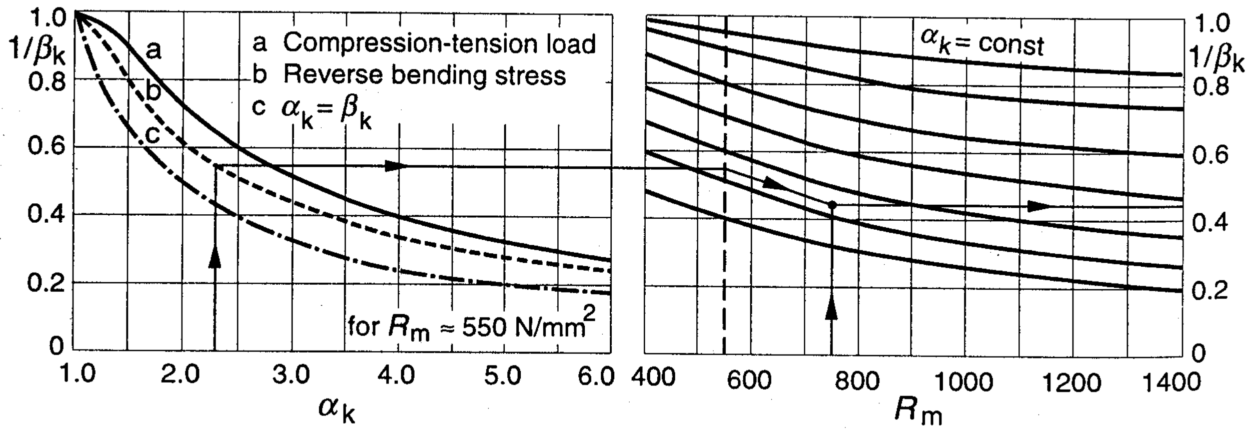


Figure 8.12. Determining the fatigue notch factor β_k using the RÜHL method [8.6].
Sample reading: $R_m = 750$ N/mm², $\alpha_k = 2.3 \;\Rightarrow\; \beta_k = 2.22$

Reference Stress

When designing transmission shafts reference stress σ_v arising from bending and torsional stress is calculated on the basis of the shape modification energy hypothesis as

$$\sigma_v = \sqrt{\sigma_{b,\,max}^2 + 3\,(\alpha_0\ \tau_{t,\,max})^2}\ , \tag{8.18}$$

with the "effort ratio"

$$\alpha_0 = \frac{\sigma_{perm} \text{ according time profil of } \sigma}{\sigma_{perm} \text{ according time profil of } \tau} = \frac{\sigma_{perm}(\sigma)}{\sigma_{perm}(\tau)}\ , \tag{8.19}$$

which takes into account the influence of various load scenarios for σ_b and τ_t. The torsional stress component τ_t is transformed to the time profile of the bending stress component σ_b, using α_0. Assuming the same safety margins for σ_b and τ_t, common instances are

$$\left.\begin{array}{l}\sigma_b \text{ reversing}\\ \tau_t \text{ reversing}\end{array}\right\}\quad \alpha_0 = 1\ , \tag{8.20a}$$

$$\left.\begin{array}{l}\sigma_b \text{ reversing}\\ \tau_t \text{ pulsating}\end{array}\right\}\quad \alpha_0 = \frac{\sigma_{b,\,W}}{\sigma_{b,\,Sch}} \approx 0.7\ . \tag{8.20b}$$

The effort ratio α_0 is thus derived from the permissible stresses of two time-dependent types of load. The influence of the effort ratio α_0 can be represented graphically by entering the pairs of permissible stress component values in a σ_b-τ_t plot (Figure 8.13).

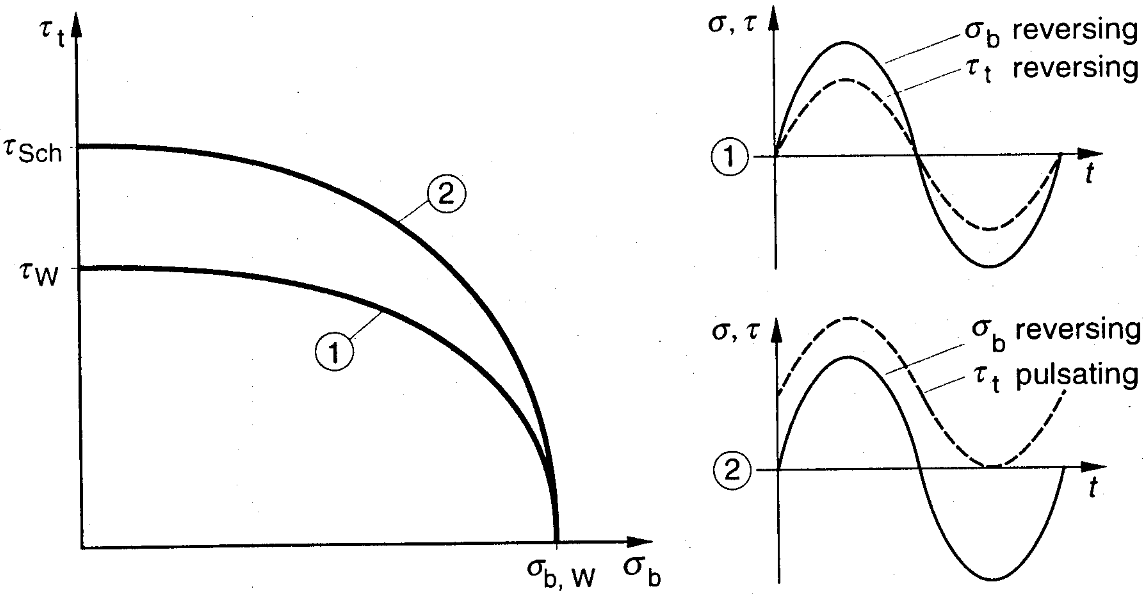


Figure 8.13. Boundary lines of the acceptable stress components

Reference Moment

When specifying the shaft diameter for a known stress, the "reference moment" M_v is often used

$$M_v = \sqrt{M_{b,\,max}^2 + 0.75\,(\alpha_0\ M_{t,\,max})^2}\ . \tag{8.21}$$

8.3.7 Preliminary Specification of the Shaft Diameter

A rough preliminary design of the transmission shafts is required along with a determination of the centre distance for the first outline design of a transmission. The minimum diameter of a solid shaft can be estimated using the condition $\sigma_b = M_b \,/\, W_b$ and Equation 8.21. Where $W_b = \pi / 32\, d^3$ and $\sigma_v = \sigma_{b,\,perm}$, then

$$d_{min} = 2.17 \sqrt[3]{\frac{M_v}{\sigma_{b,\,perm}}} \; . \tag{8.22}$$

8.3.8 Designing for Fatigue Strength

The reference stress σ_v calculated using Equation 8.18 depends on the maximum engine torque T_{max} (see Section 8.3.1). If the shaft is to have good fatigue resistance, T_{max} must be tolerated for the entire service life. The strength requirement is therefore

$$\sigma_v \leq \sigma_{b,\,perm} = \frac{\sigma_{b,\,w}\, b_s\, b_0}{S_D} \, , \tag{8.23}$$

$\sigma_{b,\,w}$	fatigue strength under reverse bending stresses,
b_s	surface effect (various finishes, Figure 8.14a),
b_0	size effect (stress gradient, etc., Figure 8.14b),
S_D	safety margin for endurance failure.

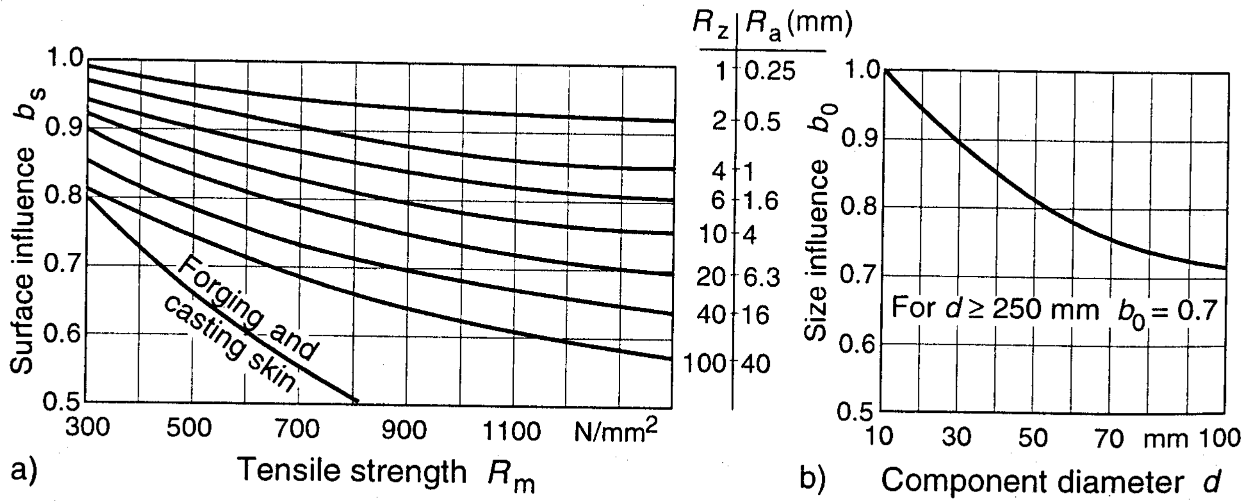


Figure 8.14. For circular cross-sections: **a)** Surface effect b_s, **b)** Size effect b_0

The occasional operating conditions where the transmission input torque is greater than the maximum engine torque T_{max} (e.g. clutch engaging jolts) are ignored when designing for fatigue strength. With a fatigue-resistant design ($\sigma_{b,\,perm}$ significantly smaller than R_m) and robust materials these are normally tolerable. The stress peaks are dissipated by local plastic flow. But a strength analysis is required if extremely severe and frequent impacts are anticipated.

8.3.9 Designing for Operational Integrity

Considerations are different when designing for a finite life. In this case the maximum engine torque T_{max} does not have to be endured indefinitely; it is sufficient if the transmission shaft does not fail during the required service life under the stress of a given load profile. The calculations required to determine the service life are extremely extensive, and are therefore normally carried out by computer.

An example follows to illustrate how the calculation can be performed manually. The requirements for the calculation are that:

- The "design duty cycle" for the transmission concerned is available.
- The proportion of revolutions of the transmission input shaft related to the cycle, as a function of the currently selected gear i and the stress class m is known.
- If a shaft other than the transmission input shaft is to be calculated, then the revolutions of this shaft must be calculated relative to the cycle with the ratio of the currently selected gear.
- In the critical cross-section, the reference stress σ_v must be determined for each gear at a stress T_{max}.

The reference stress σ_v (T_{max}), which arises when the transmission input torque T_{max} is applied, was calculated in Section 8.3.6, Eq. 8.18. If the load T_G at the transmission input is now varied, σ_v varies proportionally. Therefore the following relationship applies

$$\sigma_v(T_G) = \sigma_v(T_{max}) \frac{T_G}{T_{max}} . \tag{8.24}$$

σ_v (T_G) is thus a reference stress value in the critical cross-section, and depends on the gear selected, i, and the load T_G of the m-th stress class.

The service life calculation proceeds in a similar manner to the PALMGREN-MINER damage accumulation hypothesis described in Section 7.4.3. In contrast to the gearwheel service life calculation, transmission shafts are subject to a torque that varies with the gear currently selected. As a consequence, the distance travelled in the various gears must be taken into account as well as the distance covered in the various stress classes.

The number of possible load cycles is given by Equation 7.27 on the basis of the design duty cycle

$$z = \frac{N_D}{\sum\limits_{i=1}^{j} \sum\limits_{m=1}^{n} h_{im} \left(\frac{\sigma_{v,\,im}}{\sigma_D} \right)^{+k}} , \tag{8.25}$$

where $\sigma_{v,\,im}$ reference stress in the i-th gear, under load T_G of the m-th stress class given by Equation 8.24,
h_{im} proportion of transmission shaft revolutions related to the design basis cycle,
σ_D fatigue strength,
N_D number of oscillations at which fatigue strength is reached,
k exponent of the Wöhler curve equation (c.f. Section 7.4.1),
i index of gears 1 to j,
m index of stress classes 1 to n.

8.3.10 Normal Shaft Materials

The characteristics of normal shaft materials most important for strength design are summarised in Table 8.3.

Table 8.3. Materials commonly used in automotive shafts, and their characteristics [8.11] (figures in N/mm^2)

Description		R_m	R_e; $R_{p0.2}$	$\sigma_{b,W}$	$\sigma_{b,Sch}$	$\tau_{t,W}$	$\tau_{t,Sch}$
Tempering steel	25 CrMo4	800–950	530	430	730	300	450
	34 Cr4	750–900	550	425	–	–	690
Case hardening steel	16 MnCr5	900–1400	640	520	770	370	520

8.4 Calculating Deformation

Confirming that the deflection and bending angles of transmission shafts do not exceed the permissible limits is as important as analysing their strength.

The strength design computation shown in Section 8.3 is easy to program. This method is no longer relevant to calculating the deflection of transmission shafts with graduated diameters. For simpler cases the CASTIGLIANO method or the MOHR graphical method could be used. But the loads that occur in reality are normally more complex, so that special programs are required to calculate the bending line.

One common method is the *beam deflection transfer matrices method* [8.10]. Both the load profile and the deflection profile can be calculated using this method. Only the principles of this method will be given here.

The starting point is to divide the beam into sections in which the load q_i and the flexural strength $E_i\, I_i$ are constant (Figure 8.15). For each section, the relation between deflection f, bending angle φ, bending moment M_b and transverse load Q is described by the system of differential equations

$$\frac{df}{dx} = \varphi(x); \quad \frac{d\varphi}{dx} = \frac{M_b(x)}{E_i\, I_i}; \quad \frac{dM_b}{dx} = Q(x); \quad \frac{dQ}{dx} = q_i(x) = \text{const}\,. \tag{8.26}$$

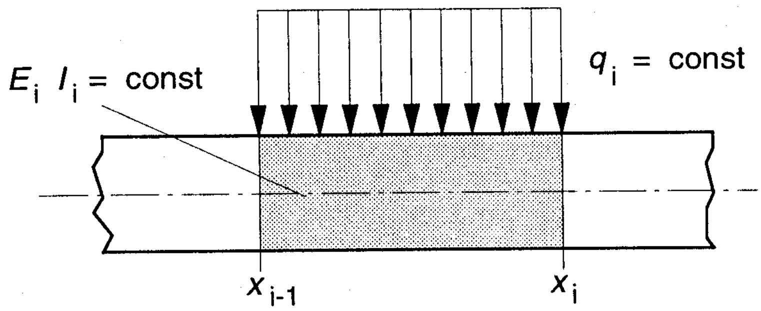


Figure 8.15. Beam section for calculation using transfer matrices

A system of equations describing the values at point x_i as a function of the values at point x_{i-1} can be found by definite integration. The relationships can be easily represented in matrix form, hence the name "transfer matrices". There are some problems when introducing the boundary conditions, but the problem can be reduced to a simple programmable form by overlaying different solutions. Finally, the deflection f, bending angle φ, bending moment M_b and transverse load Q can be calculated at any point along the shaft.

This method also enables statically indeterminate cases to be calculated by specifying appropriate boundary conditions. For example, with a three-bearing shaft the deflection at the three bearings must be set equal to zero.

For comparison with the permissible deformations, the resultant deflections and bending angles must be calculated from the components in both planes:

$$f = \sqrt{f_y^2 + f_z^2} \, , \tag{8.27}$$

$$\tan \varphi = \sqrt{\tan^2 \varphi_y + \tan^2 \varphi_z} \, . \tag{8.28}$$

Toothed gearing systems are very sensitive to shaft deformation; tilting in particular can easily lead to canting of gearwheels or to edge pressure in the bearings. The requirements for permissible deflections and bending angles are correspondingly high (Table 8.4).

Table 8.4. Reference values for permissible deflection and bending angles for shafts of gearwheel transmissions

Shafts	Deflection	Bending angle
General rule for gearwheels	$f_{perm} \leq 0.01\ m_n$ m_n standard module	$\tan \varphi_{perm} \leq \dfrac{2\, d_w}{10^4\, b}$ d_w pitch circle diameter b face width
Reference values for gearing	$f_{perm} \leq$ 0.02–0.06 mm	$\tan \varphi_{perm} \leq$ 0.005 for spur gears $\tan \varphi_{perm} \leq$ 0.001 for bevel gears

8.5 Flow Chart for Designing Transmission Shafts

Figure 8.16 shows a summary flow chart for calculating transmission shafts, referring to the equations and tables in Chapter 8.

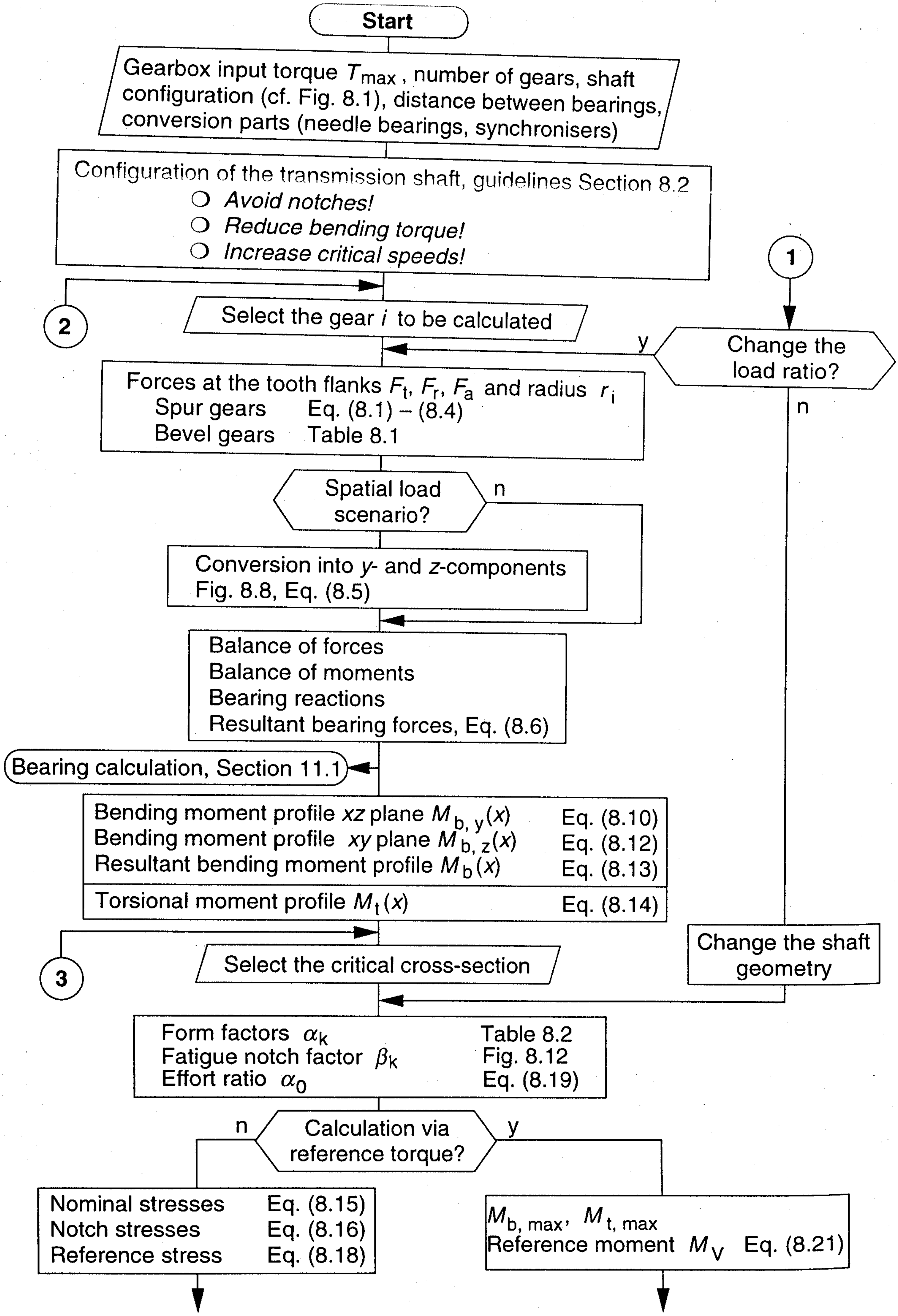


Figure 8.16a. Flow chart for designing transmission shafts

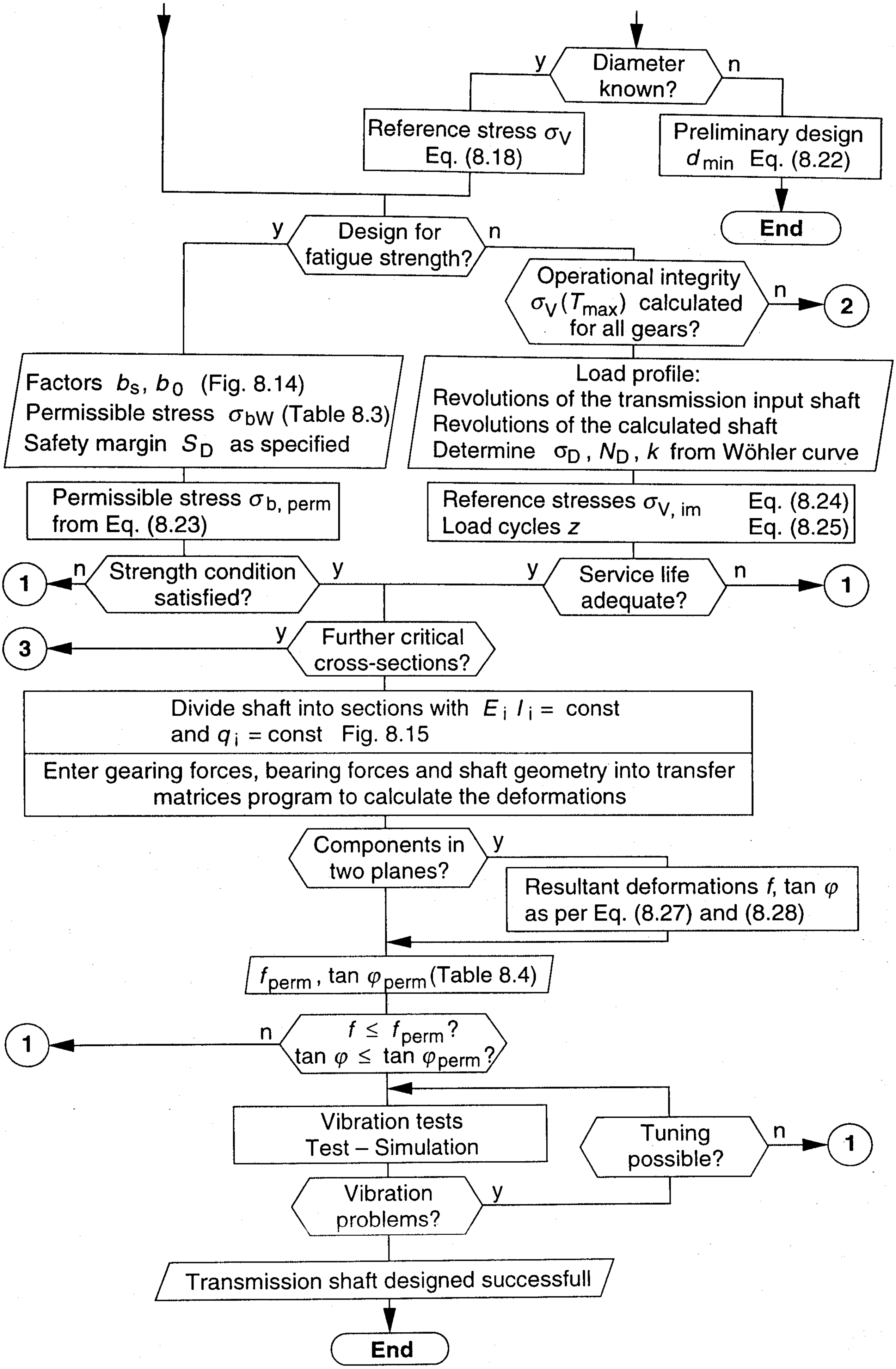


Figure 8.16b. Flow chart for designing transmission shafts

9 Gear Shifting Mechanisms, Layout and Design of Synchronisers

Changing connections

Vehicle transmissions require devices to match the ratio, and thus the power available, to the prevailing driving conditions. "Power matching" is one of the four main functions of a vehicle transmission. In manual gearboxes, changing gear is controlled and carried out by the driver. In fully automatic gearboxes the transmission control unit effects the change of ratio. Semi-automatic gearboxes relieve the driver of these tasks to varying degrees, depending on the degree of automation (see Sections 6.6 and 6.7). In the case of automatic transmissions certain functions, such as Neutral, Reverse, and Park are controlled by the driver using a shift or selector lever. Devices for changing the ratio in continuously variable transmissions are not considered here.

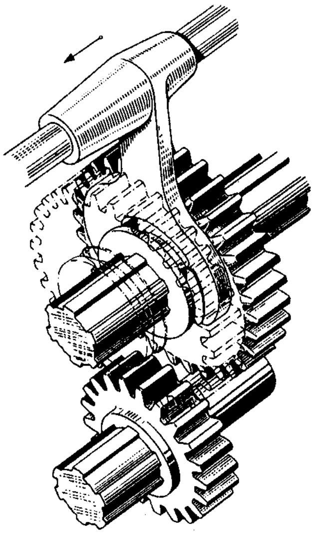
The shifting device plays an important role in the interface between driver and vehicle. It is a major factor determining perceived ease of use. The components used in a shifting device depend largely on whether shifting gear involves interrupting the power flow. See also Section 6.3.1 "Shifting with Power Interruption" and Section 6.3.2 "Shifting without Power Interruption". Other factors are the type of vehicle (passenger car or truck), the type of drive (front-wheel or rear-wheel drive) and the operating conditions. In the following discussion a distinction is made between

- *internal shifting elements*
  Shifting elements inside the transmission, such as selector bars, selector forks, synchronisers, belt brakes, and
- *external shifting elements*
  Shifting elements outside the transmission, such as selector levers, four-bar linkages, turning-shaft remote controls, and cable controls.

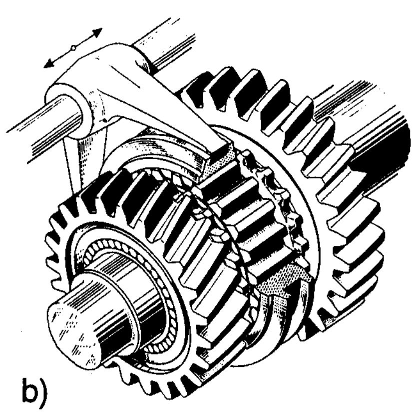
Figure 9.1 shows internal shifting elements for engaging gearwheels into the power train. A distinction is made between form locking clutches (e.g. dog clutch) and frictional clutches (e.g. multiplate clutch).

Since the variety of designs and combinations of internal and external shifting elements is virtually unlimited, only the basic principles are described in this chapter. Sections 12.1 to 12.4 examine some typical examples of existing designs. The bulk of this chapter is devoted to the design and configuration of synchronisers.

Figure 9.1. Internal shifting elements in vehicle transmissions. →
a) Sliding gear; **b)** Dog clutch engagement; **c)** Pin engagement; **d)** Synchroniser without locking mechanism; **e)** Synchroniser with locking mechanism; **f)** Servo lock synchroniser mechanism (Porsche system); **g)** Hydraulically activated multi-disc clutch for power shift transmission; **h)** Hydraulically activated multi-disc brake for planetary gear [9.1]



a)



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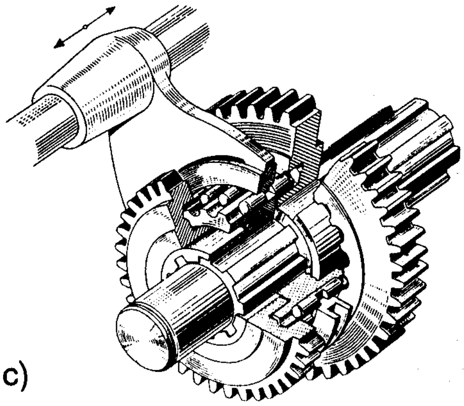

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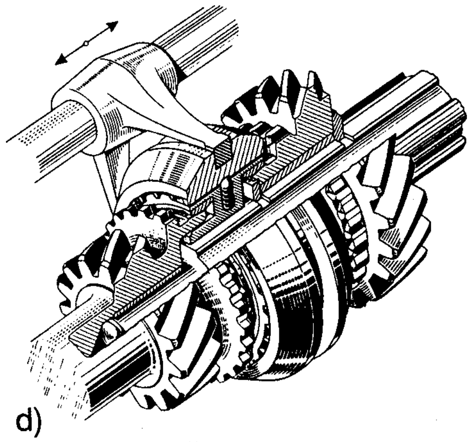

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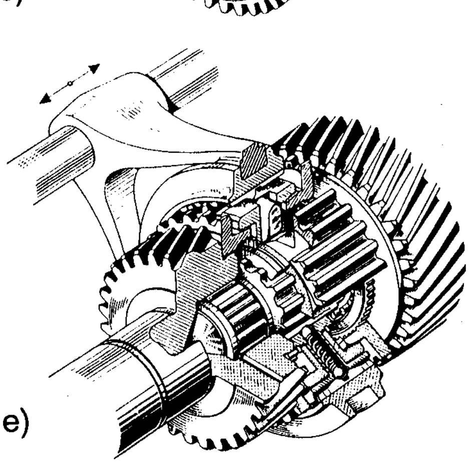
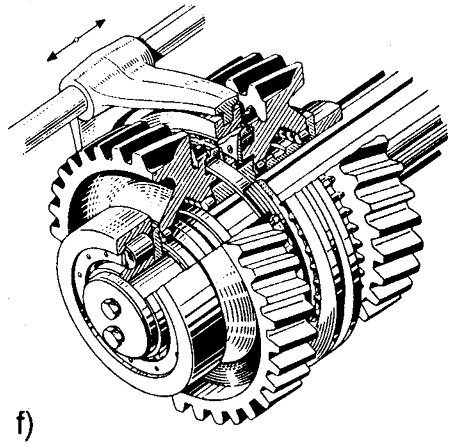


e)



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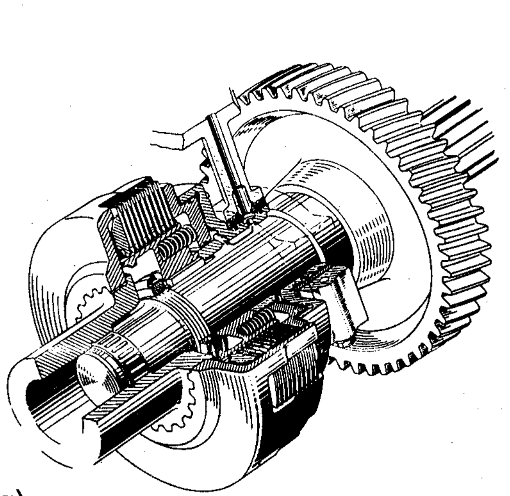
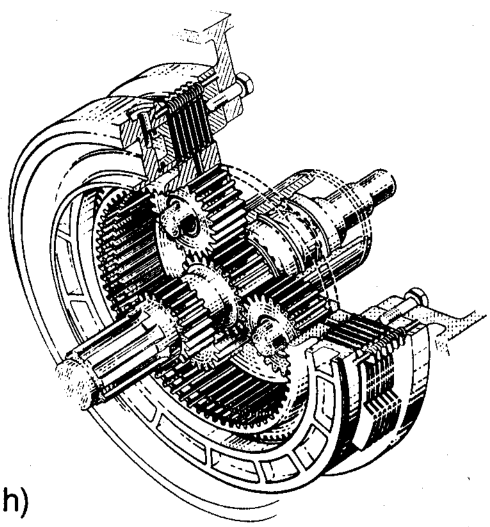


g)



h)

9.1 Systematic Classification of Shifting Elements

The morphological table gives an overview of shifting elements (Table 9.1).

Table 9.1. Morphological table for shifting elements

Parameters	Configurations (shifting elements)				
Generate shifting force	Mechanical, manual effort	Electro-hydraulic	Electro-pneumatic	Electro-magnetic	Spring power
Example	Shift lever	Automatic transmission, Fig. 9.1h	Commercial veh. range unit, Fig. 12.8	Electro-magnetic clutch	"Snapper"
Gear selection	Selector bars, levers	Ball joint, four-bar mechanism	Selector shaft, turning shaft	Control cable	Shift-by-wire
Example	3-bar shift mechanism (MB)	020 gearbox (VW), Fig. 9.4	Fig. 12.2 (ZF) Fig. 12.11 (ZF)	MQ gearbox (VW), Fig. 12.5	AS TRONiC (ZF), Fig. 12.18
Shifting	Selector fork	Gearshift fork	Piston	———	———
Example	Fig. 12.7 (ZF)	Fig. 12.2 (ZF), Fig. 12.5 (VW)	Conv. automatic gearbox, Fig. 12.12 (MB)	———	———
Frictional connection	Single/multi-taper	Spreader ring	Multi-discs	Belt	Sprag
Example	Taper synchroniser	Porsche synchroniser	Clutch, brake, Fig. 6.27	Brake, Fig. 6.24	Freewheel (automatic), Fig. 6.27
Positive locking	Dog	Pins	Sliding gear	Draw key	———
Example	Fig. 9.1b, d, e, f	Fig. 9.1c	Fig. 9.1a	Motorcycle gbox	———

A distinction is made between the following different types of shift mechanism:

- *direct shift mechanism*
  Gearshift lever on the transmission housing (especially common in commercial vehicles),
- *indirect shift mechanism*
  Gearshift lever and transmission are physically separated, "remote shifting";
  - mechanical or cable link,
  - power-assisted shifting (e.g. pneumatic, hydraulic, electric/shift-by-wire).

With the increasing number of semi-automatic gearboxes the indirect shift control systems will become more common. Such "shift-by-wire" systems are predicted to account for about 20% of the market for semi- and fully automatic transmissions in Western Europe by the year 2010 [9.15].

9.1.1 Shifting Elements for Geared Transmissions with Power Interruption

The simplest type of transmission is *sliding gears* (Figure 9.1a). The gearwheels are not constantly meshed but are shifted into the power flow as needed. Sliding gears are used for reverse gear in both passenger car and commercial vehicle transmissions.

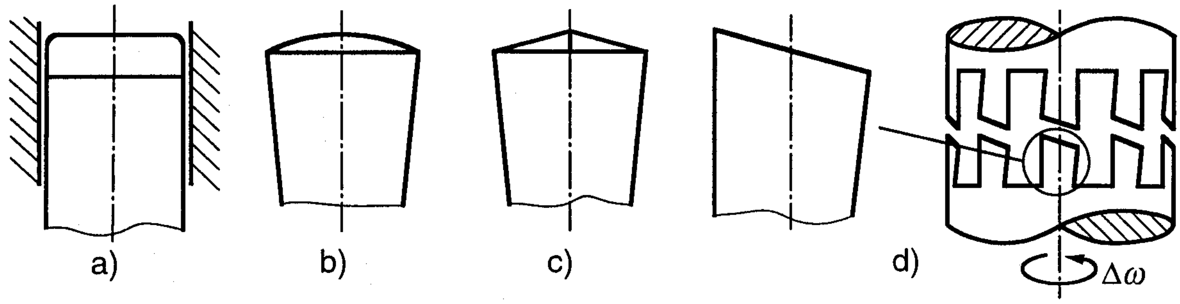


Figure 9.2. Dog shapes in unsynchronised mechanisms. **a)** Fuller dog; **b)** ZF dog; **c)** Berliet dog; **d)** Deflector dog (MAYBACH override clutch): as long as there is relative movement, the bevelled deflector surfaces prevent engagement

Unsynchronised *constant-mesh transmissions* are often found in commercial vehicle transmissions. The constant-mesh gear pairs run on ball and roller bearings and have a form locking connection to the transmission shaft via a sliding dog sleeve (gearshift sleeve) (Figure 9.1b). The gears are prevented from disengaging (gear dropout) by undercut dogs (Figure 9.2).

Gear shifting is always made up of a selecting movement and a shifting movement. The selecting movement selects the gearshift sleeve to be shifted, and the shifting movement moves the gearwheel into the power flow. Figure 9.3 shows an example of this for a *synchromesh gearbox* with direct shifting by three selector bars. The gearshift lever *1* and the ball joint *2* serve to select the gear and transmit the manual effort.

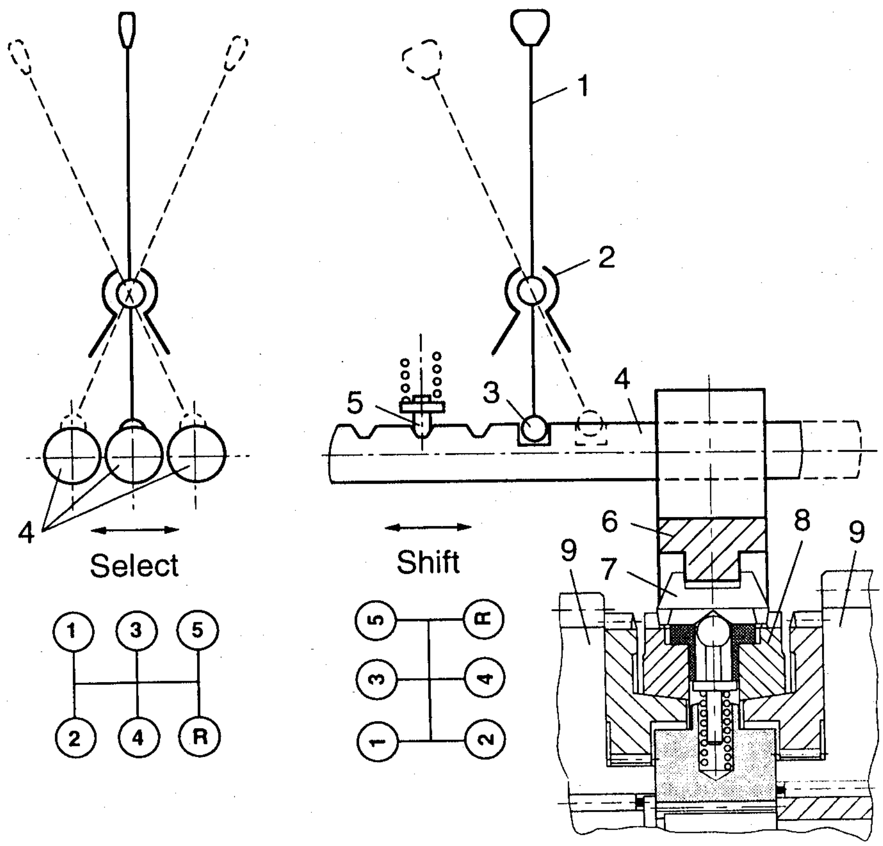


Figure 9.3. Direct shifting of a 5-speed gearbox with three selector bars. *1* Shift lever; *2* Ball joint; *3* Selector finger; *4* Selector bar; *5* Lock; *6* Selector fork; *7* Gearshift sleeve; *8* Synchroniser; *9* Idler

The space where the gearshift lever can move a gearshift sleeve is known as the gate. When a gate is selected, the selector finger *3* of the gearshift lever engages in the grooves of one of the individual selector bars *4*. The selector bar is shifted axially with a longitudinal movement of the gearshift lever, thus changing gear.

A selector fork *6* engages in the gearshift sleeve *7*. Since each selector fork can shift either of two opposing idler gears *9*, three shift positions (two end positions and one middle position) of the selector bar *4* are secured by a lock *5*. The selector forks can both be moved axially as shown, and swivelled around a fixed pivot. This is referred to as a *gearshift fork*. By selecting appropriate lever proportions, gearshift effort can be reduced at the expense of increasing the shift stroke (see also Figure 12.2).

In the case of semi-automatic and fully automatic gearboxes it is essential to ensure that only the gearwheels actually required are in the power flow. Drum selectors for example are used for this purpose in semi-automatic racing gearboxes. The drum selector can rotate in both directions, and guides the selector forks along curved paths. This system is also widely used in motorcycles.

Figure 9.4 shows the external shifting elements, and some of the internal shifting elements of a passenger car with a transverse mounted gearbox. The remote shifting is mechanical, using a four-bar linkage. The shifting arrangement shown is currently used in the VW Golf Mk. III. In the VW Passat "MQ transmission" (Figure 12.5), the four-bar linkage is replaced by a cable remote shift, with a selector cable and a shifting cable.

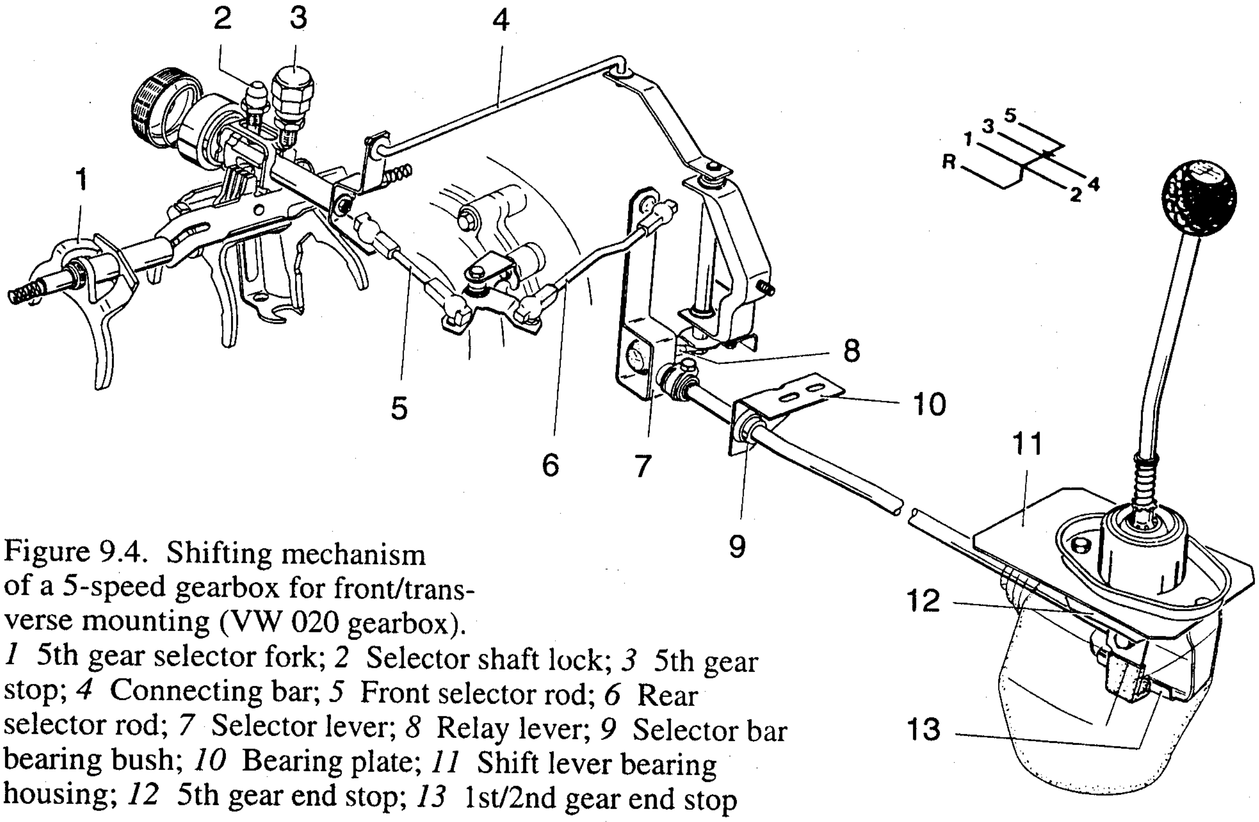


Figure 9.4. Shifting mechanism of a 5-speed gearbox for front/transverse mounting (VW 020 gearbox).
1 5th gear selector fork; *2* Selector shaft lock; *3* 5th gear stop; *4* Connecting bar; *5* Front selector rod; *6* Rear selector rod; *7* Selector lever; *8* Relay lever; *9* Selector bar bearing bush; *10* Bearing plate; *11* Shift lever bearing housing; *12* 5th gear end stop; *13* 1st/2nd gear end stop

In commercial vehicle multi-range transmissions an additional control is needed to shift the gears in the splitter unit or range-change unit. Often a switch is fitted in the grip of the selector lever, which controls a pneumatic valve.

Table 9.2 shows the twelve possible conditions under which shifting may occur, highlighting those that are critical when changing gear.

In heavy commercial vehicles, gear shifting is often servo-assisted to reduce the effort required by the driver; the existing compressed air system is then used to activate the final control elements. The shifting sequence is electronically controlled. Driver effort is reduced to varying degrees depending on the degree of automation of shifting. (See also Table 6.12 "Degrees of automation" and Chapter 13 "Engine and Transmission Management"). In [9.2] a distinction can be made between:

- *electronic transmission control with mechanical clutch activation:*
  - Electronic remote shifting: The external mechanisms are replaced by servo circuits,
  - Sequential shifting mechanism: The driver selects only whether to shift up or down. The electronic control system helps the driver by intelligent intervention,
  - Preselector shift mechanism: The electronic control unit determines the preferred gear, and gives shifting recommendations. The gearshift action is triggered by depressing the clutch pedal.
- *electronic transmission control with automatic clutch activation and engine management:*
  - The entire gearshift process is electronically controlled. The shifting can be done fully automatic or initiated by the driver.

Typical shifting mechanism designs for transmissions with power interruption are shown in Sections 12.1 "Manual Gearboxes" and 12.2 "Semi-Automatic Manual Gearboxes".

Table 9.2. The twelve possible gearshift states. ○ is non-critical and ● is critical as regards changing gear

	Shifting up		Shifting down	
	Power	Overrun	Power	Overrun
Level	○	○	○	○
Uphill	●	○	●	○
Downhill	○	○	○	●

9.1.2 Shifting Elements for Geared Transmissions without Power Interruption

Conventional automatic transmissions, consisting of torque converters and planetary gears, are shifted without interruption of the power flow, as are the fully automatic countershaft type geared transmissions used in passenger cars. In powershift transmissions the gear to be shifted is frictionally connected to the transmission shaft. The belt brake, multi-disc brake, multi-disc clutch and freewheel shifting elements are examined in detail in Section 6.6.3 "Fully Automatic Passenger Car Transmissions". Existing designs are presented and discussed in Section 12.3.

Torque is transmitted by friction in clutches and brakes; the dynamic friction coefficient μ between the engaged surfaces therefore has a great influence on the behaviour of the system and thus on ease of use. The following factors affect the friction coefficient:

- running speed,
- temperature of the friction surface,
- type of friction lining,
- type of oil and additives present.

Special organic oils, known as automatic transmission fluids (ATF), have been developed for use in automatic transmissions.

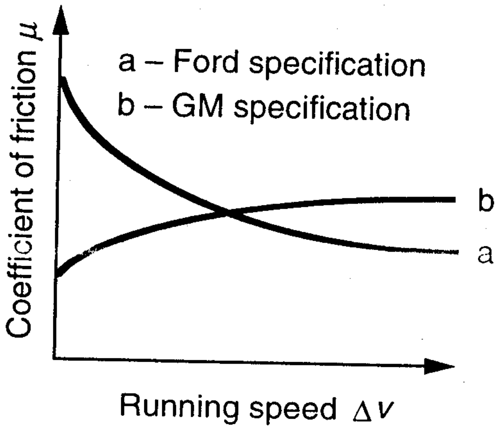


Figure 9.5. Effect of different automatic transmission fluids on the frictional characteristics of clutches or brakes in automatic transmissions

The frictional behaviour, and thus the torque transmission characteristics, can be significantly altered by the type of friction lining selected and the fluid used. Figure 9.5 shows the friction coefficient profile for two different types of fluid. When using fluid type *a*, the friction coefficient decreases as running speed increases; here the coefficient of adhesion is greater than the coefficient of sliding friction. In contrast, the additives in fluid type *b* result in an increase of the dynamic friction coefficient μ with running speed.

In the GM specification, the coefficient of adhesion is less than the coefficient of sliding friction. The vehicle manufacturer selects the characteristics of the lubricant according to the type of engine behaviour required.

Gear changing in powershift transmissions demands detailed engineering of the components involved and the tribological conditions as well as of the software (control algorithm). The reader is accordingly referred to the relevant literature [9.3].

9.1.3 Parking Lock

When the engine is switched off, vehicles with a manual transmission can be kept stationary by engaging a gear with a high ratio in addition to applying the hand brake. This option is not available in vehicles with a torque converter, since there is no link between the vehicle and the braking power of the engine. Vehicles with torque converters are therefore equipped with a "parking lock" in order to ensure they can remain stationary even under extreme conditions.

Parking locks prevent unintentional movement of the vehicle. This is achieved by locking the transmission output shaft, which is rotationally fixed to the drive wheels. The main requirements for the design of such a locking system are:

- Prevent rolling down gradients up to 30% approx.
- Safety function: inhibit locking at speeds $v \geq 3$ km/h.

The locking action is initiated by the driver moving the selector lever to engage the park position. The transmission shown in Figure 12.2 will be used as an example to illustrate the design and function of a parking lock in a conventional automatic transmission. The parking lock shown in Figure 9.6 has a radial locking pawl. Moving the selector lever *1* to the park position *4* has the following effects:

1/ The notched plate *3* rotates about the axle *2* in the same direction as the attached selector lever, until the roller on the bending spring *5* engages in the park position *4*.

2/ The push/pull rod *7* linked to the notched plate, moves the roller *12* (which runs on it) in the guide *13* parallel to the output shaft *8*.

3/ At the end of the guide, the roller runs over a roller bearing *14* fixed in the housing, pressing upwards against the sloping back of the pawl *11*. This moves upwards against the resistance of its return spring *10* as far as the parking lock wheel *9*, which is rotationally fixed to the output shaft.

4/ When the vehicle is at rest, or moving at a speed of less than 3 km/h, the pawl engages in a gap in the parking lock wheel, positively locking the driving wheels and preventing the vehicle from moving.

5/ When the vehicle is moving at a speed greater than 3 km/h, the compression spring *6* mounted on the push/pull rod is stretched, since the flank angle at the pawl and the ratchet wheel prevents the pawl from engaging. This "ratcheting" continues as long as the vehicle is moving faster than the critical speed of 3 km/h. As soon as the speed falls below this limit, the pawl engages the ratchet wheel and prevents further movement of the vehicle.

6/ The pawl is disengaged when any other automatic gear position is selected. The roller *12* moves back into the guide and frees the pawl to move downwards and out of the toothing of the parking gearwheel. This disengagement motion is assisted by the return spring *10*.

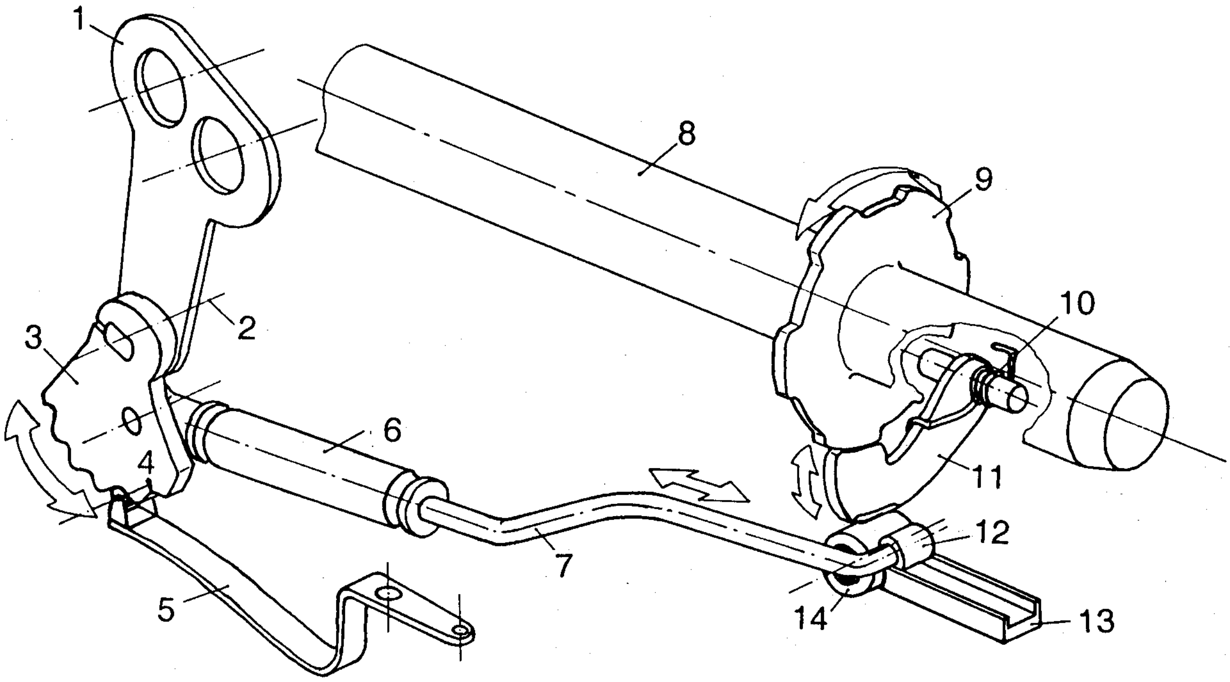


Figure 9.6. Parking lock with radially engaging locking pawl for automatic transmission with torque converter. *1* Automatic selector lever; *2* Switch shaft; *3* Notched plate; *4* Park position; *5* Bending spring; *6* Compression spring; *7* Push/pull rod; *8* Output shaft; *9* Parking lock wheel; *10* Return spring; *11* Pawl; *12* Roller; *13* Guide; *14* Roller bearing

9.2 Synchroniser Functional Requirements

This section deals with the transmission synchroniser, the most important internal shifting element. *Synchromesh gearboxes* shift with power interruption. All passenger cars with manual gearboxes have synchromesh. In 1993 approximately 60% of commercial vehicles were fitted with synchromesh gearboxes, to improve road safety (a gear can be engaged at any time) and ease of use. The life of the synchroniser becomes critical in determining the system service life in the case of large transmissions, where there are high input torques and large masses to be synchronised.

Rotating shifting dogs can only be positively locked without "grating" if they have the same circumferential speeds. A synchronising mechanism is therefore required to match the circumferential speeds of the parts to be connected in 0.1 to 0.3 seconds with the application of a minimum of force, and to prevent premature locking by blocking the shift movement.

A gearwheel transmission with multi gears may be synchronised in the following ways [9.4]:

- synchronising mechanism for each individual gear,
- central synchroniser for the whole transmission (Section 9.7),
- speed synchronisation by the prime mover (Section 9.7).

It is technically possible to dispense with synchronisers when

- there is a small gear step between the gears ($\varphi < 1.15$) or
- the masses of the gears are small, for example in a motorcycle gearbox.

Synchromesh is frequently omitted in commercial vehicles for reasons of economy and to improve transmission reliability. Transmissions without synchromesh are more robust. This is an important factor, especially in third-world countries.

A mechanical synchromesh unit as shown in Figure 9.7 frictionally matches the different speeds of the transmission shaft *6* (and the gearshift sleeve *5* rotationally fixed to it) and of the idler gear to be shifted *1*. When their speeds have been synchronised, the elements are positively engaged. This synchromesh unit incorporates a frictionally engaged clutch and a positive locking clutch (see also Figure 4.3 "Systematic classification of master clutches").

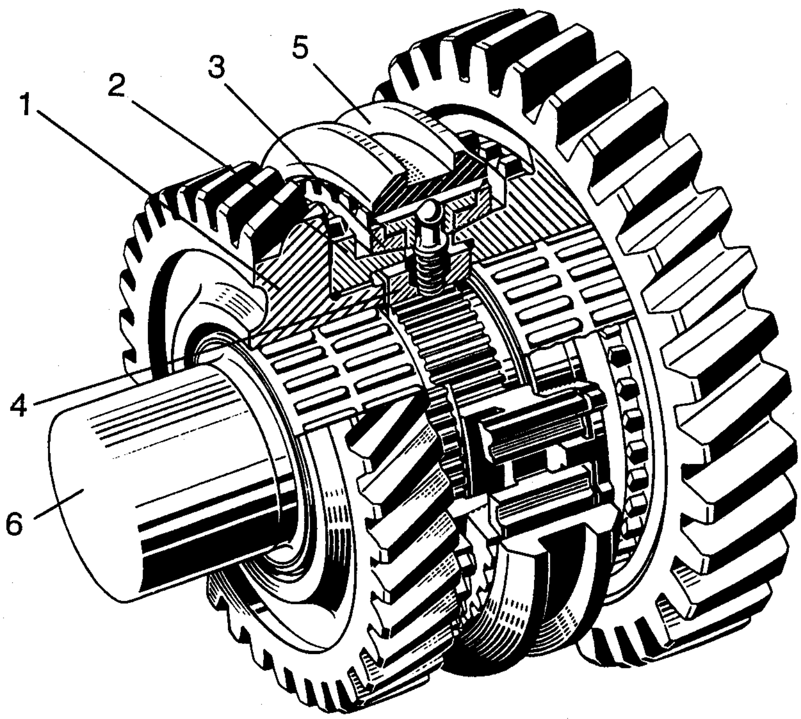


Figure 9.7. Single-taper synchroniser (ZF-B), see also Figure 9.12.

1 Idler gear with needle bearings
2 Synchroniser hub with selector teeth and friction taper
3 Synchroniser ring with counter taper and locking toothing
4 Synchroniser body with internal toothing for positive locking with the transmission shaft and constant-mesh external gearing for the gearshift sleeve
5 Gearshift sleeve with constant-mesh internal gearing and ring groove
6 Transmission shaft

9.2.1 Changing Gear

This section describes the gear changing process using as an example a notional vehicle with a 2-speed coaxial countershaft transmission (Figure 9.8).

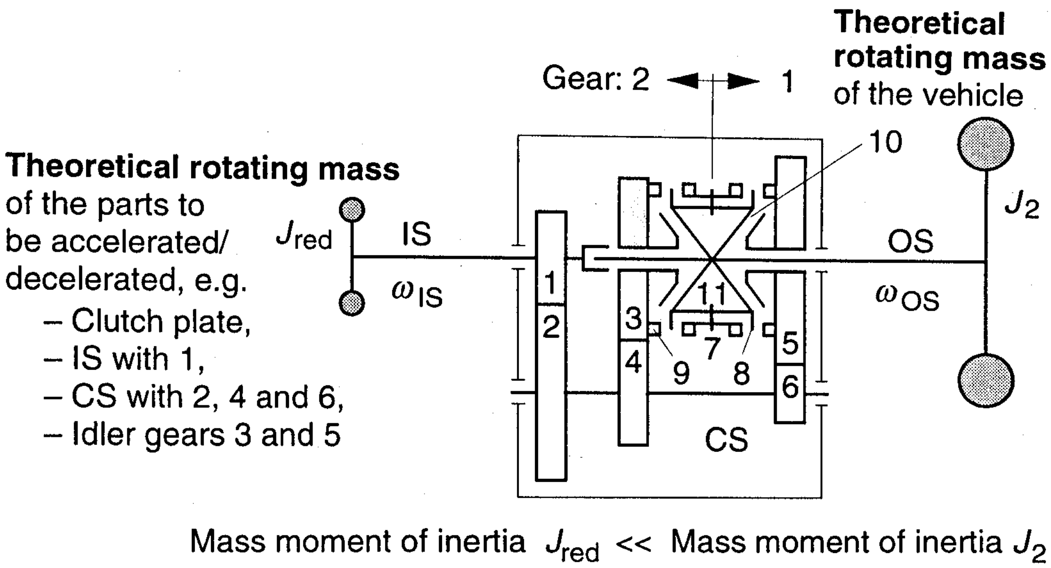


Figure 9.8. Gear changing process. *1, 2, 4, 6* Fixed gears; *3, 5* Idler gears; *7* Gearshift sleeve with dogs; *8* Locking mechanism; *9* Selector teeth; *10* Friction surfaces (taper and counter-taper); *11* Synchroniser body; *IS* Input shaft; *OS* Output shaft; *CS* Countershaft

As the speed of the vehicle v in second gear drops, there is a particular angular velocity ω_{IS} at the input shaft *IS*. When the master clutch is fully engaged, $\omega_{IS} = \omega_M$. It is not possible to shift down into first gear until ω_M in first gear after shifting is less than $\omega_{M,\,max}$ (see also Figure 4.11 "Velocity/engine-speed diagram").

The moment of inertia J_2 of the vehicle is significantly greater than the combined moment of inertia J_{red} of the masses to be synchronised. The fall in angular velocity of the output shaft *OS* during the shifting period (slipping time t_R) may thus be ignored initially, i.e. ω_{OS} = constant. This simplification is no longer acceptable for more precise models, for example changing gear on a hill.

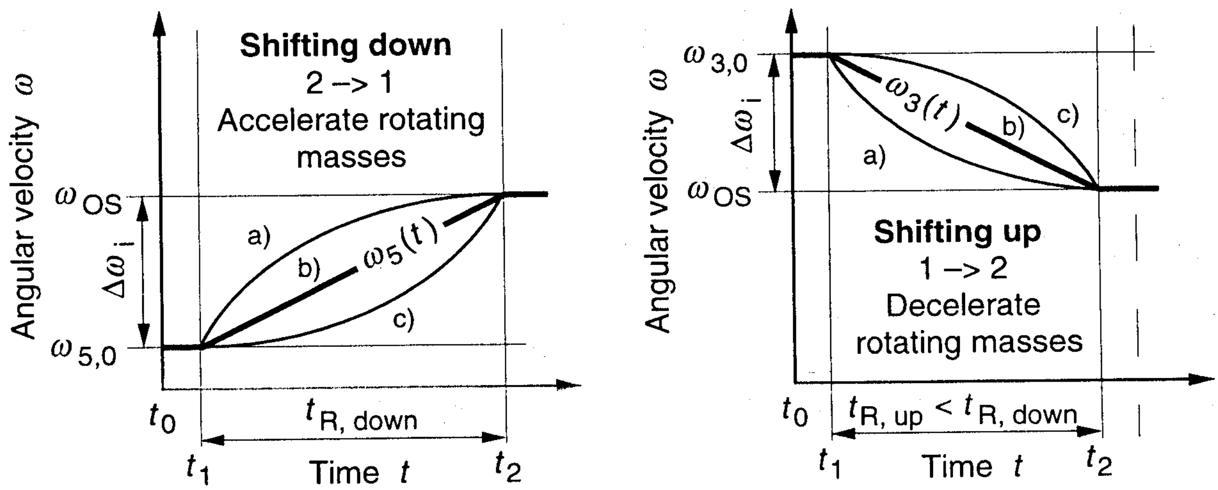


Figure 9.9. Basic angular velocity curve during synchronisation. ω increases or decreases depending on the gear-shift effort and coefficient of friction curve according to a particular law. Ideal ω curve: **a)** degressive; **b)** linear; **c)** progressive

The shifting process is initiated at time t_0 (Figure 9.9). The gearshift sleeve *7*, rotationally fixed to the output shaft, rotates with an angular velocity ω_{OS} and the idler gear *5* to be engaged rotates with an angular velocity $\omega_{5,0}$. The angular velocity difference to be adapted is $\Delta\omega_i = \omega_{OS} - \omega_{5,0}$ (Figure 9.9). After the response delay $t_1 - t_0$ the synchronisation process starts at time t_1. The idler gear *5* and the masses connected to it are accelerated when shifting down. The angular velocity ω_5 of the idler gear *5* to be shifted increases according to a certain function until a velocity ω_{OS} is reached, when the idler gear is synchronised with the gearshift sleeve *7*.

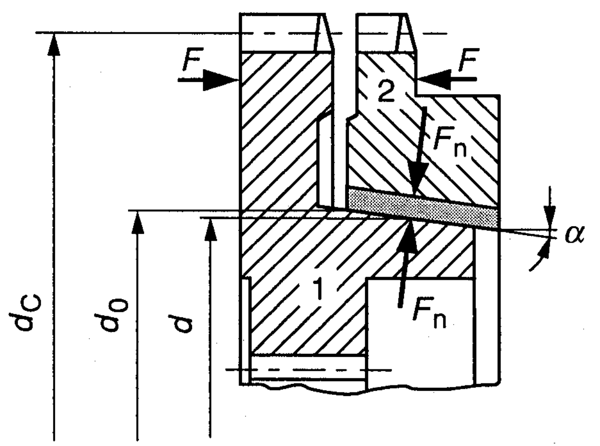
During the slipping time $t_R = t_2 - t_1$, the friction surfaces *10* slip at a relative angular velocity $\omega_{rel} = \omega_{OS} - \omega_5$. A similar result applies when shifting up from first to second gear. When the angular velocities are synchronised, a locking device *8* clears the shift movement and the gearshift sleeve *7* may be positively connected to the selector teeth *9* without "ratcheting".

9.2.2 Main Functions and Ancillary Functions

Table 9.3 shows the main functions and ancillary functions of synchronisers, along with possible mechanical solutions.

9.2.3 Speed Synchronisation with Slipping Clutch

The friction surfaces of mechanical synchronisers involved in speed synchronisation are of flat, conical or cylindrical design. Systems using friction tapers are common in both passenger car and commercial vehicle transmissions (Figure 9.10). The gearshift effort applied by the driver through the gearshift lever, selector fork and gearshift sleeve is amplified by a taper.



F	Gear shift effort
F_n	Normal force
α	Taper angle
j	Friction surfaces
d	Effective diameter
d_0	Nominal diameter
d_C	Clutch diameter
1	Clutch body
2	Synchroniser ring with counter friction surface

Single-taper	Multi-taper	Multi-disc
$j = 1$	$j \geq 2$	$j \geq 2$
α = 6–7°	α = 9–12°	α = 90°

Figure 9.10. Common formats of mechanical synchronisers, dimensions

Table 9.3. Main and ancillary functions of synchronisers

Main functions	**Comments**	**Mechanical solution**
1/ Adapt speed, accelerate or decelerate masses	– Low slipping time t_R, Table 9.4	Internal energy transfer, using the energy accumulator J_2 (motor vehicle), power flow through friction clutch
2/ Measure speed difference, determine synchronous speed	– Reliable functioning under all operating conditions	Speed comparison using friction, as a function of relative speed
3/ Locking the positive engagement until speeds are synchronised	– Forcing the gears before speeds are synchronised should be difficult or impossible	Friction lock mechanism with differential speed dependent effect
4/ Establish positive engagement and power flow	– Shift stroke s as short as possible – Ensure positive locking, prevent gear dropout	Dog clutch with undercut toothing
Ancillary functions	**Comments**	**Quantification**
5/ Ease of use	– Shifting force profile – Shifting force cycle	– Table 9.4
6/ Reliable operation under all conditions	– Low temperatures – Quick shifting	– Arctic temperatures – $t_R \leq 0.1$ s
7/ Overload capacity	– Operator error	– Abuse test
8/ Service life	– Adequate mechanical and thermal specification	– Pass. car > 150 000 km – Com. veh. > 800 000 km
9/ Efficiency	– Synchronisable masses – Slipping time – Performance limits	– Design – Permissible stress values, Table 9.7
10/ Costs	– Development/production – Replacement of wearing parts	———
11/ Weight/space constraints	– Reduce space required – Shorten shift stroke	———

The following section concentrates on taper synchronisers. Taper synchronisers may be divided into

- *inner taper synchronisers:*
  - single-taper synchronisers, e.g. Borg-Warner system,
  - multi-taper synchronisers,
- *outer taper synchronisers.*

Synchronisers using friction tapers are a special case of friction clutches with smooth friction surfaces, so the same theoretical basis applies. The normal force F_n on the friction surfaces is derived from the gearshift effort F as

$$F_n = F \frac{1}{\sin \alpha} . \tag{9.1}$$

The friction torque T_R is derived from the gearshift effort F and the coefficient of dynamic friction μ

$$T_R = j\, F \frac{d}{2} \frac{\mu}{\sin \alpha} , \tag{9.2}$$

where j is the number of friction surfaces and $d/2$ the effective radius. The simplification $d/2 = d_0/2$ is frequently used in practice. In order to prevent the tapers from self-locking the following must apply for the taper angle α:

$$\tan \alpha > \mu . \tag{9.3}$$

Equation 9.2 provides some starting points for design measures to increase efficiency and reduce gearshift effort. Multi-taper synchronisers accordingly offer less gearshift effort, or greater torque capacity, than single-taper systems. Multi-disc synchronisers have smooth friction surfaces. There is no shift effort adaptation as there is with taper systems. The power handling capacity increases, and the shift effort decreases with the number of friction surfaces, but the overall length of the unit increases.

9.2.4 Synchroniser Dimensions

Figure 9.11 shows the main dimensions of a synchroniser. The wear on the friction surfaces is usually the factor that determines service life. The shift movement s at the gearshift sleeve is approximately 10–13 mm. The permissible wear ΔS_{perm} is generally between 1 and 1.5 mm. The wear reserve of the synchroniser unit is calculated by subtracting the operating clearance from the permissible wear. The maximum wear ΔV_{max} is of the order of 0.15 mm for taper synchronisers.

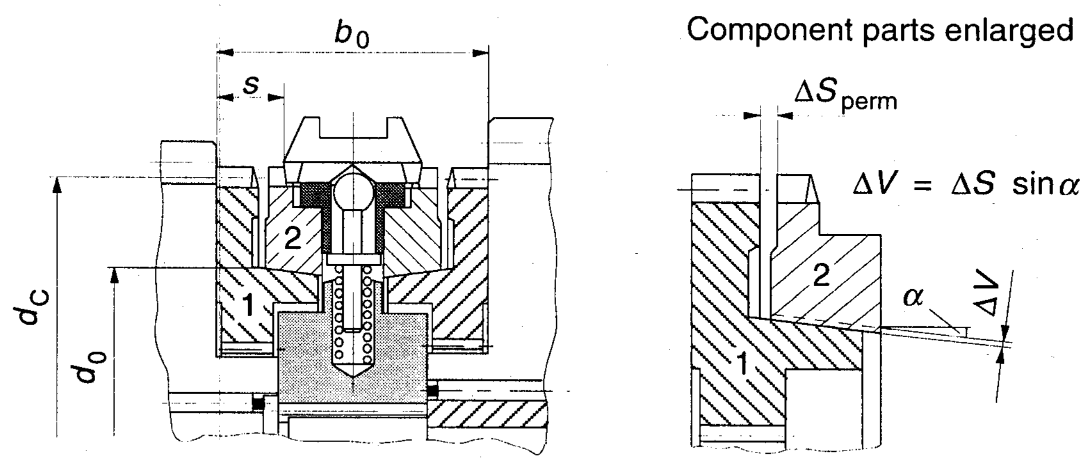


Figure 9.11. Dimensions. b_0 Overall pack length; d_0 Nominal diameter; d_C Clutch diameter; ΔS Wear path; ΔS_{perm} Permissible wear path incl. clearance; s Shift stroke at the gearshift sleeve; ΔV Wear at the synchroniser ring; α Taper angle

9.3 The Synchronising Process

Single-taper synchronisers based on the "Borg-Warner" system are widely used in manual transmissions. The various phases of the synchronising process are shown in Figure 9.12, based on the ZF-B synchroniser ("B" standing for Borg-Warner system).

The synchroniser body *4* is fixed to the transmission shaft. The synchroniser ring *3* is guided by stop bosses in the synchroniser body. These are narrower than the grooves in the synchroniser body, which allows the synchroniser ring a certain amount of freedom to twist radially.

Before the shifting process starts, the gearshift sleeve is held in the middle position by a detent. The gearshift force F triggers the axial movement of the gearshift sleeve *8*, which causes the thrust pieces *7* to act on the ball pins *6* to press the synchroniser ring *3* with its counter-taper against the friction taper of the synchroniser hub *2*. The speed difference between the gearshift sleeve *8* and the synchroniser ring *3* relative to the idler gear *1* causes the synchroniser ring to turn until the dogs engage in the groove wall. This first phase of the synchronising process is known as "asynchronising" (Figure 9.13).

The gearshift sleeve is moved further. This brings the bevels of the constant-mesh internal gearing of the gearshift sleeve *8* and the constant-mesh external gearing of the synchroniser ring *3* into contact. The main synchronisation action starts, Phase II. The gearshift force is applied to the synchroniser ring via the thrust pieces *7* and the dogs *8*, the force being divided between them. The gearing torque T_Z arising at the bevels acts so as to open the locking device. T_Z is smaller than friction torque T_R that acts to close the locking device. In the slipping phase the gearshift sleeve cannot be shifted. In the literature the gearing torque T_Z is frequently referred to as the index torque T_I, and the friction torque T_R as the taper torque T_C.

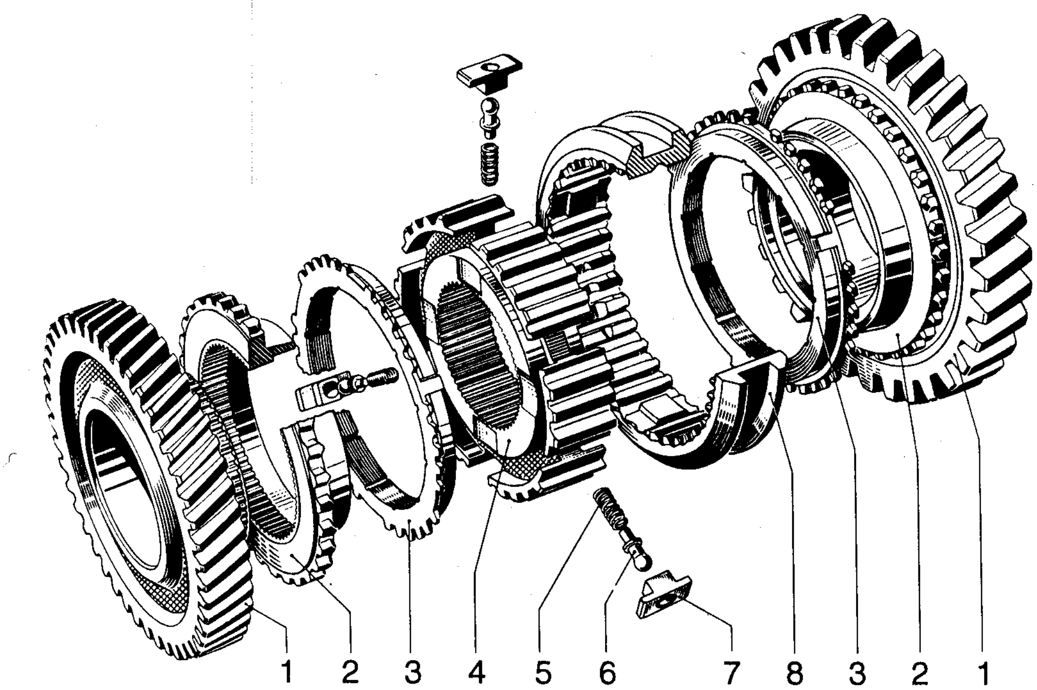


Figure 9.12. Borg-Warner system single-taper synchroniser (ZF). *1* Idler gear running on needle bearings; *2* Synchroniser hub with selector teeth and friction taper; *3* Main functional element, synchroniser ring with counter-taper and locking toothing; *4* Synchroniser body with internal toothing for positive locking with the transmission shaft and external toothing for the gearshift sleeve; *5* Compression spring; *6* Ball pin; *7* Thrust piece; *8* Gearshift sleeve with constant-mesh internal gearing

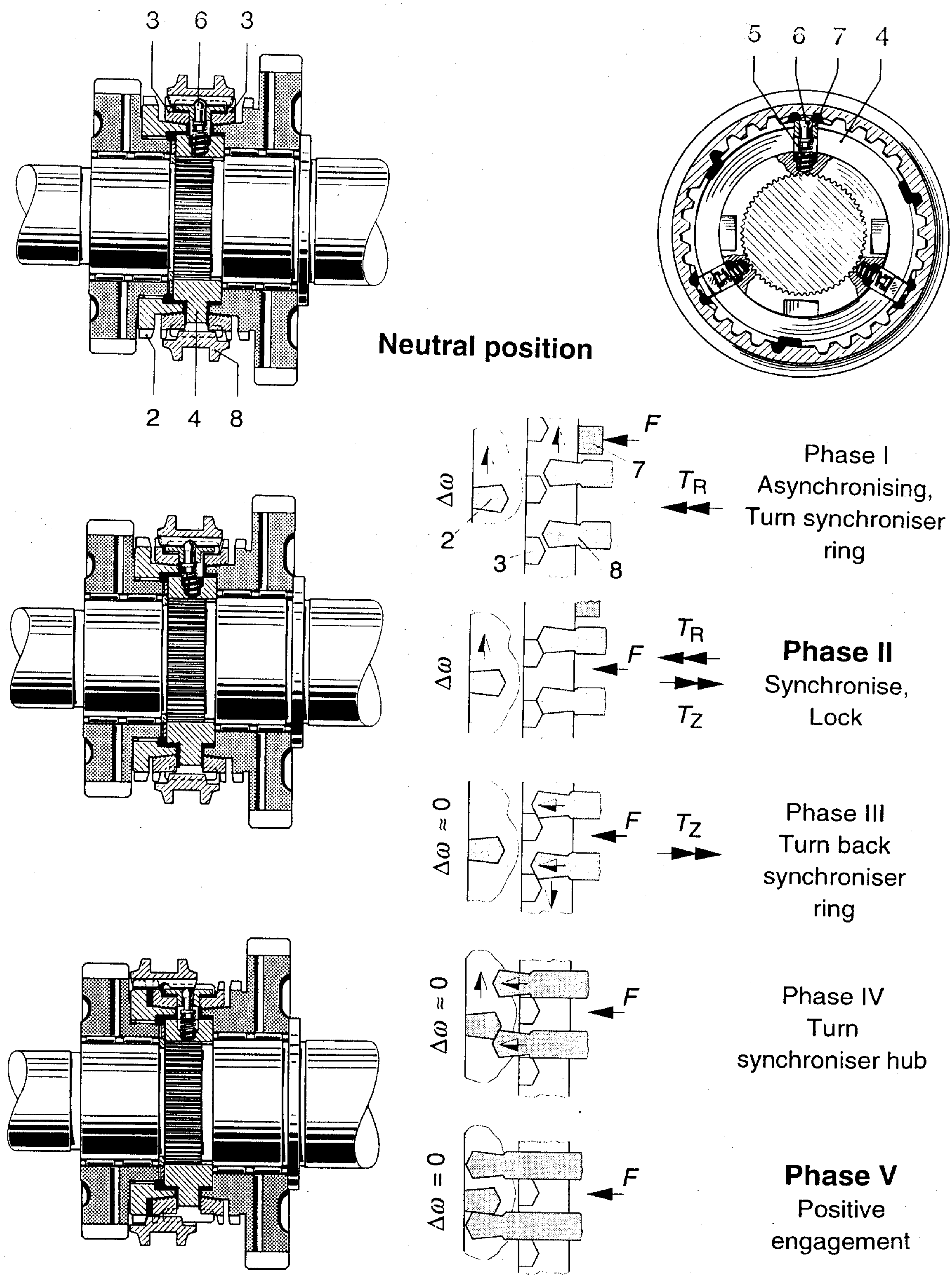


Figure 9.13. Synchronising process. Arrows with half-filled tips indicate the direction of movement, the torque arrows indicate the moments acting on the synchroniser ring. *2* Synchroniser hub with selector teeth and friction taper; *3* Main functional element, synchroniser ring with counter-taper and locking toothing; *4* Synchroniser body with internal toothing for positive locking with the transmission shaft and external toothing for the gearshift sleeve; *5* Compression spring; *6* Ball pin; *7* Thrust pieces; *8* Gearshift sleeve with dog internal toothing

When speed synchronisation has been achieved, the friction torque tends towards zero, Phase III. The unlocking process starts. The gearing torque becomes greater than the friction torque, and acts via the bevels to return the synchroniser. The gearshift force decreases rapidly in this phase. Throughout the axial movement of the gearshift sleeve, the spring-loaded ball pin slides along the inclined grooved surface. This presses it against the spring *5* into the thrust piece, until it is covered by the gearshift sleeve.

As the gear is engaged, the gearshift sleeve toothing encounters the bevels of the selector teeth of the synchroniser hub *2*. In this Phase IV the ball pin is covered. The synchroniser ring is pressed against the friction taper of the synchroniser hub only by residual pressure via the thrust pieces. This residual pressure arises from the friction between the moving gearshift sleeve and the thrust pieces (with ball pins). The gearshift sleeve toothing twists the synchroniser hub relative to the synchroniser ring. The shift movement is enabled. The gearshift sleeve positively engages the power flow between the gear pair and the transmission shaft, Phase V.

9.3.1 Ease of Use

Proper sequential co-ordination of the above functions is important to ensure that the shifting and synchronising processes are as easy to use as is required. The precise design of the parts involves a great deal of know-how, for example in the design of locating faces and in determining functional clearance.

Ease of use is especially important when drivers are evaluating a gearshift system. In mechanically operated shifting devices the manual force applied by the driver is transmitted by the gearshift lever and transmission element (e.g. shifting linkage) to the gearshift sleeve. The transmission ratio of this linkage depends on the engineering design, and normally varies between 7:1 to 12:1. The efficiency with which gearshift effort is transmitted must also be taken into account – it is frequently less than 70%. There are standard values for the maximum permissible slipping time $t_{R,\,perm}$ and the maximum permissible manual effort $F_{H,\,perm}$ (Table 9.4).

Table 9.4. Standard values for acceptable manual effort $F_{H,\,perm}$ and slipping time $t_{R,\,perm}$

Standard value	**Gear**	**Passenger car**	**Commercial vehicle**		
			Main gearbox	Splitter unit	Range-change unit
Permissible manual effort $F_{H,\,perm}$	1–z	< 120–80 N	< 250–180 N	Pneumatic	Pneumatic
Permissible slipping time $t_{R,\,perm}$	1–z	< 0.25–0.15 s	< 0.4–0.25 s	0.15 s	0.2 s

Figure 9.14 gives an example illustrating the relationship between gearshift effort F at the gearshift sleeve and the slipping time t_R. The gearshift effort actually applied by the driver depends very much on driving style and traffic conditions. Very low external temperatures have a major impact on the manual effort required and slipping time. However, external temperature has little effect on the overall ease of operation, since the gearbox oil warms up relatively quickly.

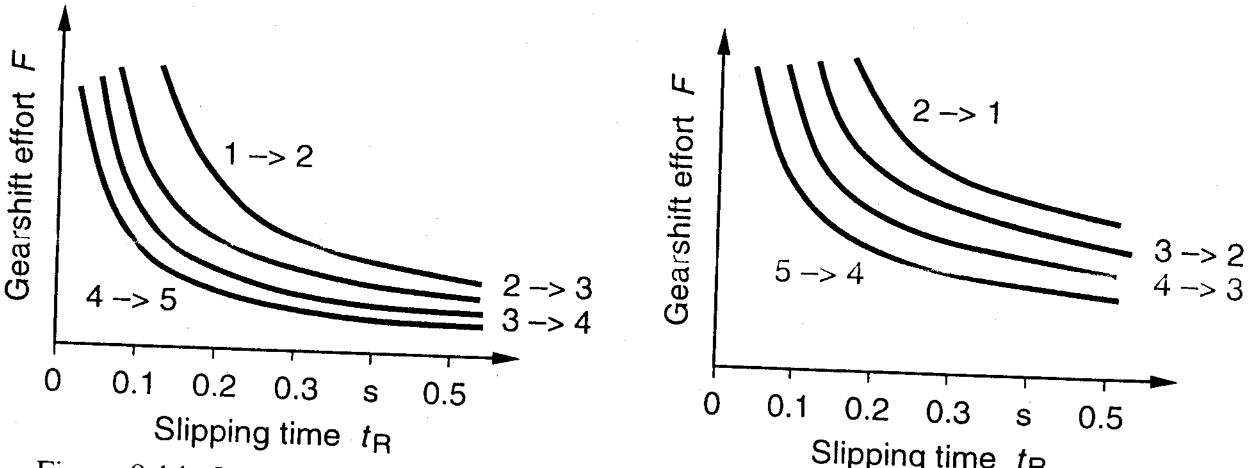


Figure 9.14. Interrelationship between shifting effort F at the gearshift sleeve and the slipping time t_R as a function of the gear selected

The main problems synchronisers cause for the operator arise from

1/ sticking,
2/ upshift grating,
3/ shifting noise.

1/ Sticking

Once the gearwheel speeds have been synchronised and the synchroniser unit has unlocked, the gearshift effort required from the driver drops noticeably before the gear is engaged. The gearshift sleeve should now twist the synchroniser hub with little effort (in Phase IV, Figure 9.13), and it should be possible to push it easily into the engaged position. Friction is generated between the gearshift sleeve and the thrust pieces (with ball pins) when they slide. If this friction is high or the clearance characteristic is unfavourable, the residual pressure on the synchroniser ring can be so high that a large gearshift effort is required to turn the synchroniser hub relative to the synchroniser ring. The driver perceives the new increase in gearshift effort as the gearshift mechanism sticking. The term "clearance characteristic" refers to the synchroniser ring detaching itself from the friction taper of the synchroniser hub.

There are engineering design measures that can be taken to counteract sticking. For example the clearance characteristic can be improved by having an "unwinding" thread in the synchroniser ring friction surface (see also Section 9.5).

2/ Upshift Grating

Upshift grating is typical of gear shifting problems that occur at low temperatures (it is often referred to as "cold scraping"). It occurs especially when the gearbox oil is cold and when shifting from first to second gear. It generally no longer occurs at temperatures above 10 °C.

During the transition from Phase III to Phase IV, after unlocking, the gearshift sleeve moves along a certain path before the gearshift sleeve and synchroniser hub are positively engaged. During this phase the ball pin is covered. Only the residual pressure now presses the synchroniser ring against the synchroniser hub. During this period the idler gear is relatively free. The drag torque can cause a speed difference to return between the gearshift sleeve and the selected idler gear, which causes the dogs to grate when the gear is engaged.

3/ Shifting Noise (Grating)

If the synchroniser does not function properly, the gearshift sleeve can engage in the selector teeth before the speeds have been synchronised. This then causes grating or scraping noises. Whether or not these noises occur depends largely on the driver's gear shifting action. Grating can occur when the driver forces the gear to engage too quickly. The slipping time is too short and the gearshift sleeve and the synchroniser hub are positively engaged before the speeds have been synchronised.

But grating may also be caused by torsional shake, made possible especially by circumferential backlash in the power train. The vibration excitation alters the coefficients of friction. This "shaking action" facilitates the movement of the gearshift sleeve dogs on the bevels of the locking teeth [9.5] (see also Section 9.4.4 "Designing Locking Toothing for Locking Effect").

9.4 Design of Synchronisers

Synchronisers are subject to high levels of stress. This applies particularly to commercial vehicle synchronisers. Figure 9.15 shows the factors affecting its functioning and service life. A single operator error may permanently damage or destroy the synchroniser.

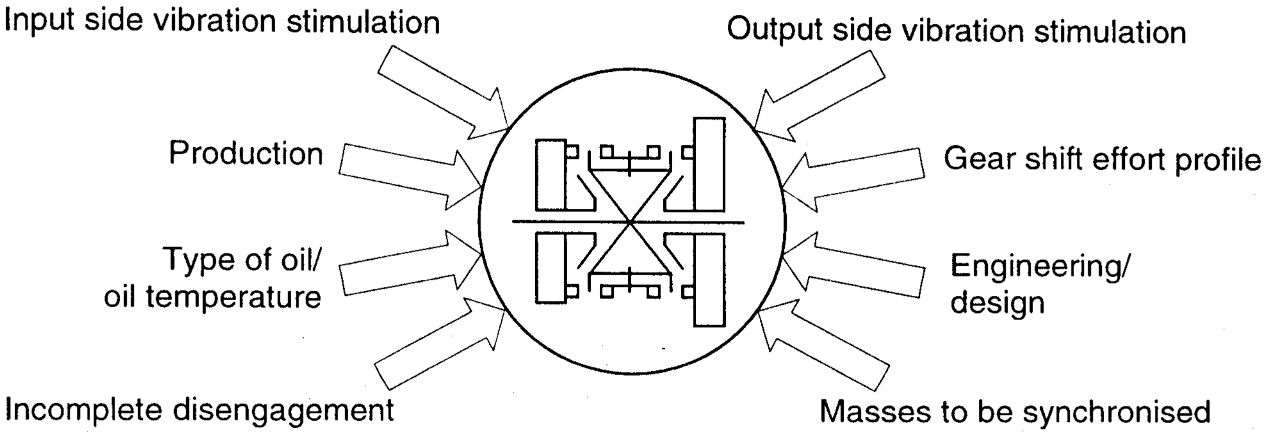


Figure 9.15. External factors affecting the function and service life of synchronisers

The principal criteria according to which synchronisers are designed are the following:

- *function:*
  - synchronisable masses, ease of use,
- *service life:*
  - mechanical stress on the selector teeth,
  - mechanical stress on the synchroniser ring,
  - thermal stress on the friction surfaces,
  - nominal service life (see Table 9.3).

9.4.1 Synchroniser Performance Limits

Figure 9.16 shows a dog gear of a commercial vehicle synchroniser hub that has been damaged by grating. There is a ring groove in the case-hardened steel (16 MnCr5 Eh) of the friction taper as "drainage" to cut through the oil film (see also Section 9.5 "The Tribological System").

Figure 9.17 shows a fracture at the stop boss of a synchroniser ring made of a special brass. Such fractures are the consequence of torsional vibration. They occur mostly in engines that run very unevenly (e.g. direct injection diesel engines).

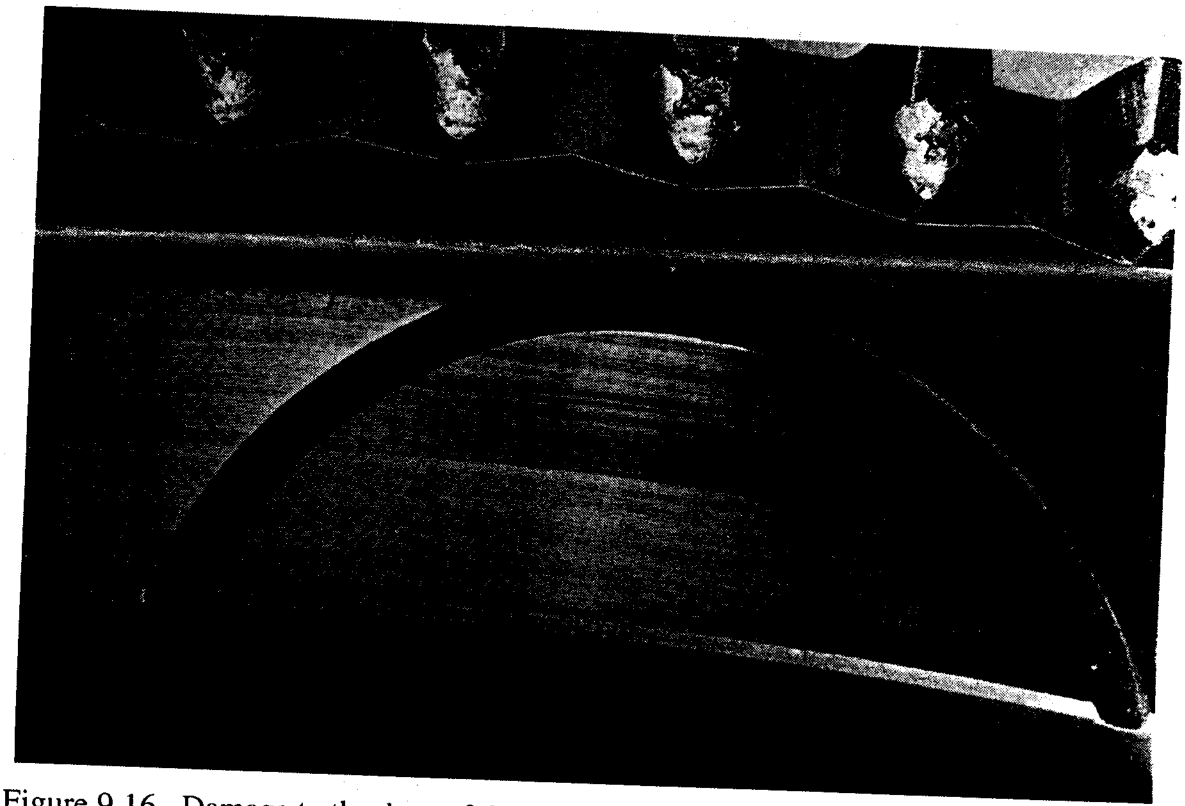


Figure 9.16. Damage to the dogs of the selector teeth caused by grating. Ring groove in the friction taper of the commercial vehicle synchroniser hub

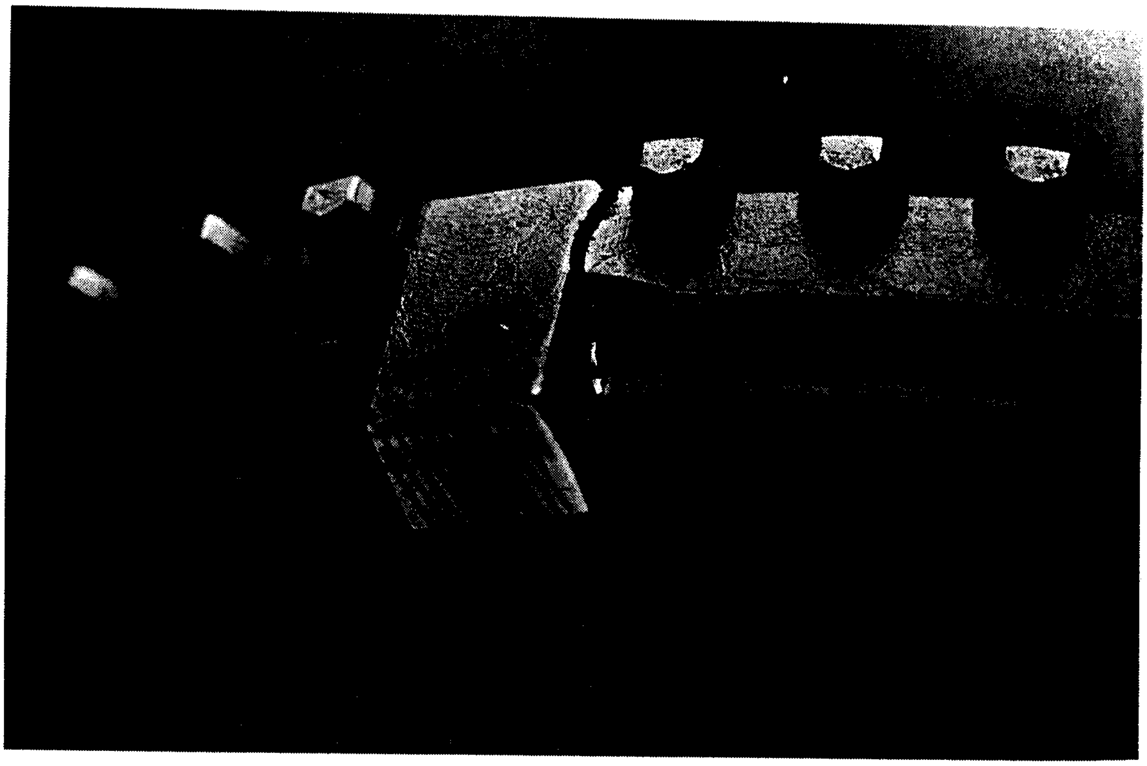


Figure 9.17. Damage to the synchroniser ring caused by torsional vibration, fracture at the stop boss

This picture shows clearly the synchroniser ring locking toothing. In a synchroniser with dimensions sufficient for the mechanical stress encountered, the thermal stress determines the performance limits.

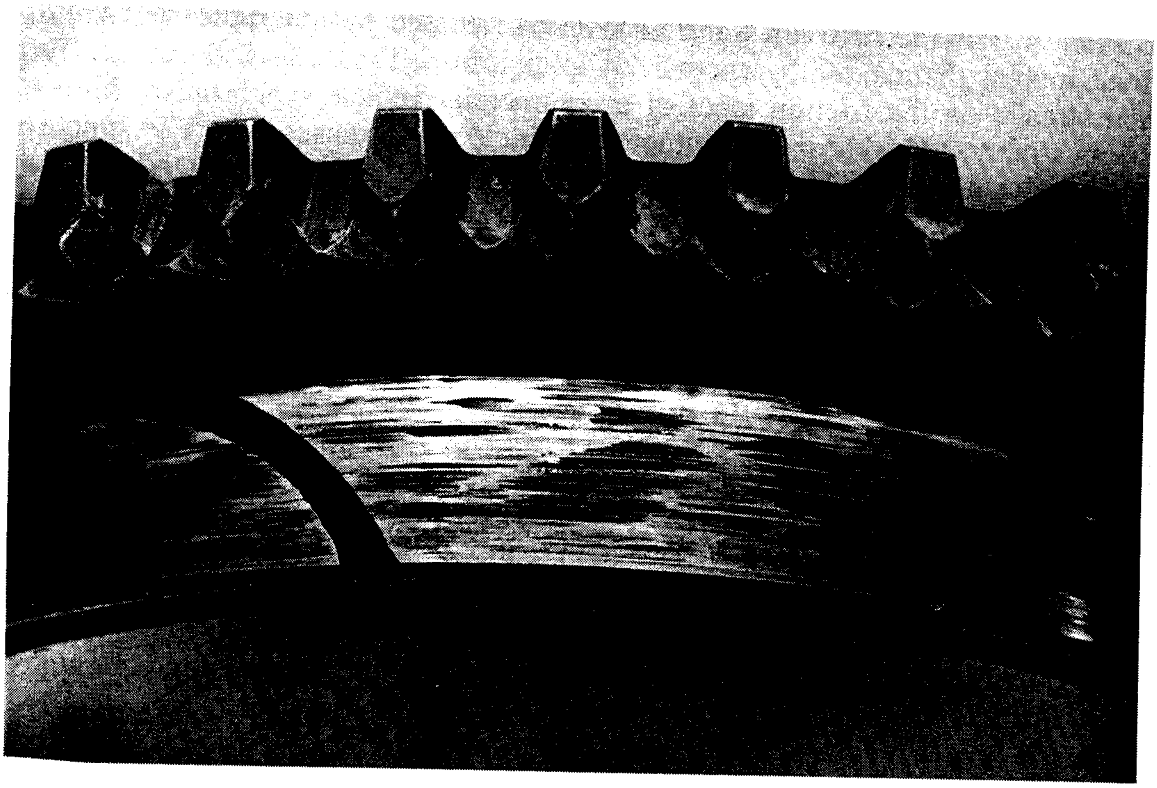


Figure 9.18. Scuffing and thermal streaks on the friction taper of a commercial vehicle synchroniser hub

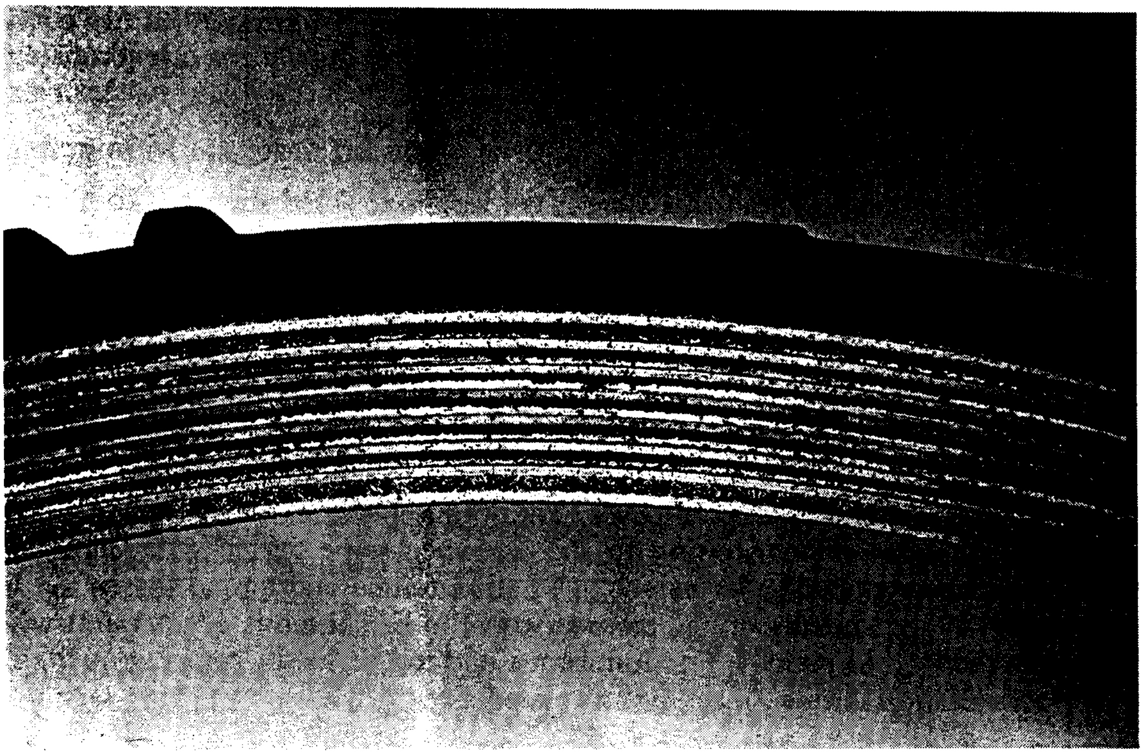


Figure 9.19. Increased wear at the molybdenum friction surface of a commercial vehicle synchroniser ring caused by thermal overload

The surface temperature ϑ can reach peak levels of up to 1000 °C at particular points in less than 0.1 seconds [9.6]. If the thermal stress exceeds the permissible levels, the friction surfaces are damaged.

A distinction is made between transient overload caused by hot spots, and continuous overload such as that caused by excessive slipping times. Figure 9.18 shows scuffing and thermal streaks on the friction taper of a commercial vehicle synchroniser hub. Figure 9.19 shows increased wear on a commercial vehicle synchroniser ring. The synchroniser ring shown has a steel ring with a molybdenum friction surface and ground-in grooving. Even without overload, thermal stress arising during normal service has a detrimental effect on synchronising action. Acceptable frictional power related to surface, $P_{A,\,perm}$, is the measure normally used to assess thermal stress.

9.4.2 Basis for Design Calculation

As shown in Section 7.4 "Operational Integrity and Service Life", not all components in a transmission are susceptible to service life calculations (see also Section 16.2.2 "Qualitative Reliability Analysis"). The service life of "B" components in the "A, B, C" analysis cannot be calculated. Synchronisers are "B" components, so design engineers have to rely upon empirical data.

In calculations relating to mechanical synchronisers the general fundamental equations for shiftable friction clutches apply, as described in [9.7].

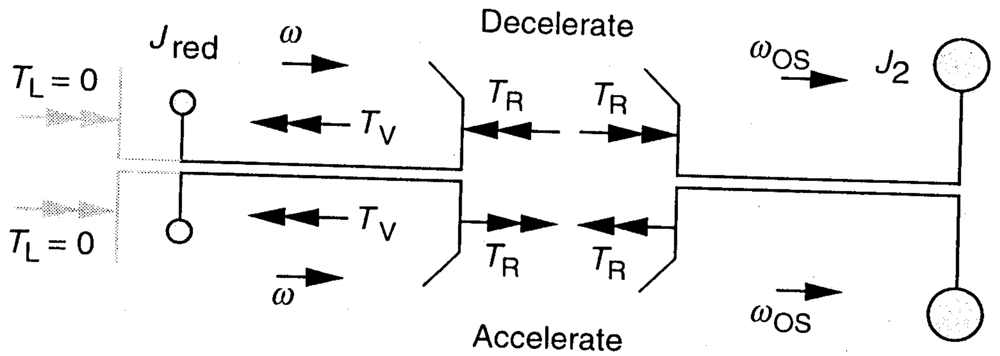


Figure 9.20. Synchronisation of two equivalent rotating masses

The torque equilibrium for a synchroniser as in Figure 9.20 is:

$$T_L + \frac{d\omega}{dt} J_{red} + T_V + T_R = 0 \;. \tag{9.4}$$

When the master clutch is fully opened, the load moment $T_L = 0$ throughout the synchronising process. The torque losses T_V are the result of bearing losses, oil churning losses, oil drag losses and oil compression losses. The torque loss figures are specific to each individual transmission. When shifting up, the gearwheel to be shifted is decelerated with the rotating masses reduced to its axis J_{red}. Friction torque and torque losses act in the same direction. When shifting down, the gearwheel to be shifted is accelerated with the rotating mass reduced to its axis. Friction torque and torque losses act in opposite directions. The acceleration torque T_B can be calculated as:

$$T_B = \frac{d\omega}{dt} J_{red} \;. \tag{9.5}$$

Equation 9.4 gives the friction torque T_R as

$$T_R = -\frac{d\omega}{dt} J_{red} - T_V \,, \qquad \left(\frac{d\omega}{dt} < 0 \Rightarrow T_R > 0;\; \frac{d\omega}{dt} > 0 \Rightarrow T_R < 0 \right) \;. \tag{9.6}$$

The power P transmitted momentarily to the synchroniser is derived from the product of the friction torque T_R and the relative angular velocity ω_{rel} of the parts to be synchronised

$$P = T_R \ \omega_{rel} \ . \tag{9.7}$$

From this the frictional work W per gearshift with slipping time t_R may be calculated thus

$$W = \int_0^{t_R} P \, dt \ . \tag{9.8}$$

9.4.3 Practical Design for Acceptable Thermal Stress

This section presents a procedure for designing synchronisers "by hand". Simplifications are needed to make this feasible. For the slipping time t_R, if the following assumptions are made:

- the gearshift effort F = constant,
- the friction coefficient μ = constant,
- torque losses T_V = constant,

then

- friction torque T_R = constant,
- change in angular velocity $\frac{d\omega}{dt}$ = constant.

The errors resulting from the simplifying assumptions made are largely offset in the calculation by the acceptable stress values. The acceptable stress values are derived from experience.

Reduction of Moments of Inertia

As a result of the steps between ratios, the masses involved in the synchronising process are subject to different angular accelerations. In order to be able to use only one angular velocity for all the masses involved in the calculation, the masses are related to one axis. This is normally the rotation axis of the idler gear to be shifted. In general:

$$J_{red,\,i} = J_i + \sum_{k=1}^{l} J_k \frac{1}{i_k^2} \ . \tag{9.9}$$

Example: When shifting the transmission shown in Figure 9.21 from second to first gear, the masses are reduced onto the rotation axis of the idler gear 7. In this case:

$$J_{red,\,7} = J_7 + (J_C + J_{IS} + J_1)\left(\frac{z_7}{z_8}\right)^2\left(\frac{z_2}{z_1}\right)^2 + (J_{CS} + J_2 + J_4 + J_6 + J_8 + J_{10} + J_{14})\left(\frac{z_7}{z_8}\right)^2$$
$$+\left[J_3\left(\frac{z_4}{z_3}\right)^2 + J_5\left(\frac{z_6}{z_5}\right)^2 + J_9\left(\frac{z_{10}}{z_9}\right)^2 + J_{11}\left(\frac{z_{10}}{z_{11}}\right)^2 + J_{13}\left(\frac{z_{14}}{z_{13}}\right)^2\right]\left(\frac{z_7}{z_8}\right)^2 \ . \tag{9.10}$$

On the assumption that the output shaft OS and the components connected to it are not subject to any change of angular velocity during synchronisation, their moments of inertia may be ignored.

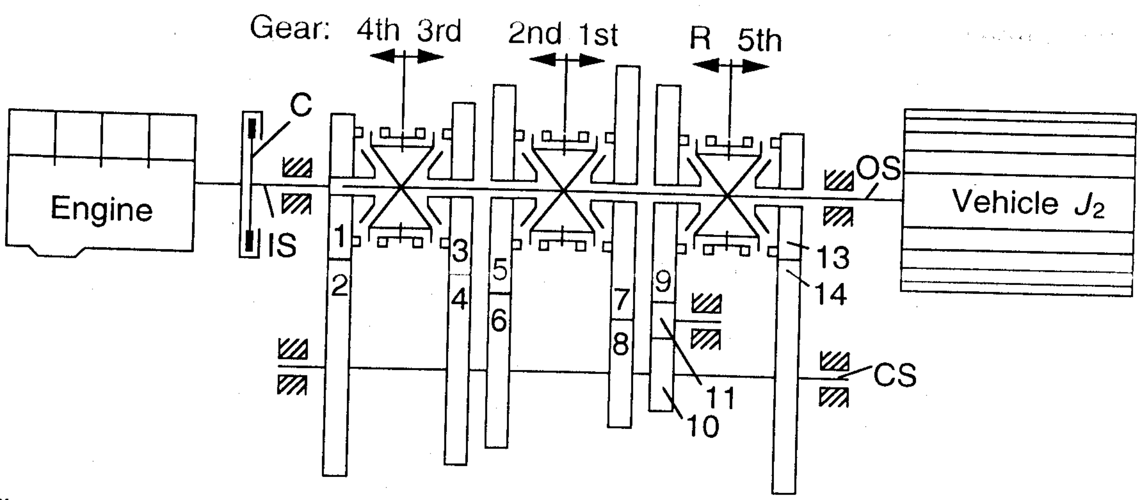


Figure 9.21. Power train with coaxial 5-speed countershaft transmission ("in-line")

Coaxial "In-Line" Countershaft Transmission

Where all the synchronisers in a coaxial countershaft transmission are mounted on the output shaft (main shaft), it is referred to as an "in-line gearbox". Commercial vehicles with a gross weight of more than 4.0 t generally have in-line gearboxes.

This reduces the moments of inertia acting on the input shaft *IS*. This enables all the idler gears involved to be calculated with one and the same moment of inertia. In this case the following reduced moment of inertia applies to the idler gearwheel *i* of the gear *n* to be shifted:

$$J_{\text{red, i}} = J_{\text{red, IS}}\, i_n^2 \,. \tag{9.11}$$

Table 9.5 gives reference values for moments of inertia $J_{\text{red, IS}}$ reduced to the input shaft.

Table 9.5. Moments of inertia $J_{\text{red, IS}}$ reduced to the input shaft (with clutch plate, without output shaft) for "in-line gearboxes"

Type of transmission	**Maximum gearbox input torque**	**Overall gear ratio**	$J_{\text{red, IS}}$
Passenger car 5-speed "in-line gearboxes", upper mid range	250 Nm	5	0.008 kgm^2
Commercial vehicle 6-speed "in-line gearboxes"	900 Nm	10	0.12 kgm^2
Commercial vehicle 9-speed "in-line gearboxes", with rear-mounted planetary range-change unit, 4 x 2 + crawler	1100 Nm	13	0.17 kgm^2

Relative Speed and Friction Speed at the Synchroniser Ring

Speeds before and after synchronisation are determined at the synchroniser of a particular gear. The synchroniser is designed to operate at the maximum relative rotational speed.

The friction speed v at the synchroniser ring has a major impact on thermal stress. The temperature at the friction surface rises exponentially with the friction speed. At the maximum angular velocity difference $\Delta\omega_i$ the friction speed is:

$$v = \Delta\omega_i \frac{d}{2} . \tag{9.12}$$

Torque Losses T_V

The torque losses T_V at the synchroniser ring of a given gear are difficult to determine by calculation. Table 9.6 shows the torque losses $T_{V,\,IS}$ as reference values measured at the input shaft. $T_{V,\,IS}$ is estimated on the basis of empirical data. Together with the permissible stress values, also determined empirically, it allows the design of a synchroniser that is viable in practice.

Table 9.6. Torque losses $T_{V,\,IS}$ at an oil temperature of 80 °C [9.8].

Empirical values	Passenger car gearbox	Com. veh. single-range gearbox	Com. veh. multi-range gearbox
Torque losses at the input shaft $T_{V,\,IS}$	2 Nm	4–8 Nm	10–14 Nm
○ "In-line gearboxes": $T_V = T_{V,\,IS}\ i_n$			
○ For any gearboxes: determine T_V at the idler gear selected, with the values given above from the input shaft (rough estimate)			

Friction Torque T_R at the Synchroniser Ring

Using Equation 9.6, the following applies for the simplifications made

$$T_R = -J_{red,\,i}\frac{\Delta\omega_i}{t_R} - T_V . \tag{9.13}$$

When shifting up (deceleration), $\Delta\omega_i < 0$, and when shifting down (acceleration) $\Delta\omega_i > 0$.

Frictional Work W

From a linear plot of angular velocity ω over the slipping time t_R, integration of Equation 9.7 gives frictional work of:

$$W = \frac{1}{2}\left(-J_{red,\,i}\ \Delta\omega_i^2 - T_V\ \Delta\omega_i\ t_R\right). \tag{9.14}$$

The frictional work must be dissipated as heat, and therefore has a negative sign. The magnitude of the frictional work $|W|$ continues to be used in practical design calculations.

Friction Power P_m

The mean friction power P_m is given by

$$P_m = \frac{W}{t_R} . \tag{9.15}$$

Specific Stresses

When designing synchronisers for permissible thermal stress, the stress values calculated are related to the "gross friction surface area" A_R. Detail features of the friction surface such as grooves and slots are ignored when considering "gross friction surface area". The proportion of the surface that actually comes into frictional contact during the synchronising process cannot be precisely calculated anyway. A_R is made up of the sum of the various gross friction surface areas (e.g. as in the case of multi-taper synchronisers).

$$A_R = A_{R,1} + A_{R,2} + \ldots + A_{R,j} = \sum_{i=1}^{j} A_{R,i} \,. \tag{9.16}$$

The error resulting from ignoring part of the contact area is taken into account in the permissible stress values. The levels of stress expected from the calculation are compared with the stress values permitted for the specific material and application. Table 9.7 gives reference values for the permissible stresses of the following popular friction surface combinations: steel friction taper/uncoated special brass synchroniser ring, and steel friction taper/steel synchroniser ring with a molybdenum friction coating.

Table 9.7. Design data [9.8]. Standard values for the common friction surface combinations steel/special brass and steel/molybdenum

Reference values	**Coefficient of friction**	**Permissible friction speed**	**Specific frictional work**	**Specific frictional power**	**Contact pressure**
	μ	$v = \Delta\omega_i \dfrac{d}{2}$	$W_A = \dfrac{\lvert W \rvert}{A_R}$	$P_A = \dfrac{\lvert P_m \rvert}{A_R}$	$p_{R,i} = \dfrac{\lvert F_n \rvert}{A_{R,i}}$
		v_{perm} (m/s)	$W_{A,perm}$ (J/mm²)	$P_{A,perm}$ (W/mm²)	$p_{R,perm}$ (N/mm²)
Steel/ spec. brass	0.08–0.12	5	0.09	0.45	3
Steel/molybdenum	0.08–0.12	7	0.53	0.84	6

These design data have to be considered in the context of the calculation algorithm and its simplifications. Transient peak loads significantly higher than those given may be tolerated. Peak values for specific frictional work W_A in the synchroniser ring friction linings [9.9] are as follows:

- special brass: 1.2 J/mm²,
- molybdenum: 1.5 J/mm²,
- paper: 2.5 J/mm²,
- scatter sinter: 4.0 J/mm².

The specific friction power P_A is the critical stress in the case of synchronisers subject to high levels of thermal stress. The friction pairing is capable of regenerating itself to a certain degree. A friction lining that has been slightly damaged by violent shifting can regenerate itself by subsequent gentle shifting. Violent shifting can moreover improve frictional efficiency by roughening the smooth lining surface created by gentle shifting. The permissible friction speed v_{perm}, defined by the material pairing, restricts the attainable friction surface diameter d.

But in about 90% of applications it is not the specific stresses that are the limiting factor in using a synchroniser, but the ease of use, which is determined by the variables slipping time t_R and manual effort F_H.

Discussion of the Design Equations

The smaller the slipping time required, the greater the friction torque that must be transmitted – see Equation 9.13. The friction power P to be transmitted increases as $\Delta\omega^2$. Both the slipping time t_R and the difference in angular velocity $\Delta\omega$ are dependent on operational and design factors. There is little scope for influencing the masses involved in synchronisation, expressed by their reduced moment of inertia $J_{red,\,i}$. The friction speed v at the synchroniser ring increases according to Equation 9.12 with the effective diameter d. The main starting points to consider when redesigning existing frictional synchronisers and developing new ones are thus:

Design Measures:

- enlarge the friction surface A_R : – external taper synchroniser,
- increase the number of friction surfaces j and enlarge the friction surface A_R : – multi-taper synchroniser, – multi-plate synchroniser,
- transmission of gearshift effort: – taper angle α, – lever-assisted synchroniser,
- engageability: – clearance characteristic,
- oil supply: – oil ducts, drip edges, deflectors.

Material Measures:

- increase the permissible stress values with "new" friction surface pairings,
- increase the friction coefficient μ.

Most common single-taper synchronisers have reached their performance limits in commercial vehicle transmissions and in the lower gears of passenger car transmissions. In these situations they are replaced by double-taper and triple-taper synchronisers if necessary [9.10, 9.11].

Calculation Procedure

Figure 9.22 shows an algorithm for the thermal design of synchronisers, based on the simplifications, equations and tables presented above.

The procedure is iterative. First the "simplest" solution is calculated, normally a single-taper synchroniser from the standard production range. If the requirements are not met, then the loops of the algorithm are executed repeatedly with varying design, structure and material parameters, until the synchroniser selected meets the requirements. Economic constraints have to be taken into account as well as the technical parameters.

9.4.4 Designing Locking Toothing for Locking Effect

In most common mechanical synchronisers the locking effect depends on the same principle. The friction torque T_R acts to lock the synchroniser, and is opposed by an opening torque T_Z (frequently also referred to as index torque T_I). As long as there is a speed difference, the locking friction torque is greater than the opening torque. This is illustrated below by the example of a locking tooth design (Figure 9.23).

The gearing torque T_Z arising at the dog bevels acts as an opening torque and is calculated from the friction coefficient μ_D between the locking and shifting dogs:

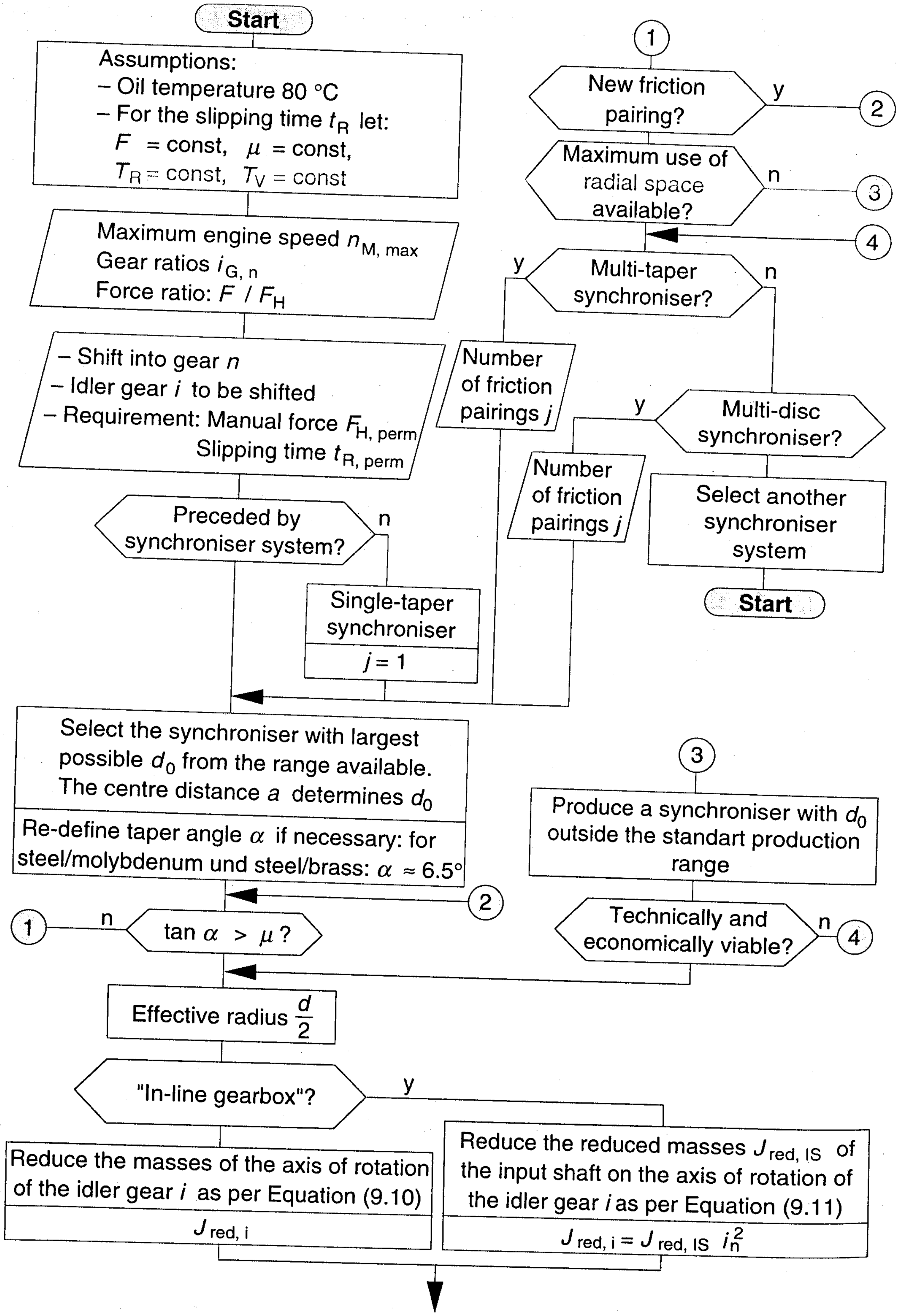


Figure 9.22a. Algorithm for the thermal design of synchronisers

Shift up: $\Delta\omega_i < 0$, Shift down: $\Delta\omega_i > 0$

Determine T_V : measurements, empirical values (Table 9.6)

Friction torque Eq. (9.13): $T_R = -J_{red,i} \dfrac{\Delta\omega_i}{t_{R,perm}} - T_V$

Shifting force Eq. (9.2): $F = \dfrac{2\ T_R \sin\alpha}{j\ \mu\ d}$

Manual force: $F_H = \dfrac{|F|}{\textit{Force ratio}} \dfrac{1}{\eta_{Linkage}}$

$F_H \leq F_{H,perm}$ — n → 1

Calculate the specific stresses, compare to Table 9.7

Determine the gross friction surface A_R Eq. (9.16): $A_R = \sum_{i=1}^{j} A_{R,i}$
(with no deductions for grooving)

Frictional work per shift Eq. (9.14): $W = \dfrac{1}{2}\left(-J_{red,i}\ \Delta\omega_i^2 - T_V\ \Delta\omega_i\ t_R\right)$

Specific frictional work Table 9.7: $W_A = \dfrac{|W|}{A_R}$

$W_A \leq W_{A,perm}$ — n → 1

Average frictional power Eq. (9.15): $P_m = \dfrac{W}{t_R}$

Specific friction surface stress Table 9.7: $P_A = \dfrac{|P_m|}{A_R}$

$P_A \leq P_{A,perm}$ — n → 1

Frictional speed Eq. (9.12): $v = \Delta\omega_i \dfrac{d}{2}$

$v \leq v_{perm}$ — n → 1

Specific friction surface stress Table 9.7: $p_{R,i} = \dfrac{F}{A_{R,i} \sin\alpha}$

$p_{R,i} \leq p_{R,perm}$ — n → 1

The synchroniser selected satisfies the requirements

End

Figure 9.22b. Algorithm for the thermal design of synchronisers

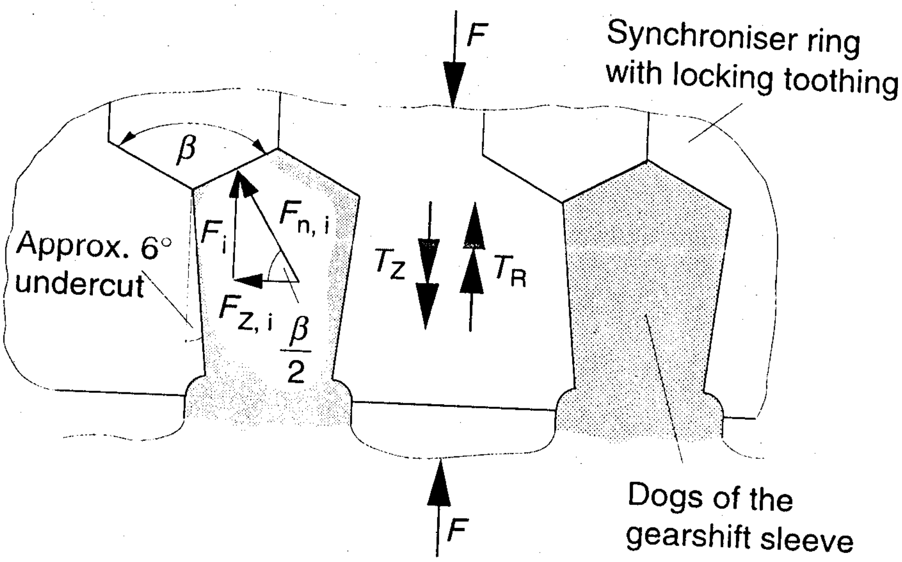


Figure 9.23. Analysis of forces at a locking toothing. Opening torque T_Z

$$T_Z = \frac{F\,d_C}{2}\left(\frac{\cos\frac{\beta}{2} - \mu_D \sin\frac{\beta}{2}}{\sin\frac{\beta}{2} + \mu_D \cos\frac{\beta}{2}}\right) = \frac{F\,d_C}{2}\left(\frac{1 - \mu_D \tan\frac{\beta}{2}}{\mu_D + \tan\frac{\beta}{2}}\right). \tag{9.17}$$

For the friction coefficient μ_D: $\mu_D \approx 0.09$. It is hardly possible to give the actual value of μ_D in operation. The torsional vibration mentioned above causes the dogs of the gearshift sleeve to be "rattled" by the locking toothing.

The opening torque may thus be described in a simplified manner, ignoring the coefficient of friction μ_D, as

$$T_Z = \frac{F\,d_C \cot\frac{\beta}{2}}{2}. \tag{9.18}$$

In designing the locking toothing it is assumed that the entire gearshift effort acts on the locking teeth, so that no excess force is conducted to the synchroniser ring via the thrust pieces. The gearshift sleeve is prevented from engaging for as long as the locking condition

$$T_Z < T_R \qquad \text{with} \qquad T_R = j\,F\,\frac{d}{2}\,\frac{\mu}{\sin\alpha} \tag{9.19}$$

is satisfied. Substituting from Equations 9.18 and 9.19 it follows that

$$\frac{F \cot\frac{\beta}{2}\,d_C}{2} < \frac{j\,F\,d\,\mu}{2\sin\alpha}. \tag{9.20}$$

This results in the following design equation for the bevel angle or opening angle β of the locking toothing

$$\cot\frac{\beta}{2} < \frac{j\,\mu}{\sin\alpha}\,\frac{d}{d_C}\,\frac{1}{S} \qquad \text{with} \qquad 105° < \beta < 125°. \tag{9.21}$$

The locking safety factor S is introduced in order to assess the locking effect. If the opening angle β falls below the lower limit indicated, then "grating" occurs; if the opening angle exceeds the upper limits, the gearshift effort increases and ease of use suffers. In Equation 9.21 the diameter ratio d/d_C appears as an additional influencing variable affecting synchroniser characteristics.

9.5 The Tribological System

The synchroniser ring with its friction layer, the friction taper of the synchroniser hub and the lubricant together constitute a tribological system. Both the engineering design characteristics and the tribological characteristics of the synchroniser affect the ease of use and the service life of a transmission. The locking toothing geometry and the taper angle must be matched to the friction coefficient of the material pairing used.

To achieve high levels of friction torque T_R combined with the least possible gearshift effort F, the dynamic friction coefficient μ must be as large as possible, in accordance with Equation 9.2. Such a high friction coefficient can only be achieved by boundary friction (see Figure 11.2 "STRIBECK curve"). Boundary friction or boundary layer friction is defined as a frictional state in which the normal force F_n is no longer transmitted by hydrodynamic pressure, not even partially (mixed friction) [9.9]. If only part of the normal force is transmitted by hydrodynamic pressure, this is defined as mixed friction. The friction surfaces are only separated by a boundary layer a few nanometres thick, made of chemically formed reaction layers (see also Section 11.2 "Gearbox Lubricants"). The combined action of the lubricant with the structure and the chemical composition of the friction materials influences the boundary layer and thus the friction coefficient [9.12].

In order to counteract the hydrodynamic formation of a lubricant film, the friction surface of the synchroniser ring is provided with grooves, and in the case of commercial vehicles, that of the synchroniser hub as well. Common types of such "drainage" grooves are:

- threaded grooves in the synchroniser ring (unwinding thread – see "Sticking"),
- axial grooves in the synchroniser ring and/or friction taper of the synchroniser hub,
- circular grooves in the friction taper of the synchroniser hub (commercial vehicles).

The function of the grooving in the friction surfaces is to cut through the film of oil and dissipate some of the frictional heat by means of the oil (see also Figures 9.12, 9.16 and 9.19).

9.5.1 Materials

The pairing of materials affects the service life and reliability of a synchroniser. The wear characteristics of the friction pairing must be matched (Table 9.8). The main requirements of a frictional pairing are:

- almost non-wearing with high friction coefficient μ,
- material is easy to machine,
- low material costs,
- almost constant friction coefficient throughout service life,
- resistant to overloading.

Table 9.8. Friction pairings of taper synchronisers

Friction surface of the synchroniser hub: highly wear-resistant	**Friction surface of the synchroniser ring:** scuffing-resistant, wearing	
Case-hardened steel 16 MnCr5 Eh, 20 MoCr4 Eh, with 60 HRC	*Commonly used:*	
	Uncoated special brass rings	Passenger cars, corundum blasted after turning, to texture the friction surface
	Steel rings with molybdenum thick film approx. 0.5 mm thick	Commercial vehicles, grooves ground in
	Trend:	
	Scatter sinter friction linings	Special brass powder with non-metallic constituents
	Molybdenum thin film	Applied to shaped synchroniser ring by flame or plasma spraying
	Paper friction linings	Fibre composites with organic matrix glued to synchroniser ring

Table 9.9 gives a comparative evaluation of friction materials.

Table 9.9. Comparative evaluation of synchroniser ring friction linings (WAGNER [9.9]). Improvement: +++ substantial, ++ noticeable, + little, O none; Deterioration: – little, – – noticeable. Basis of comparison: Synchroniser ring made of special brass

Characteristics	**Special brass**	**Scatter sinter**	**Molybdenum**	**Paper**
Synchroniser ring wear	O	++	+	– –
Synchroniser hub wear	O	O	–	O
Specific frictional work $W_{A,\ perm}$	O	+++	++	++
Coefficient of friction	O	+	+	++
Coefficient of friction constancy	O	+	O	++
Overload capacity	O	++	+	–
Oil compatibility	O	++	+	++

9.6 Engineering Designs

All major synchronisers in use adapt rotational speed by means of slipping friction clutches. In all designs except the Porsche synchroniser, the process of synchronisation and the generation of locking torque are achieved in a manner similar to that in the example discussed in Section 9.3. This section discusses some different types of synchroniser design.

Single-Taper Synchroniser

See Figures 9.7, 9.12 and 9.13 and the associated variations.

Double-Taper Synchroniser

In manual transmissions the design should aim to make gearshift effort equal for all gears. Multi-taper synchronisers are therefore increasingly being used in the low gears (first and second). The number of friction tapers and the friction materials used will depend on the intended use. For example, the use of a three-taper synchroniser may make it possible to use low-cost special brass synchroniser rings [9.11].

Two taper friction surfaces are required to achieve synchronisation in the case of double-taper synchronisers (Figure 9.24).

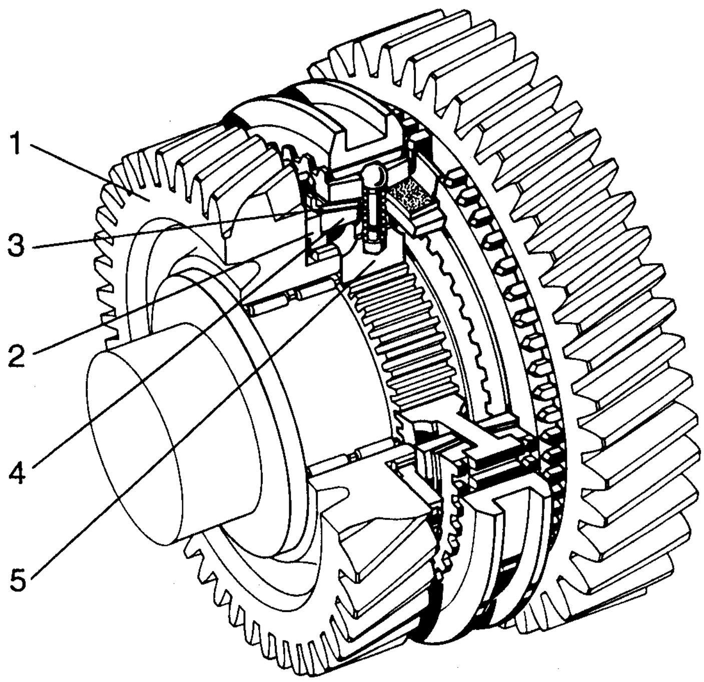


Figure 9.24. Double-taper synchroniser (ZF-D).
1 Idler gear
2 Synchroniser hub with dog gear
3 Double-taper ring
4 Counter-taper ring
5 Synchroniser body

The link between the double-taper ring *3* and the synchroniser hub *2* is rotationally fixed using several dogs, but axially flexible. The counter-taper ring *4* is rotationally fixed to the synchroniser body *5*. The gearshift effort is reduced and the torque capacity, i.e. performance, is increased because of the increased number of friction surfaces and the larger friction surface area of the double-taper synchroniser.

The parallel multi-taper design requires closer manufacturing tolerances and therefore entails higher production costs. The double-taper synchroniser is therefore used only in the lower gears.

External-Taper Synchroniser

In the Mercedes-Benz external-taper synchroniser system (Figure 9.25) the synchroniser unit *3* is fixed by an annular spring *2* to the idler gear *1*. The synchroniser ring has three inward-facing locking lugs *6*, which engage in corresponding grooves *7* in the idler gear. It can turn relative to the wheel both circumferentially and axially once the annular spring has been overcome.

When shifting, the gearshift sleeve is pressed against the synchroniser ring. The friction torque turns the synchroniser ring as far as it will go. Its locking lugs *6* are then so placed before a bevel in the idler gear that the gearshift sleeve *5* and synchroniser ring *3* can move no further as long as the friction torque is not equal to zero.

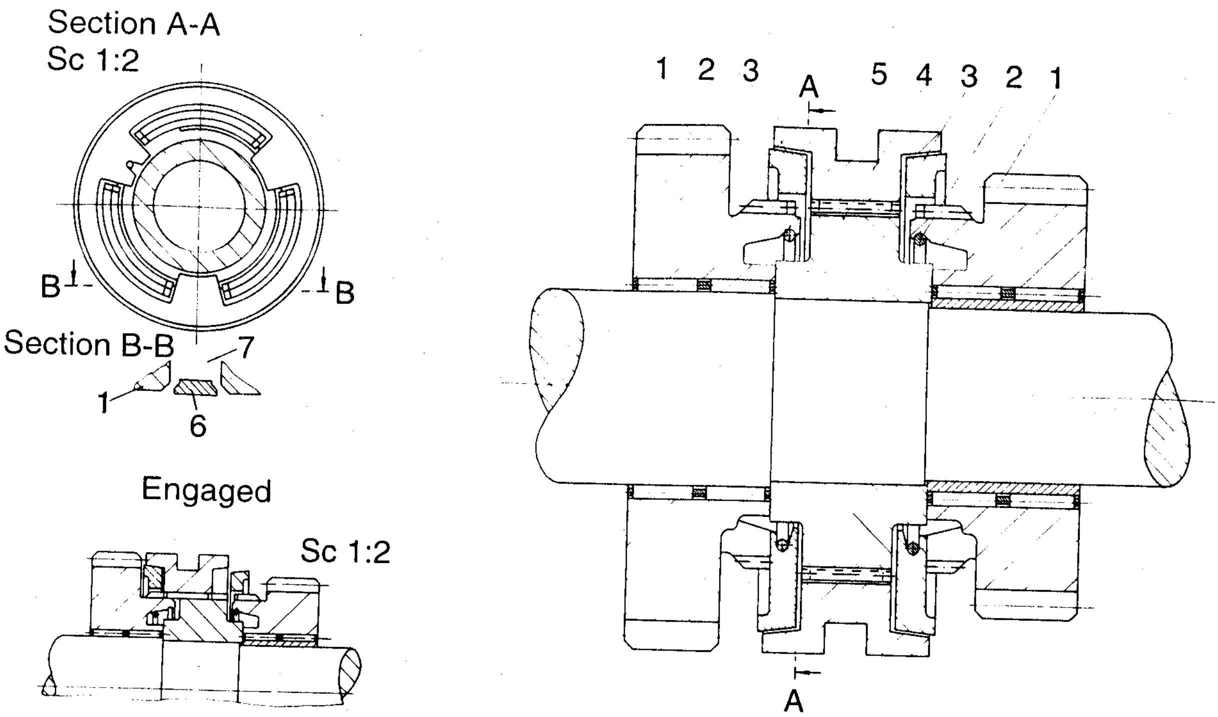


Figure 9.25. Mercedes-Benz external-taper synchroniser system. *1* Idler gear with dog gear; *2* Annular spring; *3* Synchroniser ring; *4* Synchroniser body; *5* Gearshift sleeve; *6* Locking lug; *7* Groove in idler gear

When the speeds are synchronised, the sloping surfaces slide over each other and turn the synchroniser ring back. The noses of the synchroniser ring are pushed into the grooves *7* of the idler gear. Positive engagement can now take place by means of the dogs. When the gearshift sleeve is moved, the annular spring *2* is pushed out of its groove, and slides along the taper surface under the selector teeth. The radial tension of the annular ring exerts an axial restoring force on the synchroniser ring, and moves it into its initial position when the gear is released.

The ratio of the effective diameter d to the clutch diameter d_C is greater than 1. According to Equation 9.21, the opening angle β can be reduced relative to the Borg-Warner system, with the same safety factor S; ease of use is improved. The friction surfaces are located outwards, as compared to synchronisers based on the Borg-Warner system. In accordance with Equation 9.2 this arrangement results in reduced gearshift effort, and in lower specific stresses because of the increased friction surface area A_R. Because of the larger effective diameter d, the friction speed v increases in accordance with Equation 9.12, and the synchronisable speed difference falls.

Locking-Pin Synchroniser

The Spicer or Tompson synchroniser shown in Figure 9.26 is a locking-pin synchroniser. The gearshift sleeve *3* has six drill holes and is rotationally fixed to the transmission shaft, but axially linked by a sliding connection. The locking pins *4* engage in the drill holes parallel to the axis. They are each rigidly connected to a synchroniser ring *2*. The conically countersunk drill holes are larger than the tapered part of the locking pin, enabling the synchroniser pin to turn by a certain amount. As long as there is a speed difference, the opening torque at the conical surfaces of the locking pins and the drill holes is smaller than the locking friction torque. The gearshift sleeve does not slide. When the speeds are synchronised, the circumferential component of the gearshift effort prevailing at the bevelled locating face of the drill holes compresses the compression spring *5*. The gearshift sleeve slides along the locking pins and causes the dogs to engage.

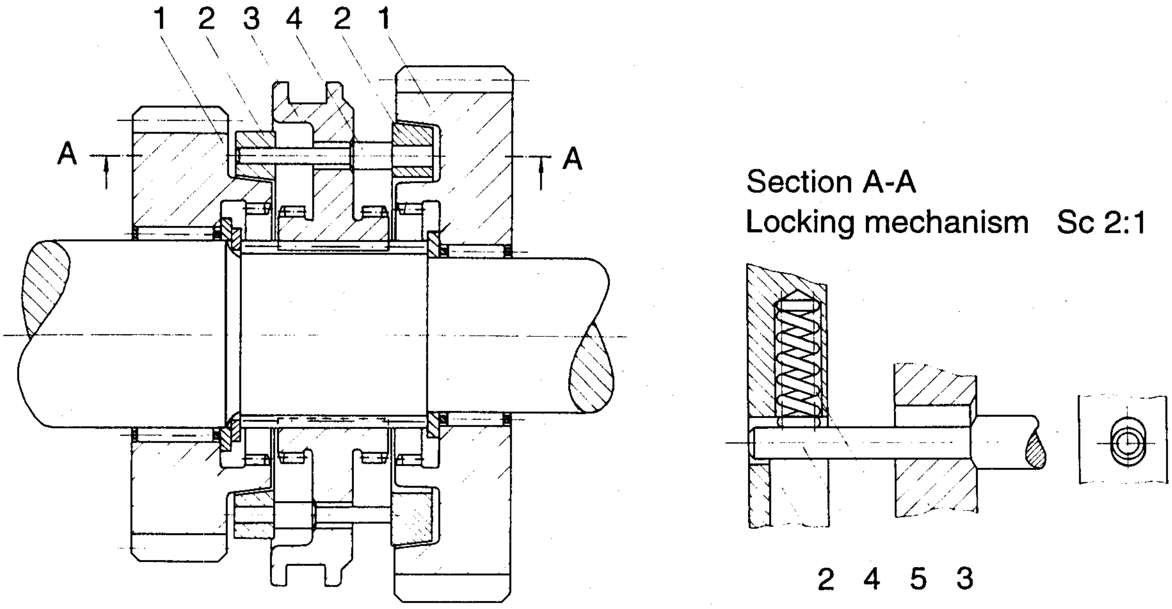


Figure 9.26. Locking-pin synchroniser. *1* Idler gear with dog gear; *2* Synchroniser ring; *3* Gearshift sleeve; *4* Locking pin; *5* Compression spring

Contrary to synchronisers based on the Borg-Warner system, the shifting dogs are mounted on a smaller diameter and the friction surfaces on a larger diameter. In accordance with Equation 9.2, the same friction torque T_R is achieved with less gearshift effort F at the synchroniser ring, because of the larger diameter d. The increased friction surface area A_R results in lower specific stresses. The serial configuration of friction surface and shift movement results in a greater overall package length b_0 than for synchronisers based on the Borg-Warner system.

Multi-Plate Synchronisers

The multi-plate synchroniser in its present form has been developed from the multi-disc clutches used in powershift transmissions (Figure 9.27). Because of its large power transmission surface A_R, it is suitable wherever there is a requirement for very high synchroniser performance.

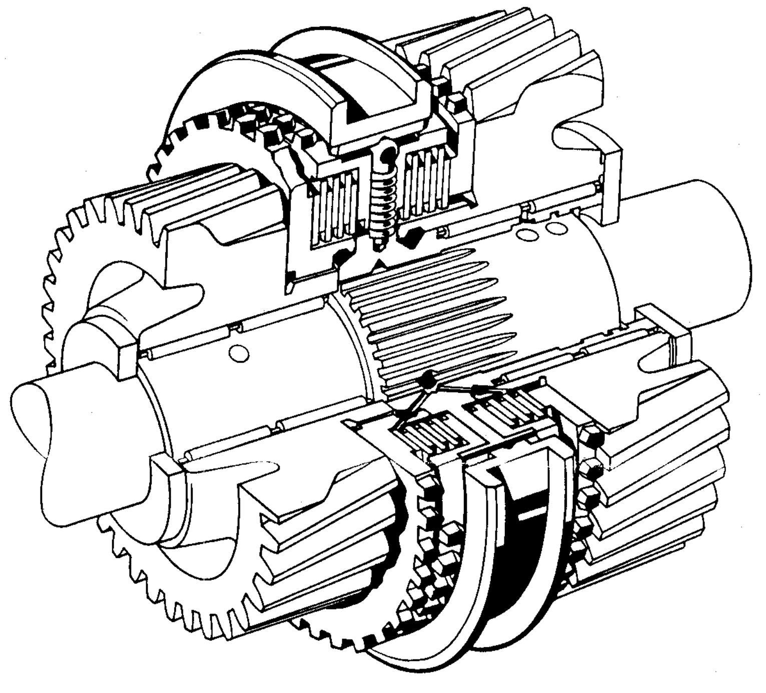


Figure 9.27. Multi-plate synchroniser

The taper angle α of a multi-plate synchroniser is 90°. To be operated with the same gearshift effort as a single-taper synchroniser with α = 6.4°, according to Equation 9.2 a multi-plate synchroniser of the same effective diameter must have j = 9 friction surfaces. The lengths of the two synchronisers are then roughly equal. Multi-plate synchronisers are complex and costly.

Porsche Synchroniser

The Porsche system locking synchronisation (Figure 9.28) has a self-reinforcing locking effect, preventing premature gearshift action before speeds have been synchronised. The Porsche synchroniser requires relatively little gearshift effort, but its high manufacturing cost means it is no longer of practical significance. The quality of the synchronisation process of the Porsche synchroniser is very much subject to variations in the friction coefficient. The synchronisation process will be only briefly described.

The slotted synchroniser ring *4* located in front of the selector teeth is crowned. It has to be squeezed together in order to slide into the gearshift sleeve *6*. When there is a speed difference, the synchroniser ring is twisted by the friction torque until it rests at the stop by means of the pad *8* and the locking belt *3*. This gives rise to radial forces which press the locking belts outwards and prevent the synchroniser ring from being pressed together. The greater the axial clamping force of the gearshift sleeve, the more strongly the slotted synchroniser ring is pressed outwards by the locking belts. After synchronisation there is no more effective spreading force. The synchroniser can then be pressed together, and the gearshift sleeve can slide over it.

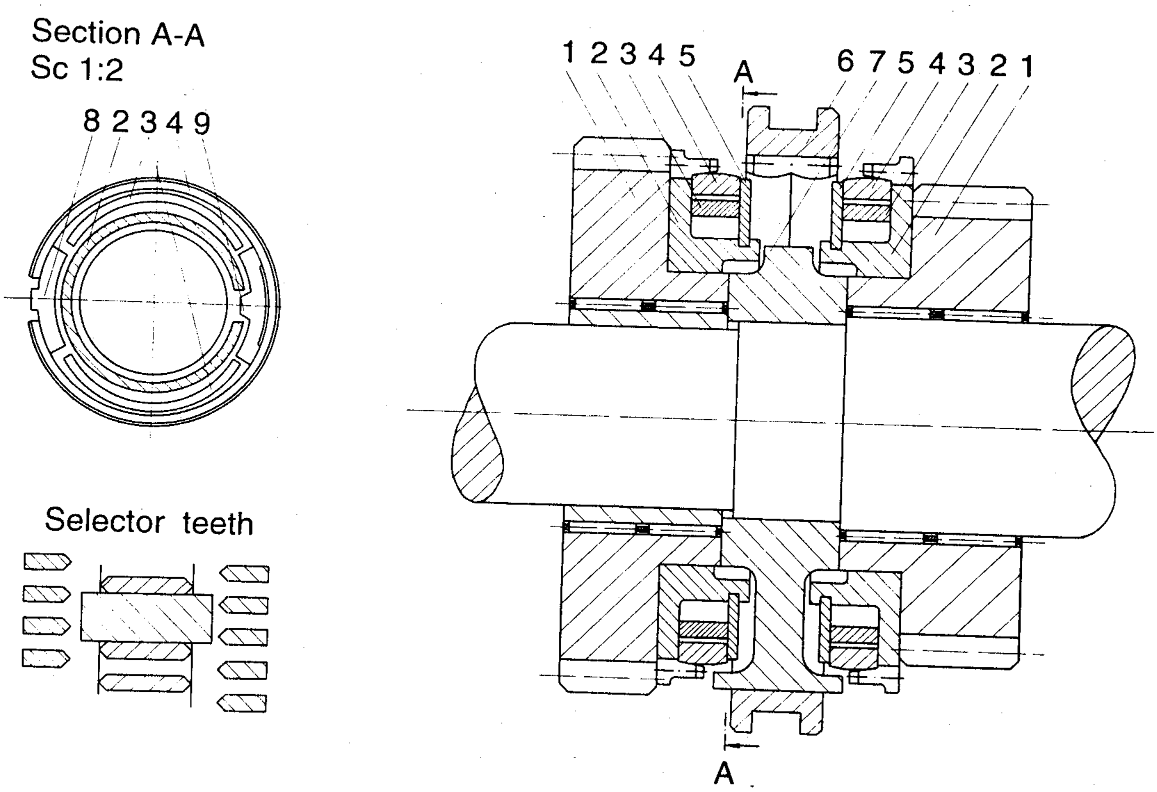


Figure 9.28. Porsche synchroniser. *1* Idler gear; *2* Synchroniser hub with dog gear; *3* Locking belt; *4* Synchroniser ring; *5* Circlip; *6* Gearshift sleeve; *7* Guide sleeve; *8* Pad; *9* End stop

9.6.1 Detail Questions

Synchroniser pack production technology is of crucial importance in determining final costs [9.13].

Ensuring Positive Engagement

The dogs of the gearshift sleeve and the synchroniser hub are undercut by about 6° in order to prevent gear dropout (see Figure 9.23).

Inspection and Testing Technology

When the design calculations for the synchroniser have been completed, tests must be carried out to verify and refine the values. Duty cycles are established in complex, expensive vehicle measurements. They provide practical data on gearshift effort, shifting time, oil temperature and frequency of shifting. These data are fed into the simulation and rig test runs.

By varying individual parameters in the rig test, information can be gathered on the friction coefficient, gearshift effort, friction torque, friction power rating, rotational speed and state of wear.

Abuse Test

Heavy-duty synchroniser tests are used to investigate the performance margin of synchronisers against overloading. This involves shifting gear with high levels of shifting force F and short slipping times t_R under unfavourable conditions (e.g. low gearbox oil temperature). In practice the gearshift profile depends on the individual driver and is therefore random.

It is true of all components that the type and number of operator errors are the major factor determining service life and functioning, and this is particularly true for synchronisers. The General Motors "abuse test" used in the U.S.A. is well known. In this abuse test it is assumed that the driver presses the synchroniser ring against the friction taper of the synchroniser hub without operating or fully opening the master/gearshifting clutch. This also brings to bear the load moment T_L indicated in Equation 9.4. In practice this represents the "high-performance" driver who changes gear without using the clutch. This test is carried out using a high level of gear shifting force ($F > 2000$ N for passenger cars) and with a high speed difference.

9.7 Alternative Transmission Synchronisers

As an alternative to having a synchronising device for each individual gear, a gearbox can also be synchronised in the following ways [9.4]:

1/ central synchronising device for the whole gearbox,

2/ speed matching by the internal combustion engine.

Work is again being carried out on such synchroniser devices in the light of advances in engine/transmission management, with a particular view to their use in semi-automatic and fully automatic passenger car and commercial vehicle gearboxes.

1/ Central Synchroniser

In central synchronisers (Figure 9.29), only one synchroniser unit is needed for all upshift and downshift operations. Energy input and output is external. The electrical transmission control determines the relative speeds of the parts that are to be positively engaged, controls the synchroniser unit that carries out the speed synchronisation, and initiates the gearshift action. A brake retards the masses to be synchronised when shifting up, and a booster motor accelerates the masses when shifting down.

The principle of central synchronisation was developed in 1972, and incorporated in the SST-10 SA Spicer transmission for use in heavy commercial vehicles.

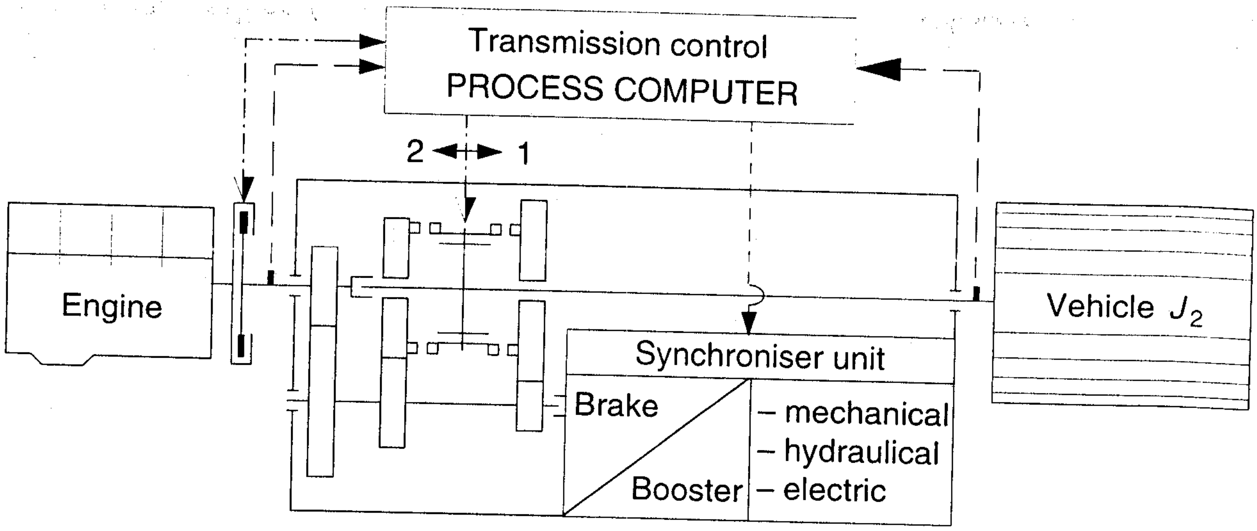


Figure 9.29. Principle of central synchronisation

In 1993 Mannesmann Sachs introduced a design for a parallel hybrid drive (Section 3.2.3, Figure 3.11), in which a dry clutch and also an electric alternator are installed in the clutch housing of the vehicle transmission [9.14]. The small permanently excited synchronous alternator operates together with a motive power battery and the rear-mounted 5-speed transmission to provide limited all-electric drive (7 kW). When the mechanical transmission is shifted, the alternator is used for active speed matching.

When used with an internal combustion engine, this design corresponds in principle to the central synchroniser illustrated in Figure 9.29. When the dry clutch is open, the alternator acts as a brake or a booster. When changing gear during electric drive, speed matching is a function of the drive motor.

2/ Speed Matching with the Internal Combustion Engine

The master clutch is not opened during active speed matching using the internal combustion engine. The speed difference is eliminated by means of brief acceleration or deceleration of the engine. Deceleration has to be assisted by a brake.

A process computer controls the engine, and determines the synchronisation point. Transmission synchronisers of this type have the disadvantage that the synchronising time depends on the engine, and may therefore take too long in certain driving situations. The Faun and Siemens (later Faun and Bosch) Symo gearshift mechanism introduced in 1954, and further refined in the following years, operates on this principle.

Work has recently started on electronically controlled automatic and semi-automatic constant-mesh transmissions for passenger cars. The engine is responsible for speed matching when changing down, and the gearbox brake is responsible when changing up. The ZF-AS TRONiC unit type automatic twin countershaft transmissions going into mass production in 1997/98 with 10, 12 and 16 gears for input torque up to 2600 Nm have such a synchronising device for the main transmission (production design Figure 12.18).

10 Hydrodynamic Clutches and Torque Converters

The torque converter is not just for moving off

Internal combustion engines have a minimum engine speed. To move the vehicle from rest, the speed difference between the lowest engine speed and the stationary transmission input shaft has to be overcome. The torque converter is the standard moving-off mechanism in automatic transmissions. It converts not only rotational speed (as a clutch), but both speed and torque (as a transmission). This chapter is devoted to the torque converter. The hydrodynamic clutch and the hydrodynamic retarder are "reduced" converters, governed by the same theory.

In contrast to hydrostatic transmissions, which operate on the principle of displacement and pressure transmission, hydrodynamic transmissions use the inertia of a fluid flow. The individual components of such a transmission are fluid flow devices forming a closed fluid flow circuit. A rotary pump performs the function of machine, and the turbine that of prime mover. The mechanical energy applied through the drive shaft is converted in the pump into the hydraulic energy of the fluid, and then back into mechanical energy in the turbine, which is available (less losses arising) at the output shaft (Figure 10.1a). Friction losses in the pipelines and outlet losses make the efficiency achievable with such an arrangement very low. The crucial development was the idea of engineer HERMANN FÖTTINGER who largely avoided fluid flow losses by combining the impeller, turbine wheel and a reactor to absorb the reaction torque together in one housing. This also reduced its weight and size (Figure 10.1b).

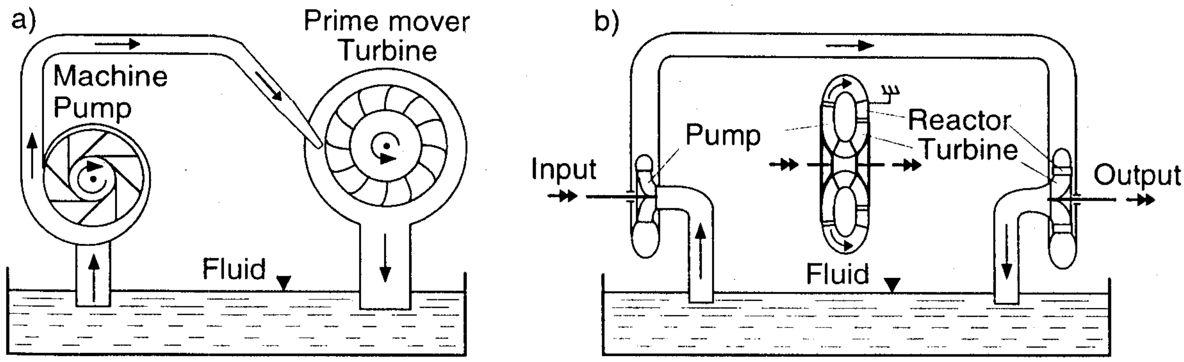


Figure 10.1. **a)** Diagrammatic view of hydrodynamic power transmission. **b)** H. FÖTTINGER's seminal invention of machine and prime mover combined in one unit [10.1, 10.2]

This Föttinger transmission, named after its inventor, is the last real basic innovation in the vehicle transmission sector (see also Section 1.2.5 "Development of Torque Converters and Clutches").

The advantages of the hydrodynamic transmission are as follows:

- *Load-dependent, continuously variable ratio changing:* Adapting the ratio to the load on the output shaft.

- ❍ *Virtually non-wearing:* No abrasion.
- ❍ *Elastic connection between engine and power train:* Vibration and torque shock loads are damped since input and output are not positively engaged.
- ❍ *Reaction effect can be eliminated:* No stalling of the engine.

But they have the following disadvantages:

- ❍ *Low efficiency over broad operating ranges:* Requires a rear-mounted gearbox.
- ❍ *Complexity of the rear-mounted gearbox:* The gearbox must be power shiftable (conventional automatic transmission, CVT) or have an additional gearshifting clutch (torque converter clutch gearbox).

10.1 Principles

A hydrodynamic clutch with the two main components impeller and turbine wheel permits no torque conversion, since no torque can act against the housing (see also Section 4.2 "Speed Converter for Moving Off"). A torque converter thus must have in addition at least one reactor to provide reaction forces (Figure 10.2).

The system uses ATF (**A**utomatic **T**ransmission **F**luid). The fluid flows through the pump, then the turbine, then the reactor, following the particular blade contour, assuming the blades are as close together as required. Figure 10.3 shows the speeds on entering and leaving the blades. The torque converter operation is shown at the *optimum point M* (Figure 10.5); i.e. the fluid suffers no impact losses in this operating mode since it always encounters the blades tangentially.

From Equation 2.5 the following torque equilibrium applies for the torque converter as a whole

$$T_P + T_T + T_R = 0 \ . \tag{10.1}$$

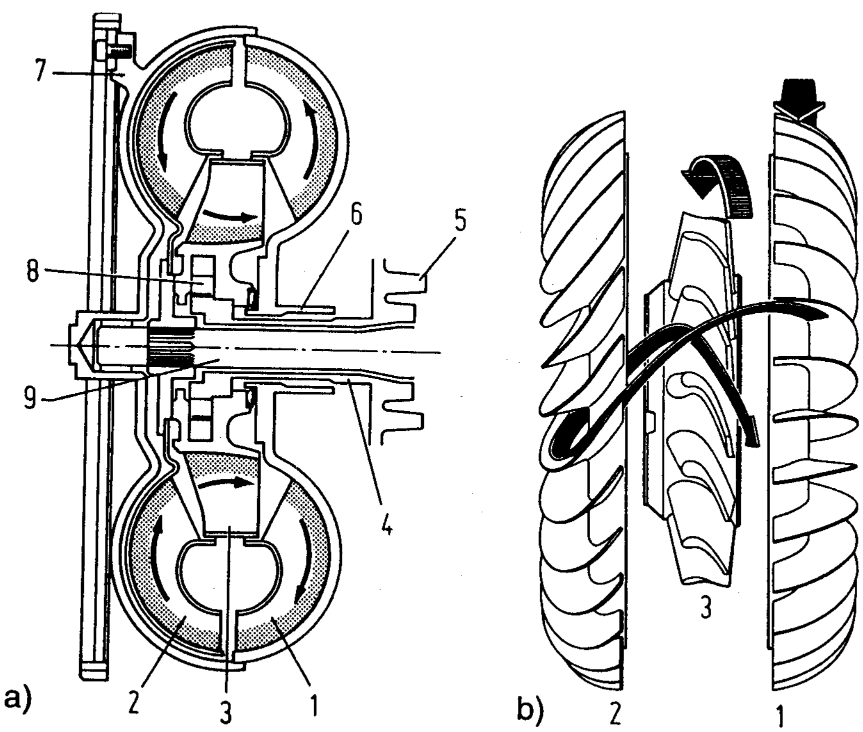


Figure 10.2.
a) Components of a torque converter [10.3].
1 Impeller
2 Turbine
3 Reactor
4 Hollow shaft bracing reactor to housing
5 Housing, fixed
6 Impeller hollow shaft for transmission fluid pump drive
7 Converter cover, linked to the impeller
8 Reactor freewheel
9 Turbine shaft (transmission input)
b) Flow pattern

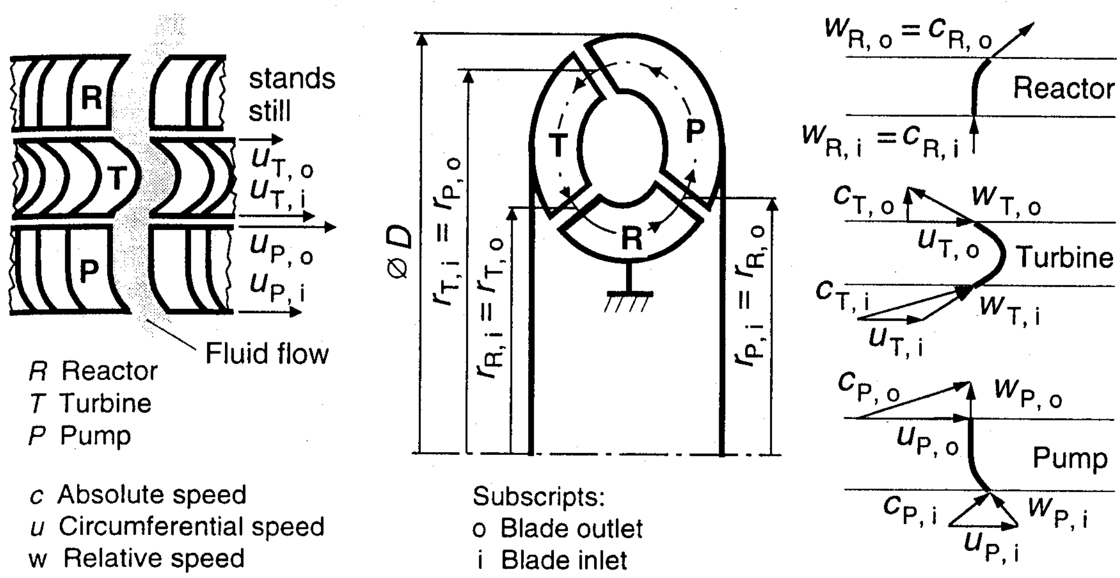


Figure 10.3. Flow cycle in the converter with flow speeds for smooth flow [10.2]

The individual torque values can be determined using EULER's turbine equation

$$T = Q\,\rho\,\Delta(r\,c_u)\,. \tag{10.2}$$

They therefore depend on the flow rate Q, the fluid density ρ and the twist difference $\Delta(r\,c_u)$ between the blade input and output. The twist is the product of the radius r and the circumferential component c_u of the absolute speed c

$$\Delta(r\,c_u) = r_o\,c_{u,o} - r_i\,c_{u,i}\,. \tag{10.3}$$

Since it is a closed system in which the fluid flow passes through all the wheels in sequence, and there is thus the same mass flow everywhere, the twist balance $\Sigma\,\Delta(r\,c_u) = 0$ results in addition to the torque equilibrium. If the power of one wheel $P = T\,\omega$, it follows, given that the reactor remains fixed, that the power equilibrium is

$$\sum P = P_P + P_T + \sum P_V = 0\,. \tag{10.4}$$

The power losses P_V are made up of friction and impact losses, windage and gap leakage. The efficiency η_{TC} of a torque converter, from Equation 4.5a, is

$$\eta_{TC} = \frac{P_T}{P_P} = \frac{T_T\,\omega_T}{T_P\,\omega_P} = \mu\,\nu\,, \tag{10.5}$$

using the torque ratio from Equation 4.3 $\mu = T_T / T_P$ and the speed ratio from Equation 4.2 $\nu = \omega_T / \omega_P$.

Torque converters are designed according to hydraulic model laws using characteristic values derived by experiment. The following two preconditions must be fulfilled for models to be considered:

- *Geometrical similarity*
  The same linear scale m for all parts relevant to hydraulic design, in this case given as the ratio of the profile diameter of the model, D_M, to that of the original, D

$$m = \frac{D}{D_M} \ . \tag{10.6}$$

- *Kinematic similarity*
  Matching speeds of the original and the model must be in the same ratio, i.e. the speed triangles must be similar. Using the designations in Figure 10.3, the following is true

$$\frac{c}{c_M} = \frac{w}{w_M} = \frac{u}{u_M} \ . \tag{10.7}$$

Substituting the circumferential speed $u = \omega D / 2$, the speed scale m_v is given by

$$m_v = \frac{u}{u_M} = \frac{\omega D}{\omega_M D_M} = m \frac{\omega}{\omega_M} \ . \tag{10.8}$$

From (10.8), the scale of the twist m_{twist} is

$$m_{twist} = \frac{\Delta(r\, c_u)}{\Delta(r\, c_u)_M} = m^2 \frac{\omega}{\omega_M} \ . \tag{10.9}$$

The flow rate Q, equal to the product of speed and area according to the continuity equation, is proportional to the product of the speed scale m_v (Equation 10.8) and the area scale m^2 (Equation 10.6)

$$m_Q = \frac{Q}{Q_M} = m_v\, m^2 = \frac{\omega D^3}{\omega_M D_M^3} \ . \tag{10.10}$$

Using Equation 10.2, the power is given by

$$P = T\,\omega = Q\,\rho\,\Delta(r\, c_u)\,\omega \ . \tag{10.11}$$

By substituting Equations 10.8, 10.9 and 10.10 into 10.11, it follows that

$$\frac{P}{P_M} = \frac{\rho\,\omega^3 D^5}{\rho_M\,\omega_M^3 D_M^5} \quad \text{or simply} \quad P \sim \rho\,\omega^3 D^5 \ . \tag{10.12}$$

Since $T = P / \omega$ it follows that for the torque

$$\frac{T}{T_M} = \frac{\rho\,\omega^2 D^5}{\rho_M\,\omega_M^2 D_M^5} \quad \text{and correspondingly} \quad T \sim \rho\,\omega^2 D^5 \ . \tag{10.13}$$

Adding the proportionality factor λ gives the law of similarity

$$T_P = \lambda\,\rho\,\omega_P^2\, D^5 \ , \tag{10.14}$$

where λ is a function of the speed ratio ν and is designated a *performance coefficient*. It can be used to compare various torque converters. The density of automatic transmission fluids $\rho = 800–900$ kg/m^3.

10.2 Hydrodynamic Clutches and their Characteristic Curves

Hydrodynamic clutches contain only a turbine and impeller; the fixed reactor to provide reaction force is not required. Torque cannot be converted, since this configuration does not allow reaction torque to be absorbed. Only speed is converted.

Torque can only be transmitted where there is a speed difference between the impeller and the turbine. The pressure difference arising from differing centrifugal forces circulates the fluid, enabling momentum exchange between the two wheels. The speed difference relative to the speed of the pump is referred to as slip S (see also Equation 4.7)

$$S = \frac{\omega_P - \omega_T}{\omega_P} = 1 - \frac{\omega_T}{\omega_P} = 1 - \nu \ . \tag{10.15}$$

The external air friction T_{fric} can no longer be ignored relative to the transmitted torque when the slip is very small, and thus the transmitted torque tends to zero. This has an impact on the efficiency profile, which drops rapidly towards zero when S is very small, i.e. for ν approaching 1. This range is not reached in normal operation where the residual slip is of the order of S=2–4% as is normal for hydrodynamic clutches (and converters). Substituting Equation 10.15 into Equation 10.5

$$\eta = \frac{T_T\ \omega_T}{T_P\ \omega_P} = \frac{T_T\ \omega_T}{\left(T_T + T_{fric}\right)\omega_P} = \frac{T_T}{\left(T_T + T_{fric}\right)}\left(1 - S\right) \ . \tag{10.16}$$

Figure 10.4 shows the characteristic curves of a hydrodynamic clutch for a constant test pump speed n_{PV}. The non-dimensional representation on the right shows the comparison of different clutches more clearly.

The profile of the performance coefficient λ as a function of the slip $S = 1 - \nu$ can be influenced by the design of the blade geometry and fluid flow path, and by altering the fluid fill level. The aim is usually to limit the torque transmitted with static output, and thereby avoid stalling the engine when idling and moving off.

The hydrodynamic retarder is a special version of the hydrodynamic clutch. The turbine wheel is in this case usually a part of the housing and remains stationary, so the clutch is only operated at the stall point S (see Figure 10.5). The torque transmitted, and thus the braking effect, is very much dependent on rotational speed, and can be controlled by the fluid level and additional adjustments (see Section 11.5 "Vehicle Continuous Service Brakes").

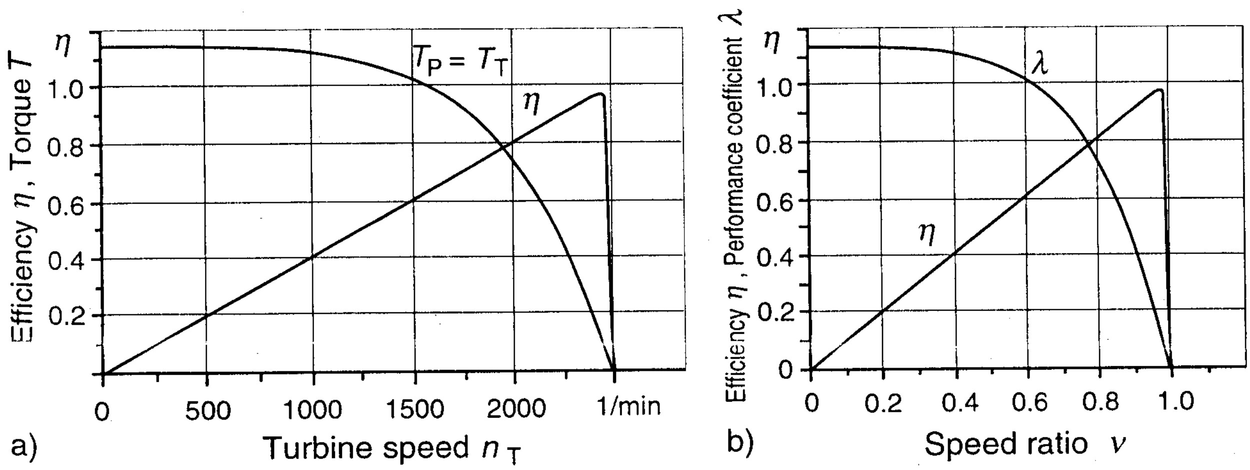


Figure 10.4. Clutch characteristics based on the example of a commercial vehicle clutch. **a)** Dimensional; **b)** Non-dimensional

The converter can absorb a moment of reaction by means of its fixed reactor, and is thus able to convert the input torque. Its efficiency is better than that of clutches at speed ratios below $\nu = 0.7$–0.8 (depending on torque converter type). Plotted against speed ratio, the turbine torque drops, at first approximation, from the stall torque with the ratio μ_{stall} linearly to $T_T = 0$ at speeds in the region of $\nu = 1$. With constant input power this gives rise to a parabolic output power curve $P_T = \omega_T T_T$, and thus a parabolic efficiency curve $\eta = P_T / P_P$ (Figure 10.5). Figure 10.5 shows the following key operating points:

- *S* Stall point, the turbine is at rest, the stall torque ratio is $\mu_{stall} = T_{T,S} / T_{P,S}$,
- *M* Optimum point (design point) with maximum efficiency,
- *C* Lock-up point $T_P = T_T$, $T_R = 0$,
- *F* Free-flow point, no turbine load $T_T = 0$.

At the design point, the point of optimum efficiency, the fluid flows smoothly from one wheel to the next.

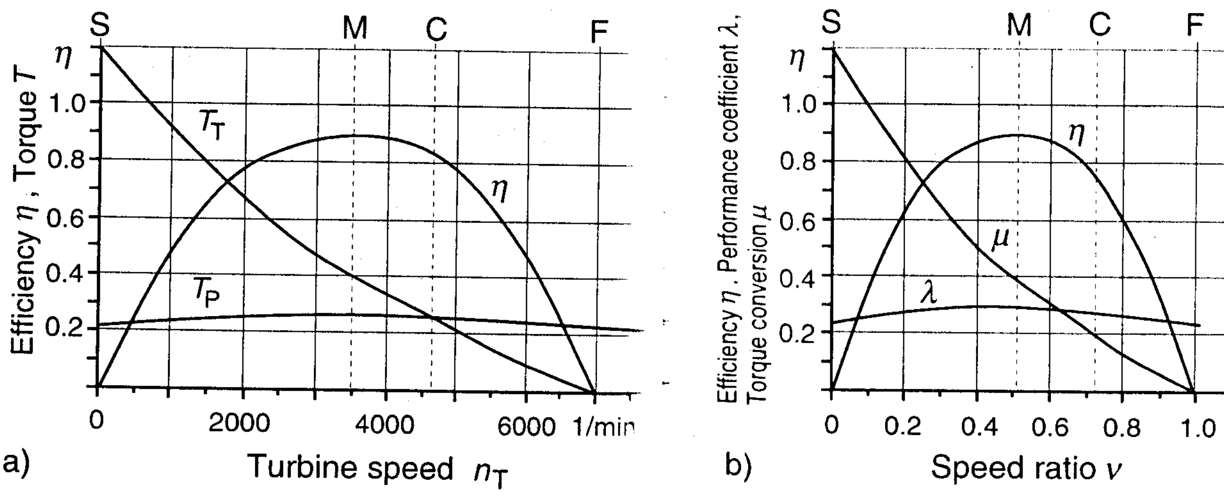


Figure 10.5. Characteristic curves of a torque converter.
a) Dimensional; **b)** Non-dimensional

The torque converter performance coefficient curve $\lambda(\nu)$ can be influenced by the configuration and design of the wheels. In the simplest configuration, with the reactor located before the pump and a single-unit turbine design, λ remains approximately constant so that the engine is evenly loaded regardless of the output speed (Figure 10.6a).

For motor vehicle use it can be more advantageous if λ falls as turbine speed increases. The engine speed is depressed by increased torque at low turbine speeds, so that the engine contributes to speed conversion. This speed reduction also gives the driver more feel for the acceleration process, since the pump and engine speed increase as road speed (turbine speed) increases (see also Figure 10.8). This falling λ characteristic curve can be achieved by locating a turbine immediately before the pump in the direction of flow. In order to still utilise the high delivery pressure and resultant high efficiency, the turbine may be of multi-stage design. This greatly increases the stall torque ratio in particular (Figure 10.6b). If a reactor is again fitted in front of the pump in a multi-stage turbine design, the value of λ can be increased, but the curve will remain largely constant over the entire range (Figure 10.6c) as in Figure 10.6a.

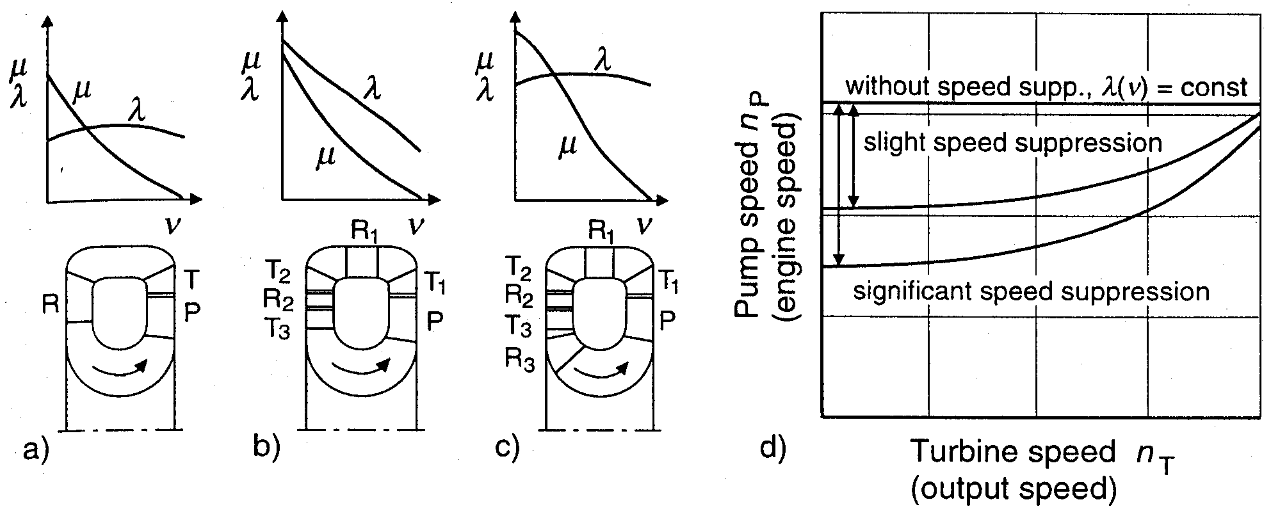


Figure 10.6. Modifying the torque converter characteristics. **a)** Single-stage; **b)** Three-stage with speed suppression; **c)** Three-stage without speed suppression; **d)** Effect of speed suppression [10.2]

10.3.1 The Trilok Converter

The advantages of the hydrodynamic clutch and the torque converter can be combined to avoid the falling section of the torque converter efficiency parabola.

In the *first phase* up to the *lock-up point C*, in which the reaction torque T_R becomes zero, the torque converter operates. In the *second phase* the reactor is released from the housing by means of a freewheel. Since the reactor now revolves freely it no longer absorbs any reaction torque. This results in the straight efficiency line typical of clutches (Figure 10.7).

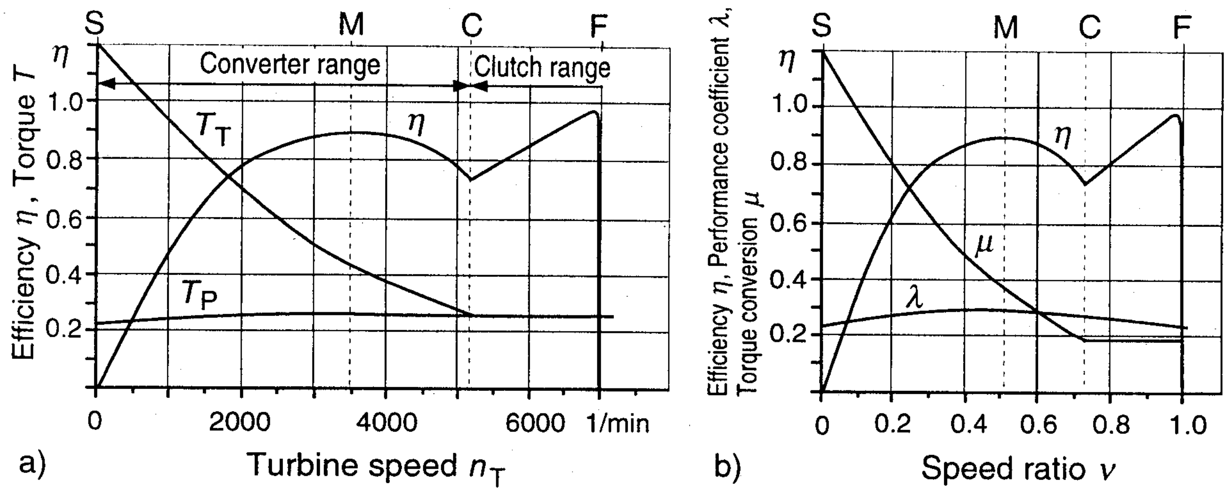


Figure 10.7. Characteristic curves of a Trilok converter.
a) Dimensional; **b)** Non-dimensional

This type of single-stage *two-phase* torque converter is called a Trilok converter, after the TRILOK research consortium that developed it. Its high level of efficiency and simple construction make it particularly suitable for vehicle transmissions, so that Trilok converters with centripetal flow through the turbine are the only type used in passenger cars.

10.4 Engine and Torque Converter Working Together

Since the torque absorption of the impeller in a torque converter without speed suppression is independent of the turbine speed, there is only one parabola in the engine performance map derived from Equation 10.14, with $\lambda = \text{const}$ as the operating curve. Three different single-phase torque converters are shown in Figure 10.8. The diameter of torque converter 1 is designed so that its operating curve intersects the full load characteristic curve of the engine at the point of rated power. The diameter of torque converter 2 was selected to keep maximum engine torque available. The two torque converters are assumed to be geometrically similar.

$$D_1 = \sqrt[5]{\frac{T_n}{\lambda\,\rho\,\omega_n^2}}\,,\quad D_2 = \sqrt[5]{\frac{T_{max}}{\lambda\,\rho\,\omega_{T,\,max}^2}}\,,\quad D_1 < D_2\,. \tag{10.17}$$

The third torque converter is characterised by a falling performance coefficient $\lambda(\nu)$ curve, a speed suppression type. The operating curve is therefore expanded into an operating map in the engine performance map, extending from the left-hand operating curve when $\nu = 0$ to the right-hand line when $\nu = 1$.

There are no points of intersection with the rated power of the engine in the case of torque converters 2 and 3. This is illustrated in Figure 10.8b, where power input is plotted against turbine speed. Torque converter 3 still approaches 95% of the rated power by means of speed suppression as the speed increases, but torque converter 2 can take up only a maximum 85% of the rated power. This has no effect on the torque converter's maximum efficiency. Whilst the point of maximum efficiency of the first torque converter is at a speed ratio of approximately $\nu = 0.75$, in the case of torque converter 2 it moves towards smaller speed ratios. Torque converter 3 lies between the two, with a broad range of high efficiency. At high speed ratios, efficiency can be improved by using a two-phase Trilok converter.

Figure 10.9a again shows torque converter 3 as above with speed suppression, but now in the form of a two-phase torque converter. At the lock-up point ($\nu_C = 0.75$), the reactor becomes free to move by means of a freewheel, and the torque converter acts as a clutch. The expanded operating map applies to this range.

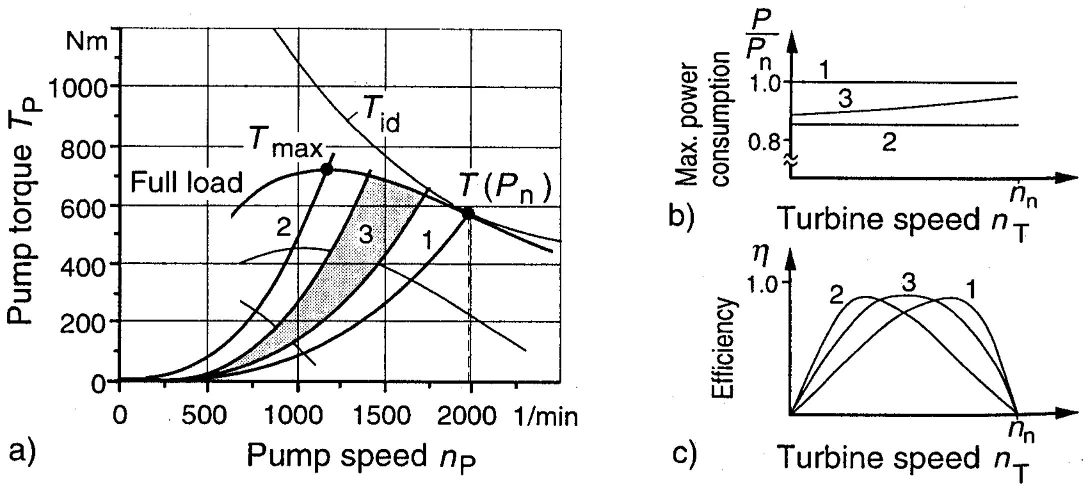


Figure 10.8. Three torque converters with different λ characteristic curve. Converter 1, 2 with constant performance coefficient $\lambda = \text{const} \Rightarrow$ operating curve; Converter 3 with speed suppression $\lambda \neq \text{const} \Rightarrow$ operating map. **a)** Engine performance map; **b)** Maximum power consumption; **c)** Efficiency

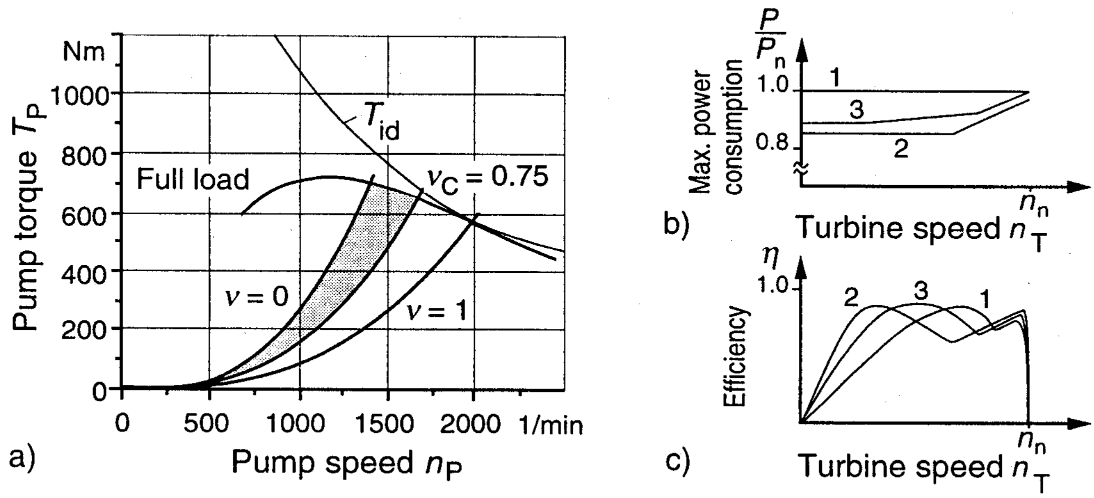


Figure 10.9. A Trilok version of the converter in Figure 10.8. **a)** Engine operating map with torque converter 3; **b)** Maximum power consumption; **c)** Efficiency

The maximum power input for these three Trilok type torque converters is given in Figure 10.9b, and their efficiency curve in Figure 10.9c. The third torque converter shows the advantages of a high starting torque and a broad range of high efficiency at intermediate values of v. Since operation at low speed ratios occurs almost only when moving off, its effect on fuel consumption is of minor significance.

Figure 10.10a shows the three Trilok converters on the turbine map. They approximate closely to the maximum demand power hyperbola through torque conversion. A rear-mounted gearbox nevertheless remains indispensable. Figure 10.10b again shows torque converter 3 in the turbine map, to establish a relation to fuel consumption. The lines of constant standardised specific fuel consumption are also shown.

Figure 10.11a shows three different versions of a torque converter in the engine performance map of a 55 kW passenger car, to illustrate the effect of "torque converter characteristic – hardness" on fuel consumption. A "soft" torque converter was created by reducing the diameter of a standard torque converter by 8.5%, and takes up only 64% of the pump torque of the standard torque converter at the same engine speed (Equation 10.19). The operating map between $v = 0$ and $v = 0.96$ is flatter in the primary map.

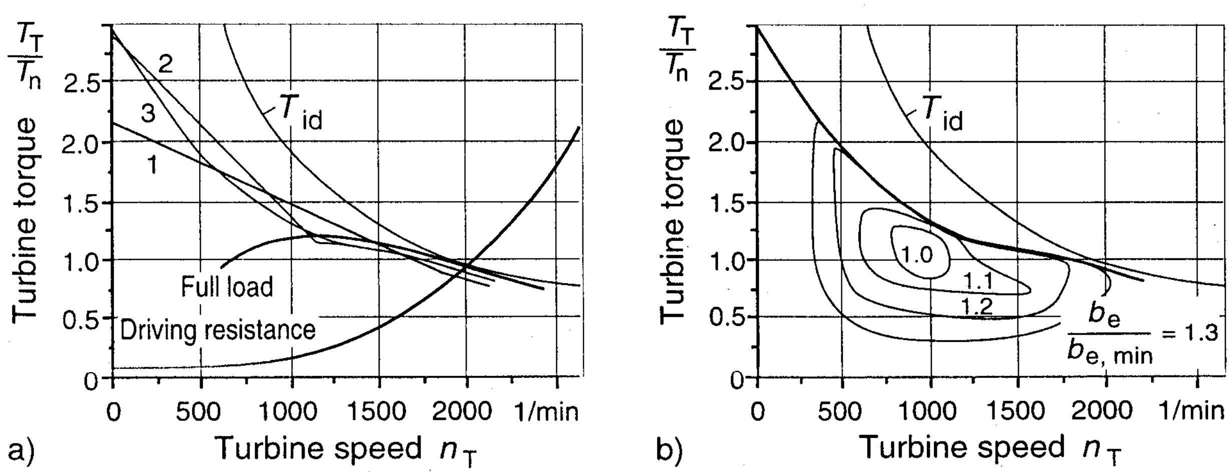


Figure 10.10. The three Trilok converters from Figure 10.9. **a)** In the turbine map; **b)** Trilok converter 3 in the consumption map

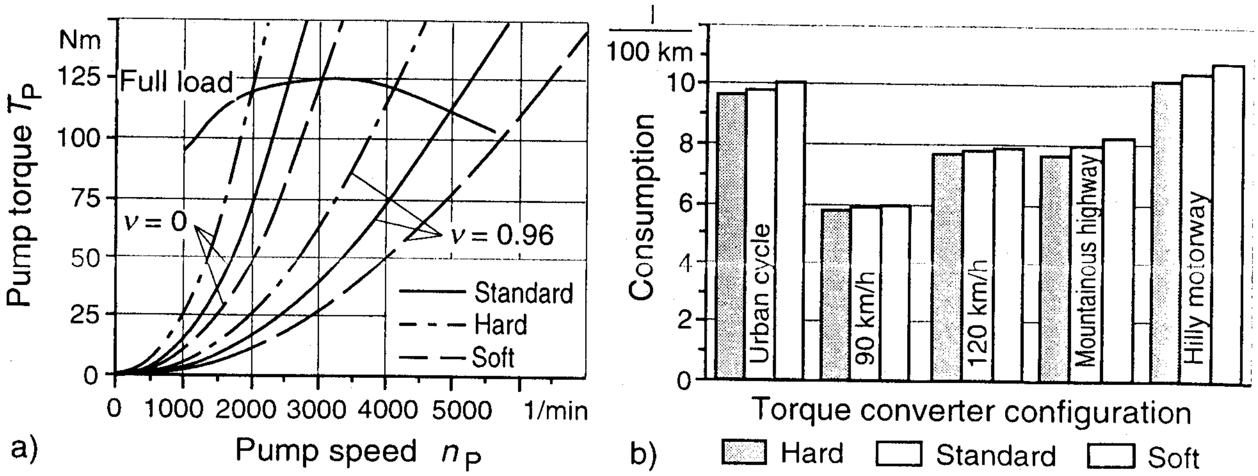


Figure 10.11. Effect of converter diameter. **a)** Primary map; **b)** Fuel consumption

A "hard" torque converter can arise from enlarging the diameter of the output torque converter. Thus for example a torque converter with a diameter 8.5% greater than the standard torque converter will take up 1.5 times the torque of the standard torque converter at the same pump speed: $T_P \sim D^5$ (Equation 10.14). Since this relates to a Trilok torque converter with speed suppression, there are three resultant operating ranges, of which only two pump parabolas each are shown in Figure 10.11a, $T_P(\nu)$ for $\nu = 0$ and $\nu = 0.96$. The engine operating points are thus displaced towards lower engine speeds in hard torque converters, and towards higher engine speeds in soft torque converters.

Figure 10.11b shows consumption under various conditions based on a simulation calculation for the three torque converters in a mid-size passenger car with a conventional automatic transmission and without torque converter lock-up [10.4]. Under full load, the interaction of the engine and soft torque converter results in operating points with greater engine power for the same torque converter drag, provided that the torque converter parabolas intersect the engine full-load curve before the maximum speed. This also makes more motive power available to the vehicle, which is reflected in the acceleration figures. But if limited engine speed means that the torque converter parabolas no longer intersect the engine full-load curve at high ν values, then the relationships are reversed. The harder torque converter displaces the engine operating points at low engine speeds, making it possible to reduce fuel consumption. A torque converter lock-up clutch is normally used with soft torque converters. This is not absolutely necessary with hard torque converters.

10.4.1 Torque Converter Test Diagram, Interaction of Engine and Trilok Converter

The torque converter test diagram, (Figure 10.12) is the basis for calculating the engine operating points and the available traction of a power train with a hydrodynamic clutch or torque converter. See also Section 5.1.4 "Geared Transmission with Trilok Converter" and Figures 5.7 and 5.8. In Section 5.1.4 the example of a passenger car torque converter was used; in this case a commercial vehicle torque converter with a profile diameter of 370 mm is used (Figure 10.12a). The torque converter is to be used in a commercial vehicle with a 150 kW diesel engine. Characteristic values of the torque converter, the torque converter test diagram, were recorded in bench tests at a constant pump test speed of n_{PV} = 1600 1/min and a varying speed ratio ν (Figure 10.12b). For some operating points the resultant values for pump torque T_P and torque conversion μ are listed in Table 10.1.

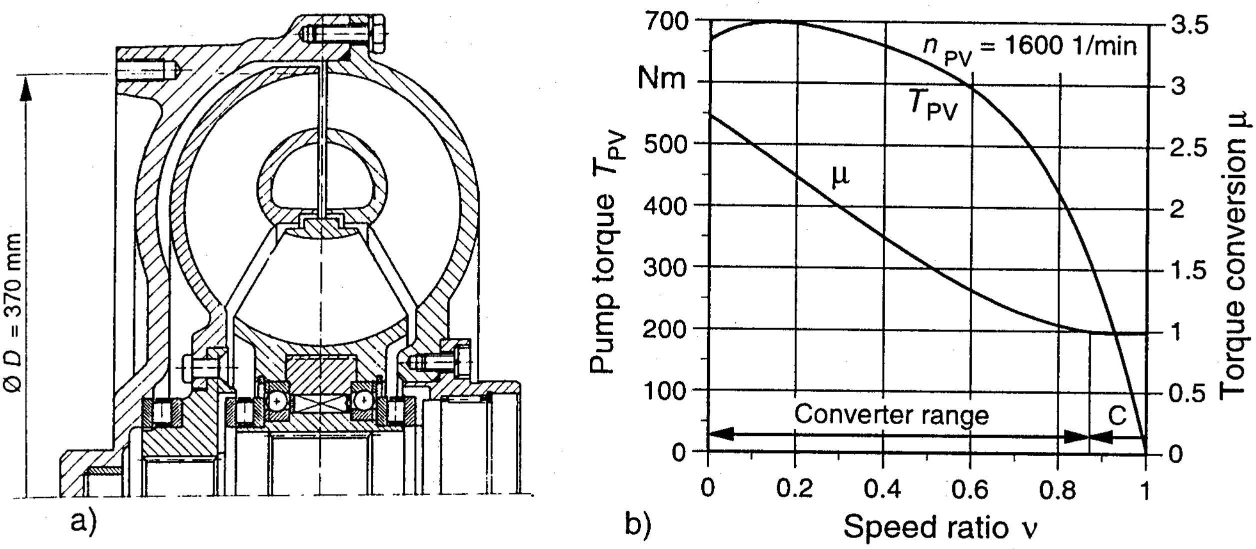


Figure 10.12. **a)** Torque converter with $D = 370$ mm; **b)** Converter test diagram where $n_{PV} = 1600$ 1/min; $\nu_C = 0.88$

To simplify the calculation, in place of the performance coefficient $\lambda(\nu)$, the factor $k(\nu)$ is used, which includes the density of the fluid and the torque converter diameter. Since

$$k(\nu) = \frac{T_{PV}}{n_{PV}^2} \tag{10.18}$$

the pump parabolas (converter parabolas) in the primary map are given by

$$T_P = k(\nu)\, n_P^2 . \tag{10.19}$$

The values of the factor $k(\nu)$ calculated using Equation 10.18 show that the converter in this example is one with speed suppression. The pump parabolas can now be calculated from this information using Equation 10.19, and recorded on the pump map for various speed ratios. This is combined with the full load characteristic curve of the engine to give the primary map (Figure 10.13a).

Table 10.1. Pump side data from the converter test diagram and conversion for full load to the turbine side

	Measurement series No.	1	2	3	4	5	6	7
Test data Pump side	n_{PV} (1/min)	1600	1600	1600	1600	1600	1600	1600
	ν	0.00	0.10	0.40	0.60	0.80	0.88	0.94
	μ	2.73	2.49	1.74	1.35	1.023	0.997	0.997
	T_{PV} (Nm)	672	690	669	585	427	301	185
Calculate converter parabolas	$k(\nu)$ (10^{-6} Nm/min^2)	262.5	269.5	261.3	228.5	166.8	117.6	72.3
Converter parabolas – full load line section	T_P (Nm)	710	715	710	705	680	625	550
	n_P (1/min)	1640	1630	1650	1770	2020	2300	2760
Conversion to turbine side	$T_T = \mu\, T_P$ (Nm)	1938	1780	1235	952	696	623	548
	$n_T = \nu\, n_P$ (1/min)	0	163	660	1062	1616	2024	2594

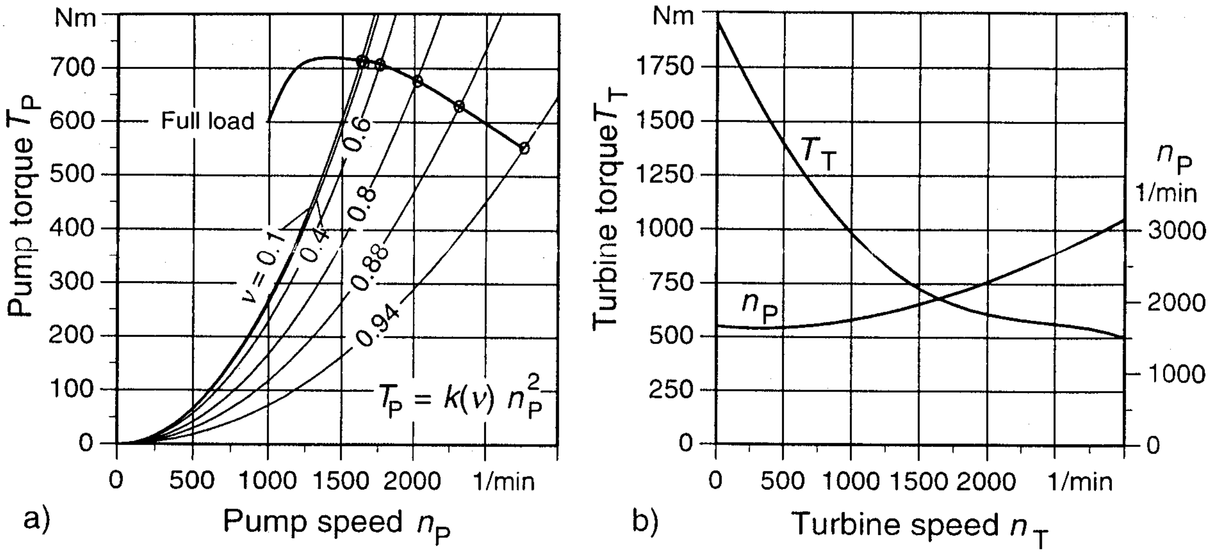


Figure 10.13. **a)** Primary map; **b)** Turbine map (secondary map)

The points at which the engine characteristic intersects the pump parabolas represent possible full load operating points. The pump torques and speeds associated with these points are read off and entered in Table 10.1. In order to derive the secondary map, the turbine speeds and the associated engine torque values also have to be calculated from the torque ratio μ and the converter speed ratio ν using Equations 4.2 and 4.3. Figure 10.13b shows the turbine map derived in this way, see also Section 5.1.4.

10.5 Practical Design of Torque Converters

Figure 10.14 shows a manual calculation algorithm illustrating the rough design of a torque converter based on the preceding example.

10.6 Engineering Designs

With the exception of partially filled clutches, the operating fluid is put under charge pressure by a pump, to prevent aeration and cavitation. The overpressure is held at a minimum value of one to two bar by means of a valve. At higher speeds the pressure created by centrifugal force is a factor, causing the pressure in the passenger car torque converter to rise above 6 bar. This pressure increases with pump speed, and is also dependent on turbine speed and thus load. The pressure is highest when the speeds of the turbine and the impeller are the same, e.g. when the lock-up clutch is engaged. In most cases the impeller is linked to a dish-shaped cover comprising the other torque converter blade wheels, to form a rotating housing. Axial forces on the shafts caused by the internal pressure can be largely compensated in this manner. This structural design also has the advantage of being simple to seal. Residual axial expansion is taken up by a shaft coupling capable of axial movement. The housing can be fitted with cooling plates to improve heat dissipation and to provide bracing. This helps to counter over-expansion caused by internal pressure, which can occur particularly in the case of large commercial vehicle torque converters.

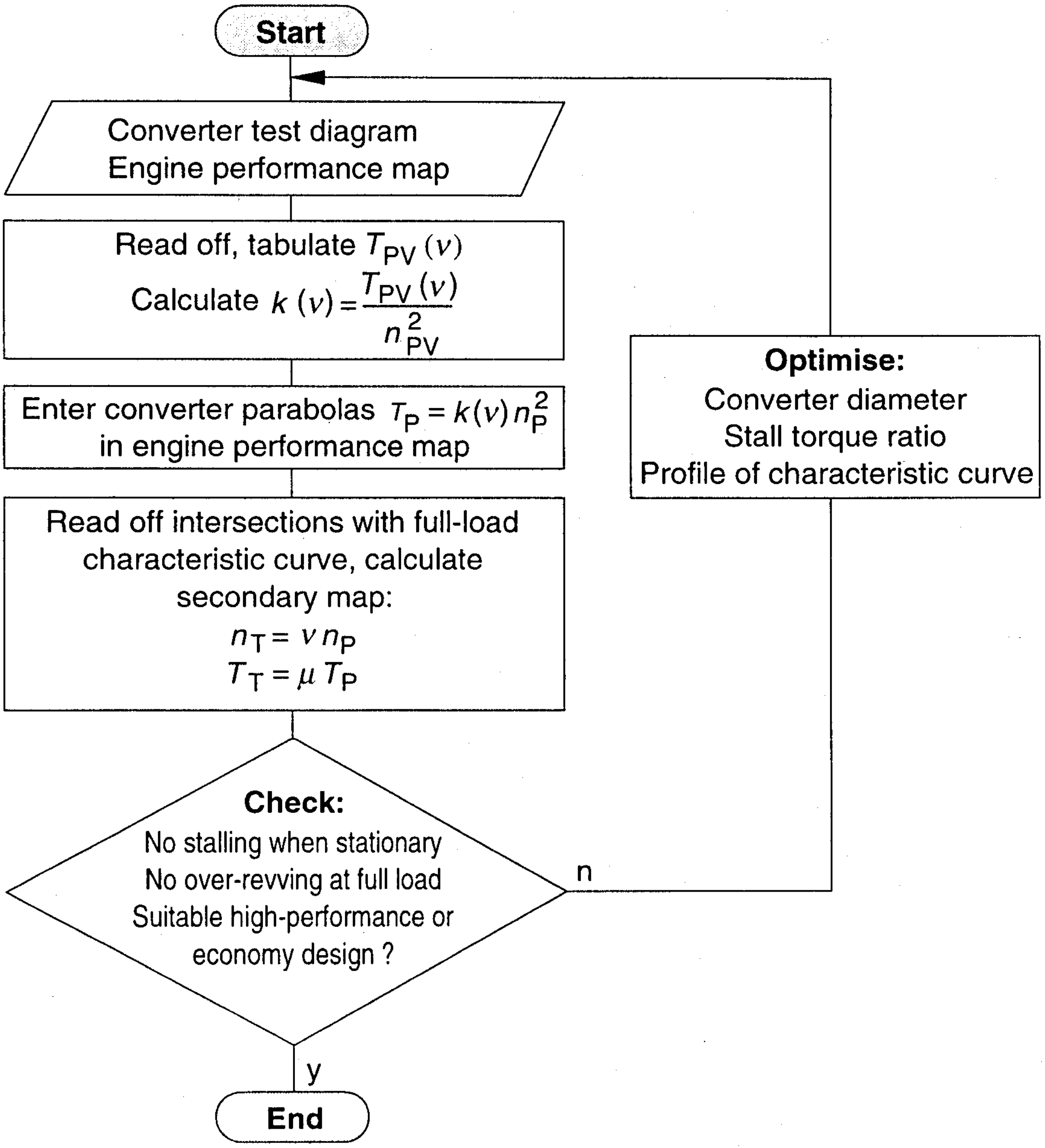


Figure 10.14. Manual calculation algorithm for converter calculation

The large external diameter of the impeller can be used to apply a toothed starter ring. Because of the low torque absorption of clutches and torque converters at low pump speeds, no further gearshifting clutch is needed when starting the engine. The individual wheels often have different numbers of blades, to suppress the symptoms of resonance. Since the engine output shaft and the transmission input shaft are fixed in the vehicle, the torque converter is normally overhung in order to avoid additional bearing friction and to minimise design effort.

The commercial vehicle torque converter shown in Figure 10.15 with a profile diameter of $D = 400$ mm is a Trilok type, which is also equipped with a lock-up clutch and a coasting freewheel. The torque converter is of sheet steel construction. The shells, inner rings and blades are made of deep-drawn sheet steel. Slots and beads in shells and inner rings determine the position of the blades fitted with lobes. When the parts are joined, the lobes are bent over and the joint welded oil-tight with an electron beam. The cover and reactor are made of diecast light alloy. The reactor runs on two ball bearings and is supported by a brake roller freewheel on a hub; it rotates freely from a speed ratio of $\nu_C = 0.8$ at the lock-up point. The hub transmits the reaction torque via a serration gear to

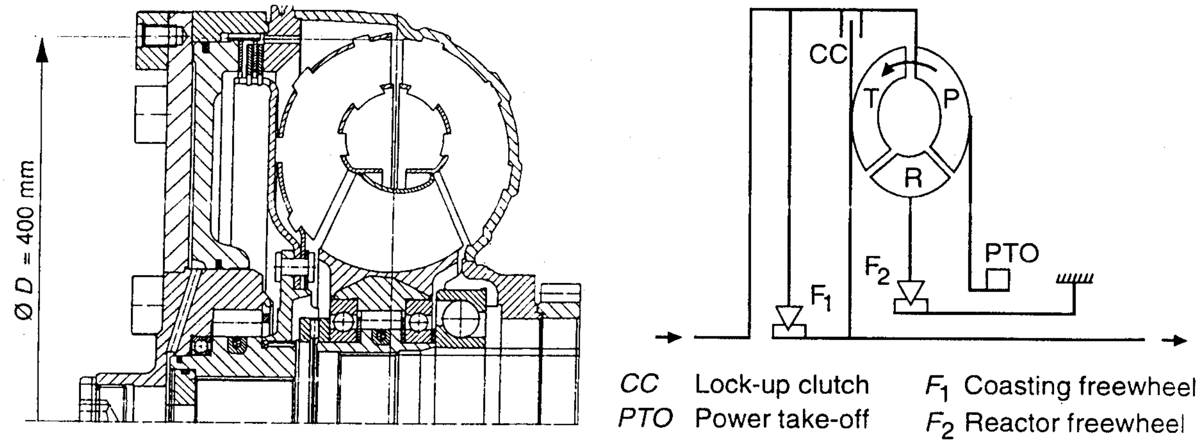


Figure 10.15. Commercial vehicle Trilok torque converter with lock-up clutch and coasting freewheel

a hollow gear that is supported at the gearbox housing. The pump shell is supported on a hub on an angular ball bearing on the output side. There is also external toothing supported by the hub to provide a power take-off for engine-driven auxiliary units (Figure 6.52). This can be used for example to drive an oil pump for the torque converter charge pressure, and for the hydraulics of a rear-mounted gearbox. The lock-up clutch closes beyond a certain speed ratio. It bypasses the torque converter by joining the impeller and the turbine wheel. The torque converter losses are "excluded", and the efficiency rises to nearly 100%. Since torque converters, unlike clutches, mostly transmit less torque when coasting, a coasting freewheel with brake rollers is also fitted. The speed of the turbine can thus never exceed that of the pump, even if the lock-up clutch is not engaged, or its maximum shiftable torque is exceeded. Since it is a torque converter for commercial vehicles, torsion dampers are not used. The loss of ride quality, entailed by direct mechanical connection in the case of passenger cars, is offset by damping elements in one of the wheels.

10.7 Design Principles for Increasing Efficiency

There are two ways of increasing the efficiency of transmissions with torque converters:

- increasing the efficiency of the torque converter, or
- partially or completely bypassing the torque converter and associated losses in certain operating ranges.

10.7.1 Torque Converter Lock-Up Clutch

The simplest way of avoiding torque converter losses is to bypass the torque converter from a particular converter speed ratio by means of an externally operated clutch (Figure 10.15). Since this negates the vibration damping effect of the torque converter, steps have to be taken to compensate for the associated loss of comfort by providing additional torsional vibration dampers, certainly in the case of passenger cars. The process of engaging and disengaging torque converter lock-up clutches is discussed in Section 5.14.

10.7.2 Power Split Transmission

The efficiency of a transmission can be increased without foregoing the advantages of a torque converter, by applying the principle of power distribution.

External Power Split Transmission

Only part of the input power flows through the torque converter of a power split transmission; the other part is transmitted purely mechanically. There are two ways the power can be split in this case. The two power trains can be given a fixed power split by means of a planetary differential gear unit, either on the input side (distributor gear) or on the output side (summarising gear). The power trains are then joined together again, either directly or by a gearbox, so that speed ratio is constant. Figure 10.16 shows a diagrammatic view of both versions. To illustrate the effect of the design on operating performance, the torque converter input power P_{TC} is also plotted between input and output, related to total input power over the speed ratio.

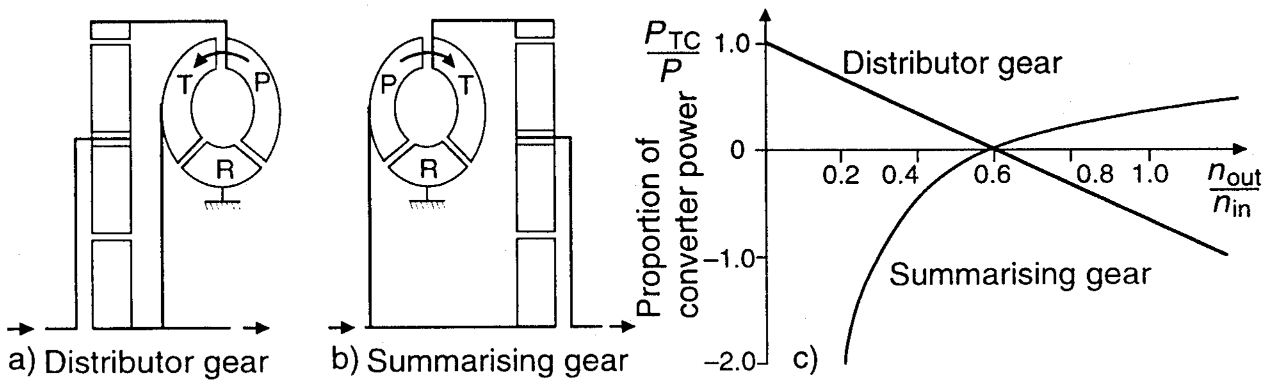


Figure 10.16. Power splitting. **a)** Distributor gear; **b)** Summarising gear; **c)** Power transmitted through the converter as a proportion of the total input power, as a function of the overall gearbox ratio. The speed ratio belonging to the zero crossover of the converter power is a function of the ratio of the summarising gear or distributor gear

Since all the power is transmitted through the torque converter at the stall point in the case of power distribution with a distributor gear, this arrangement is particularly suitable for moving off, and is therefore used mostly for low gears. Where summarising gears are used, high reactive power is generated at the stall point, which can amount to several times the input power. The gear mechanisms therefore have to be correspondingly larger and heavier. Summarising gears are therefore used primarily for high gears that always run above a particular speed ratio. It is true for both types of distribution that efficiency is greatest when the torque converter input power is zero. The operating range thus extends in the case of a distributor gear from the stall point to around the torque converter input power crossover, and in the case of a summarising gear from the crossover to higher speed ratios.

In addition to single power split transmissions there are also multiple power split transmissions (Figure 10.17a), in which the torque converter is located in a coupled gear. It is not used in motor vehicles because of its design complexity.

Internal Power Split Transmission

In an internal power split transmission, the torque converter turbine is of multipart design, or the otherwise stationary reactor also serves as a turbine at certain speed ratios (Figure 10.17b).

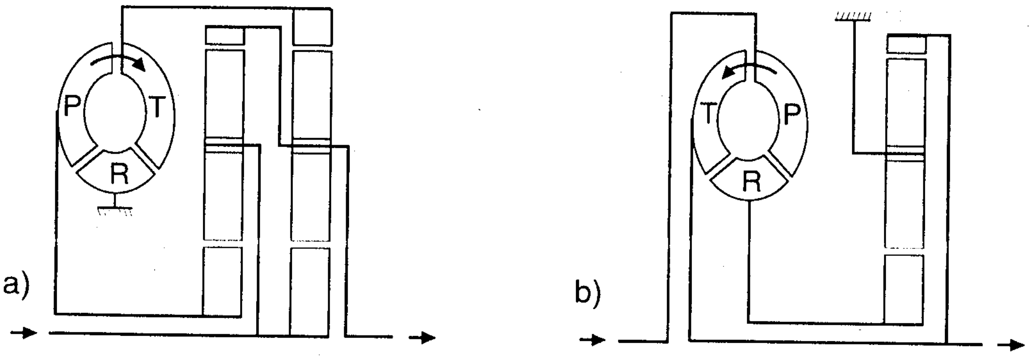


Figure 10.17. **a)** Multiple external power split transmission; **b)** Internal power split transmission

This results in two hydraulic paths that are summarised on the output side by a summarising gear. In this way the torque conversion characteristic curve can be further modified.

Examples of Power Distribution with Distributor Gear

Figure 10.18a shows a diagram of the first two gears of a truck and bus transmission with power distribution using a distributor gear. The impeller of the torque converter is linked to a sun gear of a planetary gear differential; the unit can be stalled by a clutch on the housing. The drive passes along the spider, the mechanical branch flows through the ring gear of the differential. The hydraulic and the mechanical branch are combined at the same speed on the output side of the transmission.

When moving off, the ring gear remains fixed and all the drive power is transmitted through the torque converter. As the road speed increases, so the proportion of power transmitted increases with the torque converter speed ratio. When the torque converter maximum efficiency is exceeded, the transmission shifts to direct drive and the clutch C is engaged. The pump is now at rest, and the turbine is disengaged from the output shaft by a freewheel.

Figure 10.18b shows a second example of power distribution using a distributor gear. The output and planetary gear-set has one degree of freedom less than the input set, and therefore works only as a transmission, and not as a differential. In this example too the proportion of power transmitted hydraulically drops as the output speed increases, raising the efficiency. A counter-rotating torque converter is used because of the second planetary gear-set. Since the pump and the turbine are rotating in opposite directions, this type of torque converter is most efficient at low turbine speeds – i.e. especially when moving off. To shift up, clutch C_1 is engaged, and clutch C_2 is disengaged. This eliminates the hydraulic split, and the turbine is only subject to a low level of torque because of the disengaged clutch C_2.

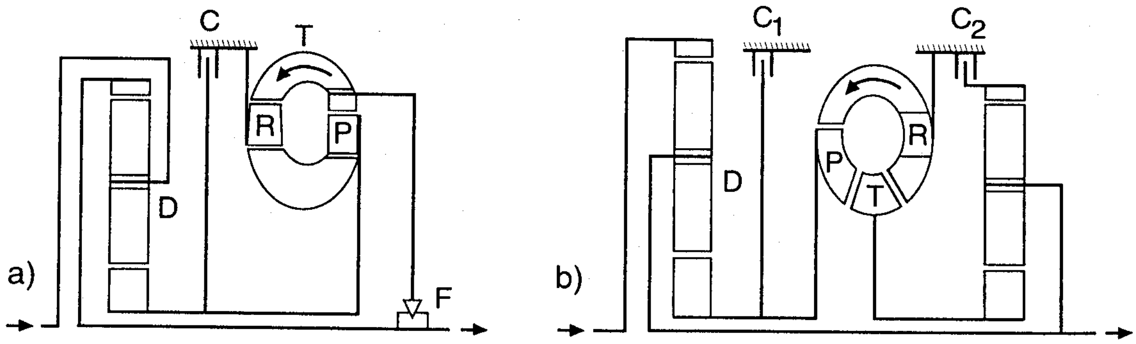


Figure 10.18. Power split with distributor gear. **a)** Truck gearbox, **b)** Bus gearbox. *F* Freewheel; *C* Clutch; *R* Reactor; *P* Pump; *T* Turbine; *D* Distributor gear

Examples of Power Splitting with a Summarising Gear

Figure 10.19 shows a diagram of power splitting with a summarising gear. All the power flows through the torque converter in first and second gear only. In third gear, the power is split between two trains at the same speed, by means of a torsionally damped clutch. The hydraulic and mechanical paths are then summarised again by the Ravigneaux set acting as a summarising gear. The output drive is through the ring gear. The Trilok torque converter used operates largely in the clutch range, and its slip has little effect on the transmission's speed ratio and efficiency because of the power distribution.

The transmission shown in Figure 10.19 is described in detail in Section 6.6.3. See also Figures 6.27 and 6.28.

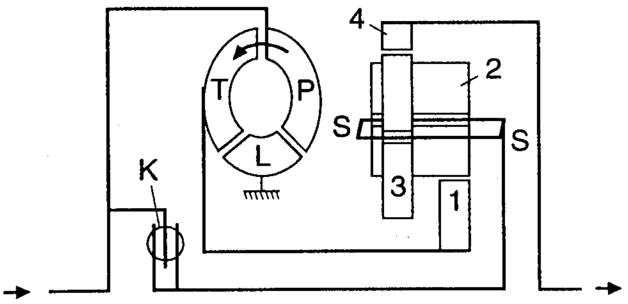


Figure 10.19. Power split with summarising gear. See Figure 6.28: Power flow in 3rd gear of a 4-speed passenger car automatic transmission. *C* Torsionally damped clutch; *P* Impeller; *T* Turbine; *1* Sun gear; *2* 1st planetary gear-set; *3* 2nd planetary gear-set; *4* Ring gear (output); *S* Shared spider

The following example of internal power split relates to a single-phase torque converter with two turbine stages (Figure 10.20).

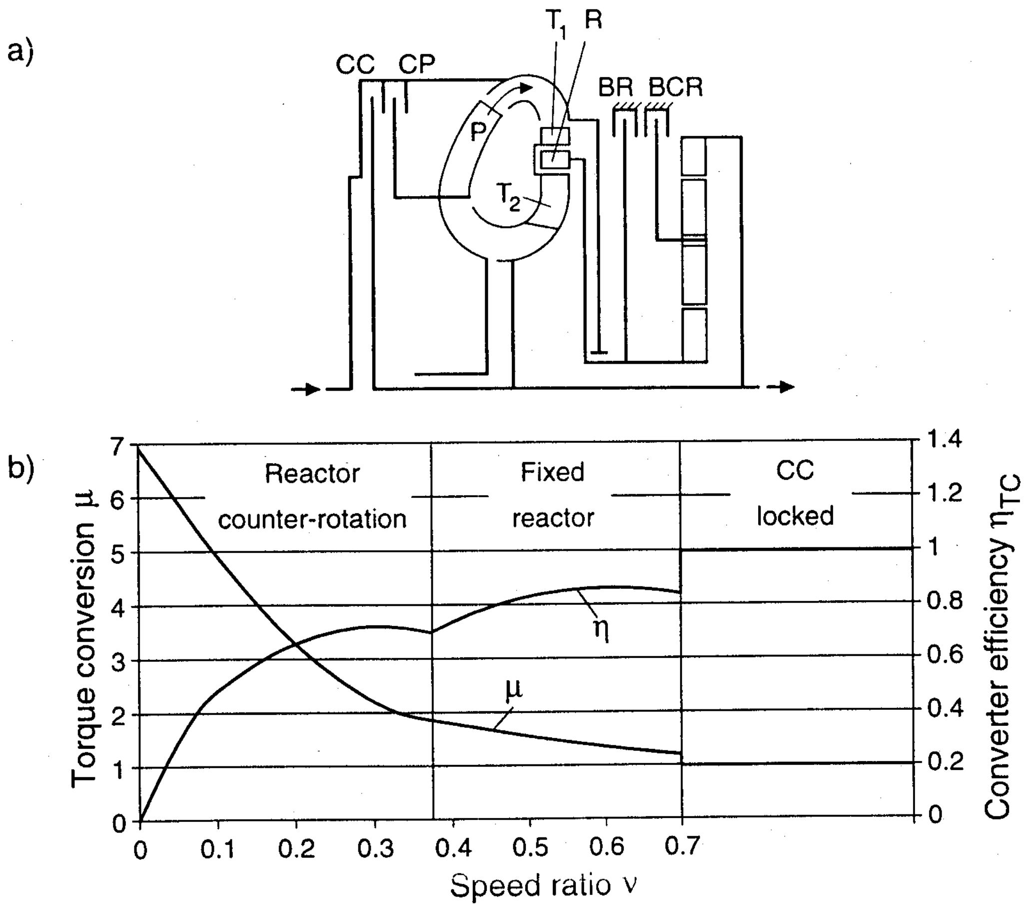


Figure 10.20. Torque converter with internal power split. **a)** Renk/SRM principle (Doromat): *CC* Torque converter clutch; *CP* Impeller clutch; *P* Impeller; T_1 First turbine stage; T_2 Second turbine stage; *R* Reactor; *BR* Reactor brake; *BCR* Counter-rotation brake; **b)** Characteristic curves

The intermediate reactor R can be shifted with the brake BCR to counter-rotation to the impeller and turbine ("double rotation"). This internal power split results in a high torque conversion (Figure 10.20). In the "reactor fixed" range, the brake BR is on. Since the single-phase torque converter has no clutch range, the torque converter is locked-up with the clutch CC when small differential rotational speeds are reached between the impeller and the turbine. The impeller can be disengaged with the clutch CP in neutral when the vehicle is at rest.

11 Notes on the Design and Configuration of Further Vehicle Transmission Design Elements

Let us proceed from what we know /ARISTOTLE/

This chapter deals with the theory, design and configuration of vehicle transmission *bearings, lubrication, housings, seals* and *retarder systems*. The aim is to give guidance for tackling these design elements. Sophisticated modern quantitative techniques, such as the Finite Element Method (FEM) for calculating housings or designing robust bearings, are not examined in fine detail. Further literature references are given where appropriate.

11.1 Bearings

The function of a bearing is to support or guide components that move relative to each other, to absorb the forces arising, and transmit them to the housing. A distinction is made between plain bearings and roller bearings depending on the type of movement involved. The bearings most commonly used in vehicle transmissions are roller bearings.

"Bearings" as machine elements become effective only when they are positioned between a supporting housing and the shaft to be supported (see also Chapter 8 "Specification and Design of Shafts"). A distinction is made between *fixed/floating bearings* and *supporting bearings* depending on the engineering design and layout of the bearings. Supporting bearings can be further subdivided into adjustable and floating bearing arrangements. Both are the same in terms of their construction. Whilst the *adjustable bearing* has minimal backlash or is even pre-loaded, in the case of *floating bearings* some axial play is deliberately left. This amounts to approximately 0.5 to 1 mm, depending on the size of the bearing.

Shafts normally have double bearings. Multiple bearings are also used in the case of long shafts with large deflections. Shafts with multiple bearings are statically undefined and require a high degree of calculation (see also Section 8.4 "Calculating Deformation" of shafts.

Vehicle transmissions have a high power-weight ratio. This makes high demands on the roller bearings. Bearings are bought-in parts, so close co-operation with the bearing manufacturer is necessary when selecting and calculating bearings. Typical requirements for roller bearings for use in vehicle transmissions are [11.1]:

- guaranteeing bearing load capacity, even when skewed as a result of shaft deflection,
- compensation for major heat expansion with light alloy housings,
- resistance of bearings to high operating temperatures, and consequent low oil viscosity,
- high radial and axial rigidity when meshing.

11.1.1 Selecting Bearings

When selecting and arranging bearings, it is necessary to consider their loading, ease of fitting and removal, and type of lubrication or lubricant. There are further requirements relating to maximum rotational speed, operating temperature, bearing clearance and tolerances, depending on the operating conditions.

The preferred bearing types for shafts in vehicle transmissions are *deep groove ball bearings, four-point bearings, angular ball bearings, cylindrical roller bearings* and *tapered roller bearings*. Deep groove ball bearings are suitable for many applications because of their true running, minimal space requirements and low price. Space and price considerations make the use of special bearings with non-standard dimensions unusual. Table 11.1 shows various types of bearing, their advantages and disadvantages, and their uses in motor vehicles.

Idler gearwheels are usually supported on single-row or double-row *needle cages* on the transmission shafts. In the case of coaxial countershaft transmissions, the main shaft runs in the drive shaft (Figure 8.1a). This type of bearing is normally fitted with a *roller cage*, and is known as a *stub shaft bearing* or *pilot bearing*.

11.1.2 Bearing Design

Vehicle transmission bearings are designed for reliability in service. This means that they are designed to be reliable for a particular period, subject to typical operational stresses. Bearings are classified as "A" components in the "A, B, C" analysis. "A" components (e.g. bearings, shafts, gearwheels) can be derived by service life calculation or rather service estimation (see also Section 16.2.2 "Qualitative Reliability Analysis" and 7.4 "Operational Integrity and Service Life").

Bearing forces in transmissions result chiefly from the gear-tooth forces of the gearwheels mounted on the shafts. To determine the necessary load profiles, see Section 7.4.2 "Load Profile and Enumeration".

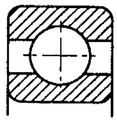
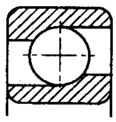
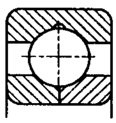
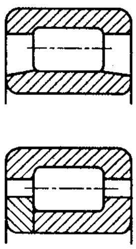
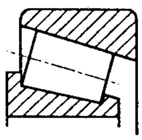
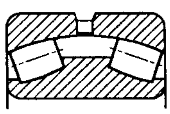
The dynamic capacity of roller bearings is calculated using a fatigue calculation to German standard DIN ISO 281. The cause of failure is attributed to pitting. The service life formula for roller bearings is:

$$L_{10} = L = \left(\frac{C}{P}\right)^p \qquad (10^6 \text{ revolutions}) \; . \qquad (11.1)$$

- L_{10} nominal service life in millions of revolutions, reaching at least 90% of a large number of like bearings,
- p service life exponent ($p = 3$ for ball bearings, $p = 10/3$ for roller bearings),
- C dynamic contact figure in N (given in the bearing catalogues),
- P equivalent bearing load in N, Equation 11.2.

The service life formula for roller bearings is based on the Wöhler curve equation (Equation 7.21). The relations are shown again in Figure 11.1. As regards the load σ_i and the bearing load P, normally shown in the Wöhler diagram, the relation $\sigma_i \sim \sqrt{P}$ applies.

Table 11.1. Advantages and disadvantages of some roller bearings

Type of bearing	Advantages (+) / Disadvantages (-)	Applications
Deep groove ball bearing 	+: Radial and axial load capacity Insensitive to angular deviation Simple installation, no adjustment Simple design, cost-effective –: Low load capacity, high loads require large external diameter Sensitive to dirt	Manual gearboxes Differential gears Transaxle units Passenger car, e.g. Fig. 12.1 Comercial vehicle, e.g. Fig. 12.8
Angular ball bearing 	+: High radial and axial load capacity Tight ways in axial and radial direction –: Radial load generates axial reaction forces	Manual gearboxes Pinion shafts Passenger cars, e.g. Fig. 12.1 Comercial vehicles, e.g. Fig. 12.10
Four-point bearing 	+: High radial and axial load capacity Small physical width –: Low axial clearance required	Mostly used just as a thrust bearing
Cylindrical roller bearing 	+: High radial load capacity Can be used without inner race Suitable types have axial load capacity Easy to disassemble and fit High resistance to dirt –: Sensitive to angular deviation Expensive	At highly stressed bearing points Manual gearboxes Transfer boxes Passenger cars, e.g. Fig. 12.1 Comercial vehicles, e.g. Fig. 12.7
Tapered roller bearing 	+: High radial and axial load capacity Inner race with roller set and outer race can be fitted seperately Simple to fix to shaft and in housing Cost-effective –: Bearing clearance adjustment when fitting Sensitive to skew position Mutual interaction Different heat expansion coefficients of shaft/housing affect bearing clearance	In pairs in manual gearboxes final drives, steering gear Passenger cars, e.g. Fig. 12.6 Comercial vehicles, e.g. Fig. 12.8
Self-aligning roller bearing 	+: High radial and axial load capacity Compensate for angular misalignment and shaft displacement –: Expensive	Transfer boxes

The dynamic equivalent load P for combined loading is derived from

$$P = X\,F_r + Y\,F_a\,. \tag{11.2}$$

F_r constant radial bearing load in N,
X radial factor,
F_a constant axial bearing load in N,
Y axial factor.

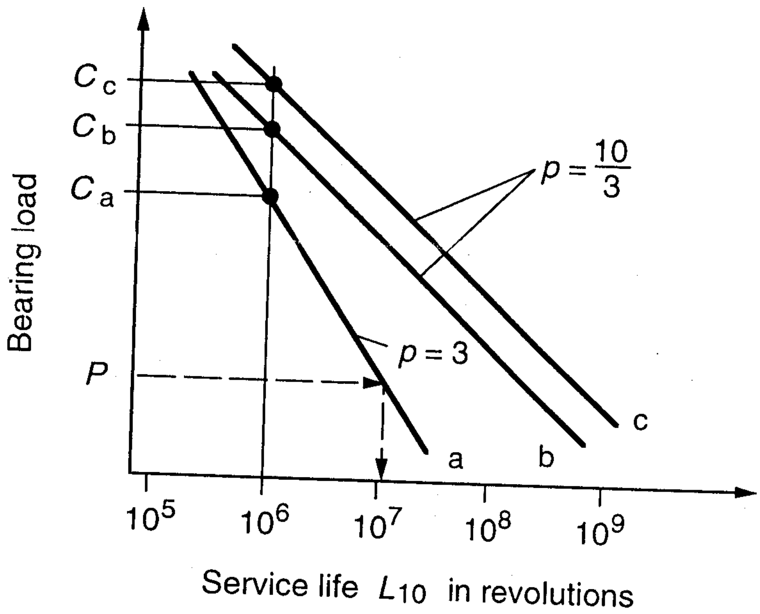


Figure 11.1. Nominal service life L_{10} – various types of bearing, 10%-probability of failure.
a) Deep groove ball bearing
b) Cylindrical roller bearing
c) Tapered roller bearing

The service life equation can also be represented in the following simplified form:

$$f_L = \frac{C}{P} f_n \,. \tag{11.3}$$

The service life function f_L is derived from

$$f_L = \sqrt[p]{\frac{L_h}{500}} \tag{11.4}$$

and the rotation speed factor f_n from

$$f_n = \sqrt[p]{\frac{33\,\frac{1}{3}}{n}} \,. \tag{11.5}$$

The service life L_h is entered in h and the speed n in 1/min. The special features of calculating floating and fixed bearings, and the individual calculation factors for the various types of bearing, are given in the applicable roller bearing catalogue.

A roller bearing is subjected to differing loads and speeds over time in a motor vehicle (load profile). The service life factor ($f_{L,1}$ to $f_{L,n}$) is initially calculated for each individual operating condition without taking into account the time fraction, with the operating state in question assumed to be a continuing state. The actual service life factor is then calculated taking into account the time fractions (q_1 to q_n), as follows:

$$f_L = \sqrt[3]{\frac{100}{\frac{q_1}{f_{L,1}^{\,3}} + \frac{q_2}{f_{L,2}^{\,3}} + \frac{q_3}{f_{L,3}^{\,3}} + \ldots + \frac{q_n}{f_{L,n}^{\,3}}}} \,. \tag{11.6}$$

The effective service life can differ from the calculated bearing service life if operational loads and speeds are not exactly known, if there are variations in loads or other variables such as inadequate lubrication, installation and assembly errors, or if dirt impairs the bearings environment. Table 11.2 gives common service life factors f_L for vehicles.

Table 11.2. Required service life factors of various motor vehicles [11.2]

Type of vehicle	Service life factor f_L
Motorcycle	0.9–1.6
Light passenger car	1.4–1.8
Heavy passenger car	1.0–1.6
Light truck	1.8–2.4
Heavy truck	2.0–3.0
Bus	1.8–2.8

Computer programs specially designed for gear calculations determine the nominal bearing service life from typical transmission load profiles [11.1]. Additional parameters are recorded and taken into account, such as:

- axial clearance or pre-load in the bearing, and load zones subject to concomitant change,
- hertzian contact stresses, pressure ellipses and edge stresses,
- edge freedoms in the contact geometry.

Greater certainty in the robust design of roller bearings in vehicle transmissions can be achieved by using refined service life calculation methods. This makes it possible to use smaller roller bearings, thus saving energy and weight [11.3].

The metallic, mineral and organic impurities contained in the oil have a major influence on bearing service life. Rolling contact with these impurities produces impressions in the tracks, depending on the type of particle. Each subsequent rolling contact produces increased stress in the area of the impression, leading to early fatigue of the track. Especially in the case of small ball bearings, this leads to substantial reductions in service life. This can be countered by choosing suitable heat treatment processes and materials, and by installing dirt protected bearings [11.4]. Clean bearings are bearings sealed on both sides and packed with grease. Their service life is longer than bearings with transmission oil flowing directly through them (see also the 5-speed transmission in Figure 12.1 and the associated versions). Typical bearing damage resulting from installation and operating errors are [11.5]:

- *Scoring*
  In the case of cylinder roller bearings, by tipping the rimless ring on edge, or because of insufficient bearing clearance.
- *Cavitation*
  In ball bearings when the static load of the bearing is exceeded.
- *Grooving*
  In different types of roller bearings as a result of small swivelling movements, or because of shocks when at rest.

Axial Load Capacity of Radial Cylindrical Roller Bearings

Certain cylindrical roller bearing designs can handle large radial forces as well as high axial forces. They can thus be used as fixed bearings or as supporting bearings, always assuming that $F_a \leq 0.4\, F_r / i$ (where i = number of rows of rollers). The axial load capacity depends on the size and bearing capacity of the contact surfaces between the roller faces and the bearing rims. It is however also affected by running speed, load duration and by lubrication (lubricant, quantity, and viscosity). Calculation of axial bearing capacity is based on the frictional power arising at the sliding contact faces. The axial continuous stress of a cylinder roller bearing can be determined using the following equation [11.6]:

$$F_{\text{a, perm}} = k_S\, k_B\, d_M^{\,1.5}\, n^{-0.6} \le F_{\max} = 0.05\, k_B\, d_M^{\,2.1} . \tag{11.7}$$

$F_{\text{a, perm}}$	permissible axial load in N,
$F_{\text{a, max}}$	axial ultimate load in N,
k_S	coefficient dependent on lubrication method, given in roller bearing catalogue (k_S = 7.5–15),
k_B	coefficient dependent on bearing model, given in roller bearing catalogue (k_B = 4.5–30),
d_M	average bearing diameter $(D + d) / 2$ in mm,
n	operating speed in 1/min.

When using radial cylindrical roller bearings for axial stress, it is essential to ensure precise production and installation of the bearing points, so that the bearing rims under load are supported over the whole locating face as far as possible.

11.1.3 Design of Roller Bearings

Please refer to the design examples in Chapter 12 for bearing design.

Selector Gearbox

The bearings used in selector gearboxes are mostly deep groove ball bearings, cylinder roller bearings, tapered roller bearings and four-point bearings. For production reasons, the deep groove ball bearing type with a groove in the outer ring is preferred. This means that there can be four holes right through the housing. The axial fixing is then by means of a locking ring in the groove.

If there are high axial forces in the transmission, the radial and axial forces must be separately absorbed in the fixed bearing. Then a cylinder roller bearing is often fitted to absorb the radial force, in combination with a deep groove ball bearing or four-point bearing to absorb the axial force.

Countershafts often have floating mountings, i.e. both bearings are capable of absorbing axial and radial forces. When cylinder roller bearings are used, their axial bearing capacity has to be borne in mind.

With shaft mountings, the inner ring is subject to radial load, and must therefore have the tighter fit. But the outer ring should also not have a loose seating because of alternating load.

Often needle cages and roller cages are fitted instead of complete roller bearings because of space constraints. The machining tolerances of the shaft seats or housing drill holes are related to the radial clearance required. The contact surfaces must at all events be hardened. Table 11.3 gives reference values for shaft and housing tolerances.

Idler gear bearings require special attention. Because of the gearwheel design, for example on account of the synchroniser hub, there is often eccentric application of force to the gearwheel. The resultant tipping movement stresses the needle cages of the idler gears (see also Figure 8.10).

Until the 1960's, idler gears were predominantly fitted with plain bearings; they were later fitted almost exclusively with roller bearings. Current practice is to use plain bearings again for idler gears that are non-critical, for reasons of cost.

Table 11.3. Reference values for transmission bearing fits

Type of bearing	Shaft	Housing
Deep groove ball bearing	k6	J6–K6 (steel or cast steel) M6–N6 (light alloy)
Cylindrical roller bearing	k6–m6	K6–M6 (steel or cast steel) N6–P6 (light alloy)
Roller bearing cages, roller cages	g5–g6	H6
Four-point bearing (for absorbing axial load)	k5	E8
Tapered roller bearing: Incidence of the inner ring Incidence of the outer ring	 h6–j6 k6	 M6–N7 J6

For idler gears in passenger car transmissions running on roller bearings, single-row needle cages are mostly used. Because of the larger widths of the gearwheels in the case of commercial vehicle gearboxes, two-row needle cages are normally used. Two-row needle cages with a one-part cage are very common. Two-row needle cages with two separate cages provide superior bearing properties, but are more expensive.

In coaxial countershaft transmissions, the main shaft runs in the drive shaft with a *stub shaft bearing (pilot bearing)*. These bearings normally use roller cages.

In the case of adjustable bearing arrangements, it is important to ensure the correct degree of adjustment, since heat expansion has to compensated for by clearance or pre-set, depending on the bearing arrangement. For example large fluctuations in bearing clearance can occur in bearing arrangements with tapered roller bearings installed in light-alloy housings, because of the difference in rates of heat expansion of the shaft and the housing [11.7]. A recommended value for the X arrangement is axial play of 0.05 mm per 100 mm distance between bearings, and for the O arrangement a pre-set which gives a bearing friction torque of 1 to 2 Nm, depending on the size of bearing. This does not apply to tapered roller bearings arranged in pairs to form a fixed bearing; in this case there should be as little clearance as possible.

Final Drives

With front-wheel drive transverse-mounted engines, the final drive consists of a helical spur gear pair, and the above engineering design guidelines apply to its bearing arrangement likewise.

Two pinion shaft configurations are possible for transverse-mounted engines and bevel-gear final drive. In one case the pinion is overhung, and in the other case mounted between the bearing points. With an overhung pinion arrangement, the gap between the two bearings should be at least 2.5 times the distance between the pinion and the first bearing. It is normal to use an adjustable bearing arrangement or a fixed/floating arrangement in the overhung configuration. Where the pinion is supported in the middle, only the fixed/floating arrangement is used. The crown-wheel bearing is generally in the form of an adjustable bearing arrangement.

The same dimensions as shown in Table 11.2 apply as fitting guidelines for the individual bearings. For the two-row angular ball bearings also used with axle gearboxes, the tolerance of the shaft should be k6 and that of the housing drill-hole J6. Final drives are systematically reviewed in Section 6.9. Section 12.5 sets out some design examples.

11.2 Lubrication of Gearboxes, Gearbox Lubricants

A tribological system consists of the following three components: Base body (e.g. rolling body), mating body (e.g. bearing shell), and an intermediary (e.g. lubricating oil). There is relative movement between the base body and the mating body. Lubricants are design elements, whose function is to keep the base bodies and the mating bodies apart under all loads [11.8]. Regardless of system conditions, dry friction, mixed friction and hydrodynamics can arise (Figure 11.2).

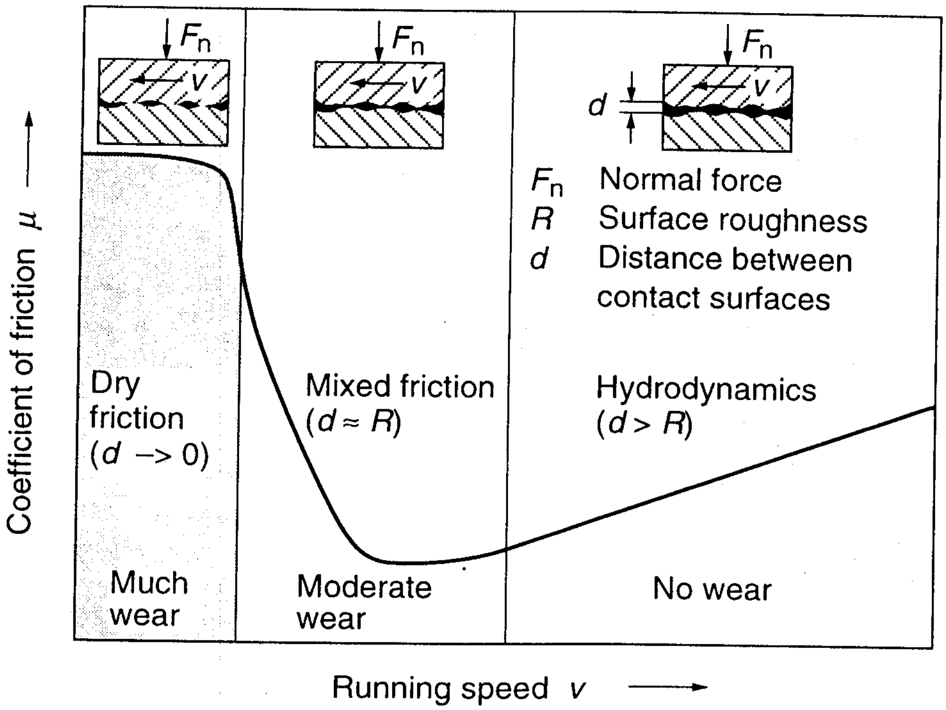


Figure 11.2. Relation between coefficient of friction and running speed [11.8] ("STRIBECK curve"; STRIBECK : Basic research into plain bearings)

As part of the move towards reducing vehicle weight, attempts are being made to reduce the weight of gearboxes, and thus the quantity of lubricant. An additional requirement is that it should not be necessary to change the lubricant throughout the vehicle's service life (lifetime lubrication). In summary, the lubricant must fulfil the following functions:

- reduce friction and wear (saving energy),
- prevent possible damage, or prevent or delay further damage where mechanisms are already damaged,
- dissipate heat,
- create hydrodynamic oil wedges,
- form barrier layers in the mixed friction zone,
- protect materials used in the gearbox against corrosion,
- non-aggressive to seals and paintwork,
- good dirt shifting/cleaning,
- good dirt removal capability,
- water separation,
- stability at high and low temperatures,
- resistant to ageing,
- low cost.

In contrast to slow-running, grease-lubricated industrial gear units, vehicle transmissions are oil-lubricated. Fluid media are better suited for creating the necessary hydrodynamic bearing film than solid materials (greases or pastes). The constant oil circulation also provides better heat dissipation from the stressed components. It is easier for the design engineer to ensure all the various lubrication points in the gearbox are washed with lubricant. It is also easier to remove impurities in the gearbox through appropriate filter systems with oil than with pasty materials.

11.2.1 Bearing Lubrication

Bearings in selector gearboxes are normally lubricated by the oil spray created by the gearwheels in the housing. Traps and guide channels have to be provided to feed bearing points in unfavourable locations. Good lubrication is particularly important with cylinder roller bearings subject to axial load, since the oil also has to dissipate the heat created by friction. The same applies to tapered roller bearings, and in this case it also has to be borne in mind that the oil has to flow from the smaller taper opening to the larger one. Gearwheel debris impairs the service life of roller bearings, so an oil circulation system with an oil filter is beneficial.

The bearings and gearwheels of bevel gear final drives are lubricated exclusively with a high-pressure hypoid oil. Whilst the bearings of the crown gear shaft are well lubricated by the oil spray, flow channels often have to be provided to take the oil to and from the pinion shaft.

11.2.2 Principles of Lubricating Gearwheel Mechanisms

When gear teeth mesh, two types of movement take place: rolling and sliding movement. The running speed is at its maximum at the beginning A or the end E of tooth contact, i.e. at the root or the tip of the tooth; at the pitch point C it is zero (Figure 11.3). The wear increases as the proportion of sliding increases.

At the tooth flanks, the most favourable lubrication conditions arise around the pitch point, whilst conditions are less favourable at the tooth tips, because of the meshing impact and high temperatures resulting from higher running speed. Figure 11.3 shows typical friction zones on the tooth flanks.

- *Boundary friction*
  There is dry friction. The tooth flanks are only separated by a boundary layer of chemical reaction products a few nanometres thick, intended to prevent metal-to-metal contact (boundary lubrication).
- *Mixed friction*
  The tooth flanks are only partially separated by a film of lubricant. There is liquid friction and dry friction at the same time. Where the surfaces touch, there is boundary lubrication.
- *Fluid friction (hydrodynamics)*
  The tooth flanks are completely separated by a film of lubricant. There is elasto-hydrodynamic lubrication.

The lubricant thus has a two-fold effect in reducing friction and wear at the tooth flanks [11.10–11.13]:

1/ *elasto-hydrodynamic lubricant film:* EHD lubricant film,
2/ *chemical protective film:* frequently also referred to as boundary layer or reaction layer.

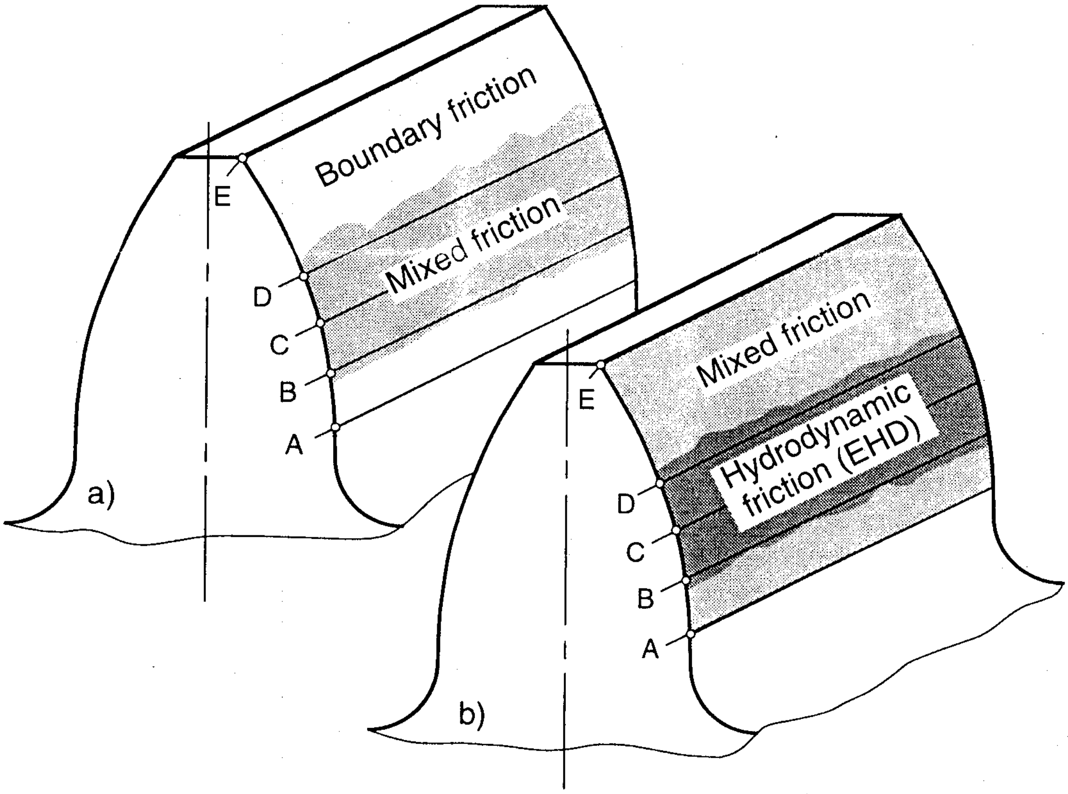


Figure 11.3. Typical friction zones on tooth flanks at high contact pressures [11.9, 11.10]. **a)** Low circumferential speed (up to 5 m/s); **b)** High circumferential speed. *A* First point of contact; *B* Internal single contact point; *C* Pitch point; *D* External single contact point; *E* Last point of contact

1/ Elasto-Hydrodynamic Lubricant Film

The lubrication process is discontinuous since the bearing film has to be re-established for each meshing action. Hydrodynamic lubrication theory of plain bearings is not applicable because of the high contact pressures for the toothing. Elasto-hydrodynamic lubrication theory has to be applied. This theory takes into account the pressure viscosity of the oil and the elasticity of the tooth flanks. Elasto-hydrodynamic lubrication is characterised by two fundamental features:

- The viscosity of the oil film increases erratically because of the high surface stress.
- Because of the high surface stress, there is elastic deformation at the tooth flank contact points. The crowned engaged tooth flanks flatten under load.

The tooth flanks are kept out of direct contact by the increase in contact surface, and the capacity of the lubricant film related to its viscosity.

The form of the lubrication gap and the pressure profile in the contact zone is shown in Figure 11.4. A pressure peak is formed before the end of the lubrication gap, and the end of the gap at the lubricant outlet is contracted. The thickness of the film of lubricant depends chiefly on the toothing geometry, the viscosity of the lubricating oil, the circumferential speed, the contact pressure, the tooth flank temperature and the surface roughness.

2/ Protective Chemical Film

If the surfaces touch with mixed friction or boundary friction, the wear-reducing additives in the oil come into effect, forming a chemical protective film on the tooth flanks [11.15, 11.16].

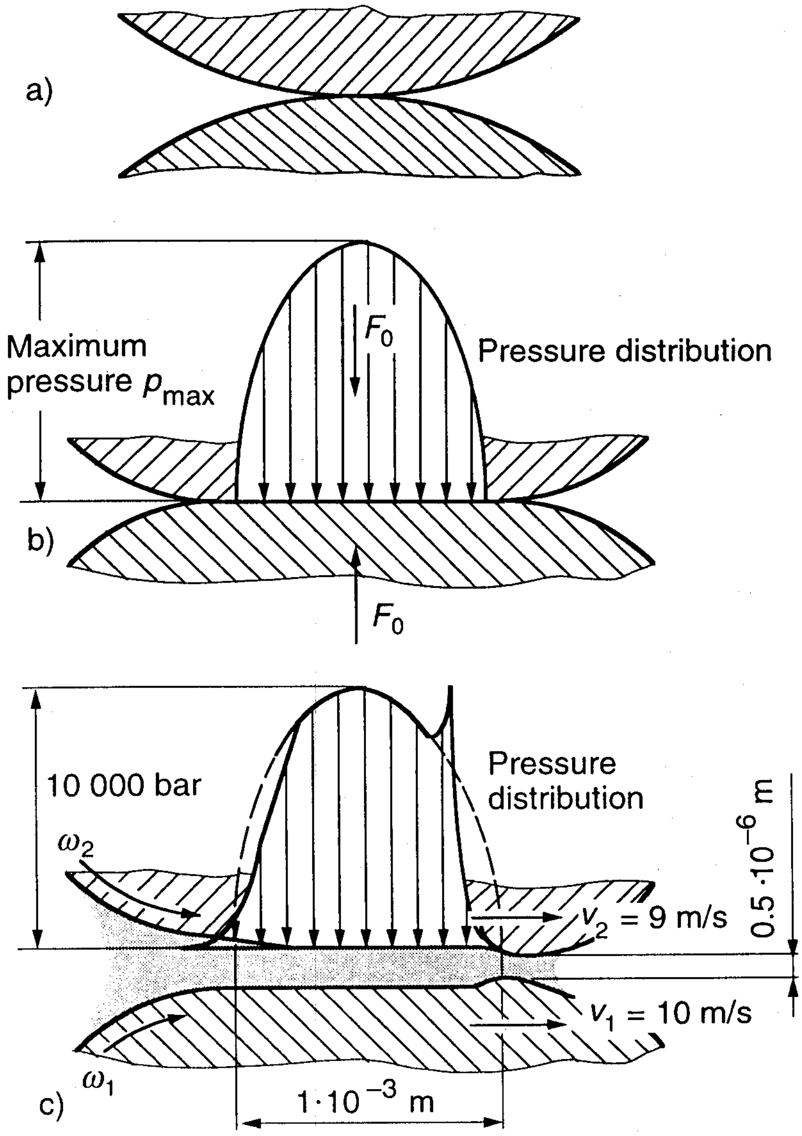


Figure 11.4. Formation of the lubrication gap with elasto-hydrodynamic lubrication [11.14].
a) Without load
b) Under load without movement
c) Under load and rotating with film of lubricant

The wear-reducing additives are called EP (Extreme Pressure) additives. The alternative term AW (Anti-Wear) has gained acceptance, referring directly to the function of these materials.

Put simply, mild EP/AW additives are first physically absorbed, and only then in the second stage (under load) are chemical reaction layers formed. They prevent the contact surfaces bonding by forming surface reaction layers with lower shear strength than in the case of pure materials [11.17] (Figure 11.5).

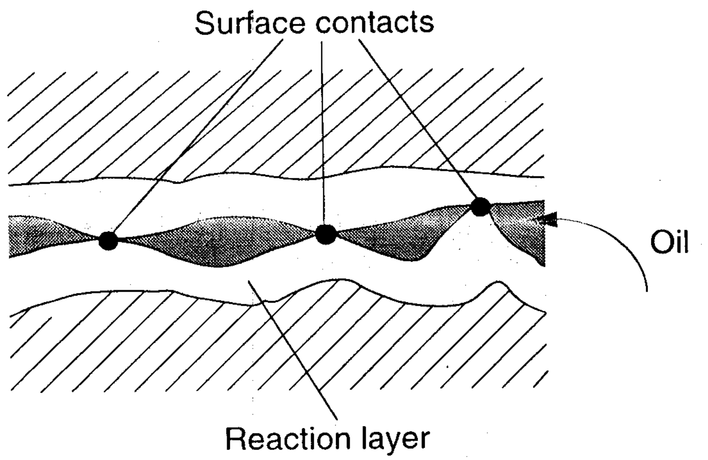


Figure 11.5. Mixed friction, surface reaction layer – protecting the metal layer beneath

Highly reactive EP/AW additives lead to measurable reaction layers even before the trigger temperature is reached. They then quickly re-generate the abraded reaction layer at very high stresses. If the tooth flanks heat up under friction, this boundary layer will be destroyed if a characteristic temperature for the lubricant is exceeded.

The composition of the reaction layers depends very much on the mechanical conditions, materials, temperature, the lubricant base and the additives. Investigations show that the chemical reaction between additive and tooth flank is the crucial factor in the scuffing load. The scuffing load speed curves are crucially dependent on the additive [11.18].

11.2.3 Selecting the Lubricant

In normal operation, the oil temperature in the sump of passenger car and commercial vehicle gearboxes is approximately 60–90 °C. Under extreme conditions, for example on mountain roads with a trailer, the oil sump temperature can reach approximately 110 °C. Oil temperatures may reach 130 to 160 °C locally.

Modern lubricants are fundamentally made up of several constituents. They consist of a base and appropriate additives. Mainly mineral oils are used as the base for manufacturing gearbox oils. The various mineral oils are distinguished by their viscosity index VI (German standard DIN ISO 2909). The viscosity index describes the high/low temperature properties of the base oil, i.e. remaining sufficiently liquid at low temperatures, without becoming excessively liquid at high temperatures. Good mineral oils have a VI of approximately 95 to 105, and high-quality oils achieve a VI of up to 150. Where extremes of temperature are anticipated (below –20 °C and above 140 °C), synthetic oils are used.

The characteristics of gearbox oils are affected to a significant degree by the additives and packages. The term "package" refers to a gearbox oil constituent made up of several additives comprising approximately 2–10% of the total volume. The most common additives for gearbox lubricants in accordance with [11.19] are:

- EP (Extreme Pressure) additives for improving high-pressure characteristics,
- corrosion inhibitors for preventing rust, verdigris and similar harmful products of oxidation,
- D/D (Detergent/Dispergent) additives for removing dirt,
- friction modifiers for reducing friction and wear,
- VI improver to improve high and low temperature performance.

This enables lubricants with different characteristics to be created by appropriate choice and composition of additives. The action of the various additives should be complementary.

11.2.4 Selecting Lubricant Characteristics

Viscosity

Probably the most important characteristic of gearbox oils is their flowability, or *viscosity*. Viscosity describes the internal friction of a fluid. A distinction is made between dynamic viscosity η and kinematic viscosity ν; it is almost always the kinematic viscosity that is quoted. It is calculated as the quotient of dynamic viscosity and density of the oil ρ

$$\nu = \frac{\eta}{\rho} \,. \qquad (11.8)$$

Lubricating oils are classified into viscosity groups. Both automotive engine oils and transmission oils are usually given SAE ratings (Figure 11.6). The appropriate viscosity for a gear mechanism can be selected in accordance with German standard DIN 51509 [11.20]. The kinematic nominal viscosity is then determined as a function of the surface stress and the running speed.

A fluid bearing film can be created by applying the EHD theory, even in friction combinations with unfavourable lubrication relations and large contact pressures, such as those encountered in meshing. The thickness of the film of lubricant can be determined by applying this theory, as a function of the stress, the circumferential speed, the effective temperature and the viscosity of the lubricant [11.21]. This film thickness can be regarded as adequate when it is larger than the average surface roughness of the tooth flank surface. Alternatively the lubricant viscosity required to provide a bearing film of sufficient thickness for a given gear/tooth system can be determined, where the operating conditions are known.

The gears resistance to scuffing and pitting also improves as viscosity increases. Damping capacity increases and load losses decrease. If the viscosity of the lubricating oil is too great, there are also negative effects. For example friction losses and thus temperatures can become very large [11.22]. Lower viscosity improves low temperature fluidity, air release and cooling capacity. Idling losses are reduced.

Selecting the viscosity is always a compromise. It is frequently determined by other components in the lubrication system, such as the torque converter.

Viscosity/Thermal Behaviour

The viscosity of lubricating oils decreases exponentially as temperature increases. The viscosity/temperature profile of mineral oil based lubricating oils (a) is a straight line in the log-log UBBELOHDE diagram (Figure 11.7). The change in viscosity depends on the oil base.

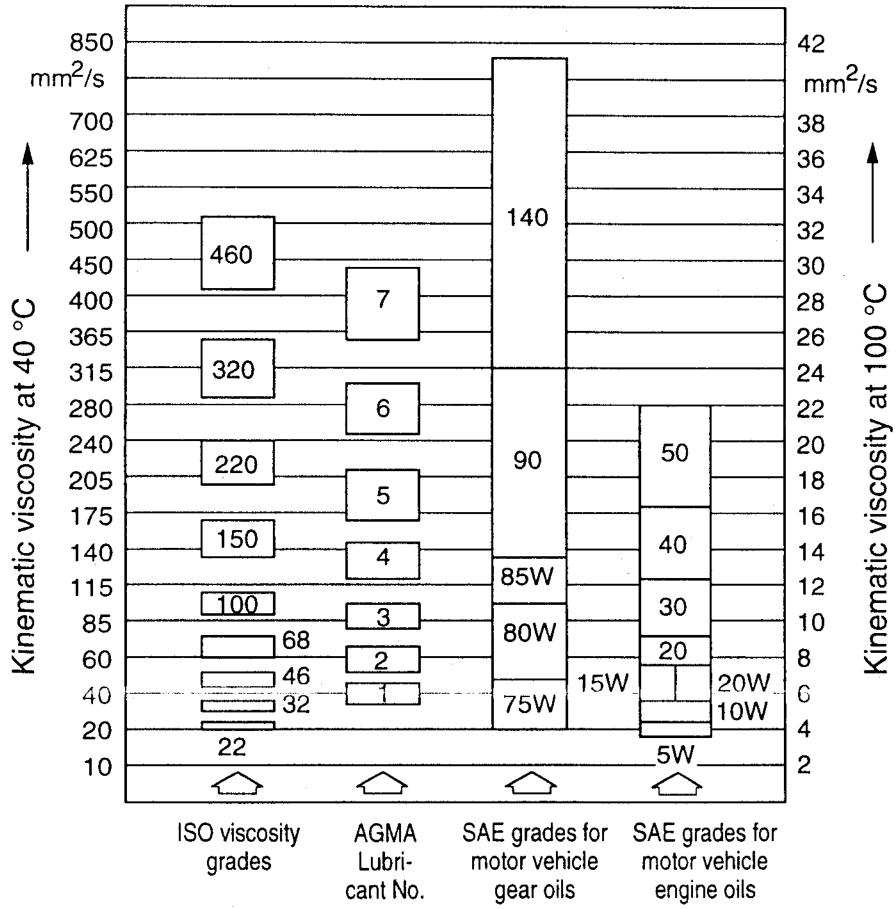


Figure 11.6. Comparison of various systems of viscosity classification (basis: VI = 100) [11.14]

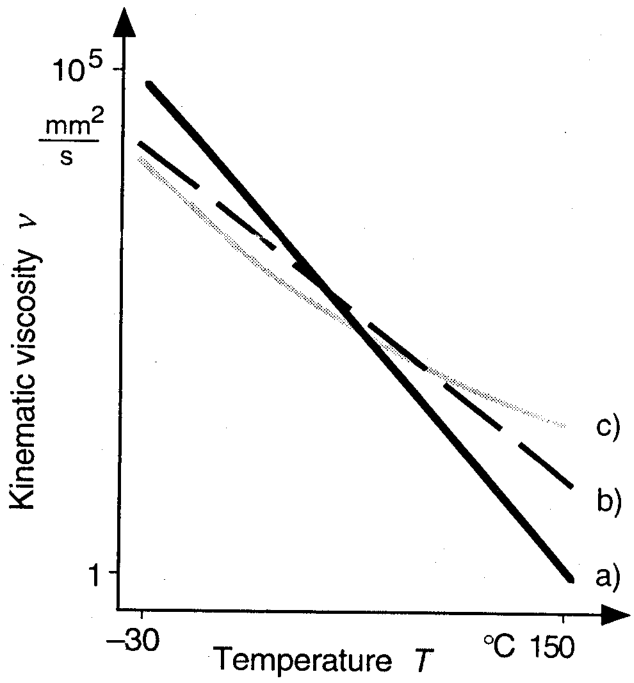


Figure 11.7. UBBELOHDE diagram (log-log scale). Basic viscosity/temperature profiles of various gearbox oils:
a) Mineral gearbox oil
b) Poly-α-oleofin based gearbox oil
c) Polyglycol based gearbox oil [11.6]

Synthetic lubricating oils based on poly-α-oleofins (b) also give straight lines on a UBBELOHDE diagram, whereas polyglycol-based oils (c) appear as curves [11.6]. The empirical "UBBELOHDE-WALTHER formula" (German standard DIN 51563) is derived as

$$m = \frac{W1 - W2}{\log T_2 - \log T_1} \ . \qquad (11.9)$$

The variables in Equation 11.9 are as follows:

- m slope; normal values are $2 < m < 5$,
- W log-log $(\nu + 0.8)$,
- ν kinematic viscosity,
- T test temperature in K.

The angle of the straight line, the slope m, is a measure of the temperature dependence of the lubricating oil. The characteristic value of the viscosity/thermal behaviour is given by the viscosity index VI (German standard DIN ISO 2909).

Viscosity/Pressure Behaviour

Very high Hertzian stresses arise briefly in vehicle transmissions during meshing (in excess of 2000 N/mm^2), and the dynamic viscosity η_p of the oil rises

$$\eta_p = \eta_0 \, e^{\alpha \, p} \ . \qquad (11.10)$$

The variables in Equation 11.10 are as follows:

- η_p dynamic viscosity at working pressure in Pa s,
- η_0 dynamic viscosity at 1 bar,
- α viscosity-pressure coefficient of the oil in Pa^{-1},
- p working pressure in Pa.

The pressure coefficients of common types of oil are in the range $0.7 \leq \alpha_{25°C} \leq 8 \ Pa^{-1}$.

Pour Point and Flash Point

The pour point describes the flow properties at low temperatures. The pour point must be at least 5 K below the lowest operating temperature.

The flash point can be ignored in all but a few critical high-temperature applications that do not apply in automotive engineering.

Foaming Tendency and Air Release

A distinction has to be made between surface foam and bubble foam. Surface foam can be prevented by design measures (baffles, settling chambers). Bubble foam is prevented by adding a small amount of silicone oil, which however impairs the air release characteristics.

Water Demulsibility

In order to prevent foam formation, any water entering the oil should not emulsify with it.

Ageing Characteristics, Oxidation Characteristics

Oil ageing is a chemical change that takes place under the influence of high temperatures and the catalytic effect of metals. Ageing is caused mainly by oxidation of oil molecules. The high flank temperatures arising in gear mechanisms, the high turbulence of the oil, and the impurities present in the oil are major factors in ageing. The resistance of gearbox oils to oxidation can be improved by oxidation inhibiting additives.

Corrosion Protection

The corrosion protection characteristics of the oil can be improved by additives that create a protective film on the metal surfaces in the gearbox, and/or neutralise the corrosive decomposition products formed in the process of oil ageing.

Seal Compatibility

Seals must not change in terms of their material characteristics under the influence of gearbox oils. They must for example not shrink, swell or become brittle. Nor should any material components be deposited which could lead to impairment of the oil's characteristics and thus of the gearbox's function.

11.2.5 Lifetime Lubrication in Vehicle Gearboxes

Lifetime lubrication of vehicle gearboxes is of particular significance in the light of the discussion about more environment-friendly and cost-effective products. Lifetime lubrication offers the following advantages:

- reduced lubricant consumption,
- reduced lubricant costs related to gearbox service life,
- low maintenance costs (down time).

Lifetime lubrication means that oil change intervals extend to the service life of the vehicle. But an oil change after a particular running-in period can probably not be avoided. Lifetime lubrication has already been introduced for some manual passenger car transmissions, but lifetime lubrication for more highly stressed commercial vehicle transmissions and automatic transmissions will only be introduced with new generations of transmission [11.23].

During the service of a transmission, the lubricant changes because of oxidation, decomposition, additive degradation, changing viscosity, and absorption of moisture and particles. This also affects the service life of the gearwheels, bearings and shifting elements. The increased demands on lubricants for lifetime lubrication are already largely satisfied by synthetic oils, given appropriate selection of oil base and additives.

11.2.6 Testing the Scuffing Resistance of Gearbox Lubricants

Two categories of testing machine types are mostly used for experimental investigation of scuffing resistance:

- gearwheel and roller type testing machine,
- two, four and five ball tester.

In the initial phase of the application in the case of the development of new and existing oils and their quality control, costly and time-consuming bench tests are necessary. With the increasing demands on lubricants and friction materials due to light-weight construction and higher power transmission, the scuffing resistance requirements or the friction pairings also increase. Scuffing resistance is finally evaluated on gearwheel test stands. The Integral Temperature Method now provides a satisfactory method for calculating scuffing resistance of production transmissions from gear bench test programs [11.24]. Investigations have shown that the Integral Temperature can also be determined without using a gear test stand, using the test results of the four-ball tester.

Gearbox scuffing resistance is evaluated on the basis of an intermediate, weighted contact temperature (Integral Temperature). The criterion of constancy of the Integral Temperature is regarded as the best method for transferring gear test stand findings to practical performance [11.25]. With this method, the permissible Integral Temperature $T_{\text{int, perm}}$ is determined with a gearwheel torque test rig (FZG-test rig, FZG: "Forschungsstelle für Zahnräder und Getriebebau") for a given lubricant/material combination:

$$T_{\text{int, perm}} = T_{\text{oil}} + a\,T_{\text{fl}} \ . \tag{11.11}$$

The variables in Equation 11.11 are:

$T_{\text{int, perm}}$	permissible Integral Temperature,
T_{oil}	oil bath temperature,
a	constant $a = 1.2$,
T_{fl}	flash (friction) temperature.

This temperature has a constant value for a lubricant/material combination, regardless of the operating conditions. If this temperature is exceeded there is scuffing damage to the gear pair (20% of the active tooth flank height of the pinion shows signs of wear).

If this calculation is carried out with the associated operating parameters for a production gearbox analogously to the calculations of permissible Integral Temperature for the gearwheel torque test rig, then the resultant Integral Temperature value T_{int} must not exceed the permissible temperature value $T_{\text{int, perm}}$ determined with the test stand for scuffing-resistant operation. The following applies for scuffing resistance:

$$S_{\text{s}} = \frac{T_{\text{int, perm}}}{T_{\text{int}}} \geq 1.2 \ . \tag{11.12}$$

The variables in Equation 11.12 are as follows:

S_{s}	scuffing resistance,
$T_{\text{int, perm}}$	permissible Integral Temperature,
T_{int}	Integral Temperature.

The test method conforms to German standard DIN 51354 (FZG-gearwheel torque test rig), i.e. at a constant circumferential speed the stress is increased at intervals until scuffing occurs. Investigations [11.26] have however also shown that as circumferential speed increases, the tooth loading transmitted without wear decreases up to a certain speed. But a further increase in speed results in another increase in scuffing wear resistance. This phenomenon is probably explained by the familiar elasto-hydrodynamics of the lubricant layer formed. The primary requirements for a gearbox lubricant according to [11.22] are:

- good FZG-performance, minimum power level 9,
- good four ball tester welding load value, starts at 3000 N welding load (steel/steel material pairing).

The four ball tester welding load is determined in accordance with German standard DIN 51350. The load-bearing capacity is conditional on effective viscosity and additive action, and increases as four ball tester welding load increases and wear dome size decreases.

A suitable Extreme Pressure additive has a major impact on scuffing resistance, as can be demonstrated with the testing methods described. Scuffing resistance can in some cases be increased fivefold by the use of suitable additives [11.28].

11.3 Gearbox Housing

The gearbox housing houses all the components of a gearbox. The following requirements have to be taken into account when designing the housing:

- absorb the active operational forces and moments,
- guarantee the exact position of the shafts and gearwheels relative to each other in the various operating states,
- ensure good heat conduction and radiation,
- insulating and attenuating the gearbox noises,
- easy to fit and remove,
- rigid construction and good strength characteristics, combined with low weight.

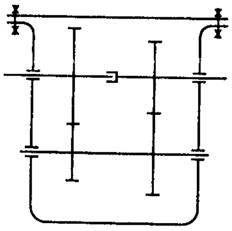
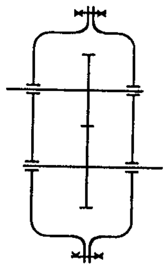
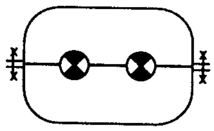
11.3.1 Gearbox Housing Design

The housing can be designed as a classical *trough housing* or as a split housing. Split housings are divided into *end-loaded* and *top-loaded* depending on the position of the plane of osculation or of the shaft. Top-loaded housings are to be found in motorcycles. The types of housing are listed in Table 11.4, with their advantages and disadvantages.

Vehicle gearboxes are predominantly of end-loaded design. The clutch housing can be flange-mounted on the gearbox (3-part housing) or integral to the housing. The latter configuration results in a 2-part housing, which involves lower manufacturing costs.

As a result of the demands to improve performance and at the same time reduce weight, motor vehicle gearbox housings are now increasingly made of diecast light alloy. Diecast aluminium is a good compromise between cost and weight. In order to avoid having thicker walls than cast iron for the same strength and rigidity, the housing has webs. These webs increase rigidity and strength, and at the same time reduce sound reflection from the gearbox housing.

Table 11.4. Types of housing. Advantages and disadvantages

Type of housing	Advantages	Disadvantages
Trough housing 	– Bearing bores easy to produce – Precise production in one clamping operation	– Unfavourable fitting conditions – Fitting cannot be automated
End-loaded housing 	– Rigid housing – Easy to fit – Fitting can be automated	– Expensive production equipment – Critical bores in two clamping operations
Top-loaded housing 	– Precise production in one clamping operation – Very easy to fit – Fitting can be automated	– Reference surface expensive to machine – Not very rigid

Vehicle transmission housing design was systematically investigated [11.29] using the Finite Element Method, and design guidelines for diecast light alloy housings derived:

- ❍ housing webs should always run in the direction of the main direct stress; thus reducing the tensile stresses on the casting by enlarging the supporting cross-section,
- ❍ webs on bearing walls should be arranged in a star shape from the bearing bores, dimensions of the webs relating to the wall thickness t_W:
  - Height $h = (3–4)\ t_W$,
  - Width $b = (1–2)\ t_W$,
- ❍ reverse gear bearing should be strengthened by high webs ($h = (3–5)\ t_W$) mounted at an angle of 0° or 90°,
- ❍ strengthen longitudinal walls with wide webs ($b = (1–2)\ t_W$) with large radius of curvature $r = 1.2\ t_W$, mount webs at an angle of 45° to the longitudinal axis of the gearbox,
- ❍ heavy webs with spacing ($s = (5–15)\ t_W$) achieves good acoustic transmission characteristics.

Figure 11.8 shows a commercial vehicle end-loaded gearbox housing made of diecast aluminium with flange-mounted torque converter clutch housing.

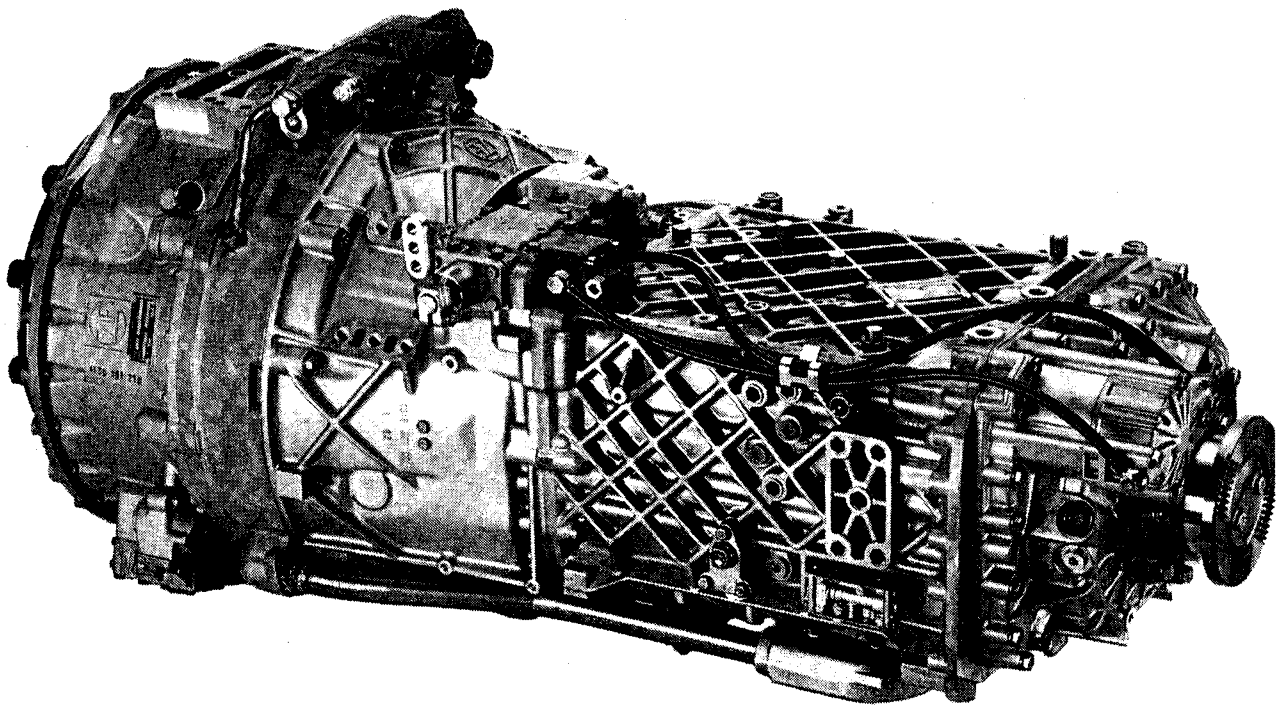


Figure 11.8. Gearbox housing of the WSK 16 S 220 commercial vehicle gear unit (ZF); end-loaded housing

11.3.2 Venting Gearboxes

Vehicle transmissions are oil-lubricated. The seals around the input and output shafts and the gearbox housing are in most applications rotary shaft seals. These sealing rings are designed to seal unpressurised fluids. The amount of air contained in the gearbox means that temperature fluctuations in a completely enclosed gearbox would cause pressure fluctuations. Because of increasing sealing edge contact pressure of the rotary shaft seal, overpressure in the housing can lead to increased heat build-up, wear of the sealing edge, and thus to leakage. As the negative pressure adjusts, it can cause air, water and dirt to be sucked in at the seals because of the falling sealing edge contact pressure.

To prevent the above problems arising at the seals, gearboxes are fitted with vents to enable the pressure to be equalised by air flowing in and out. For reasons of operational reliability and environmental pollution, there must be no emission of oil, oil foam, oil vapour or oil mist from the vents. Water, dust and dirt must be prevented from entering, and the vents must be kept clear.

Venting Modes

The laws of thermodynamics dictate two different operational modes for a breather system, because of the temperature fluctuations (Figure 11.9):

- *Venting*
  When there is excess pressure, air flows out of the housing through the breather system into the environment, because of the rising temperature in the gearbox housing. The flow of air can discharge oil, oil foam, oil mist and oil vapour.
- *Aerating*
  When there is negative pressure, air flows from the environment through the breather system into the housing, because of the falling temperature in the gearbox housing. The flow of air can draw in water, moisture, dust and dirt.

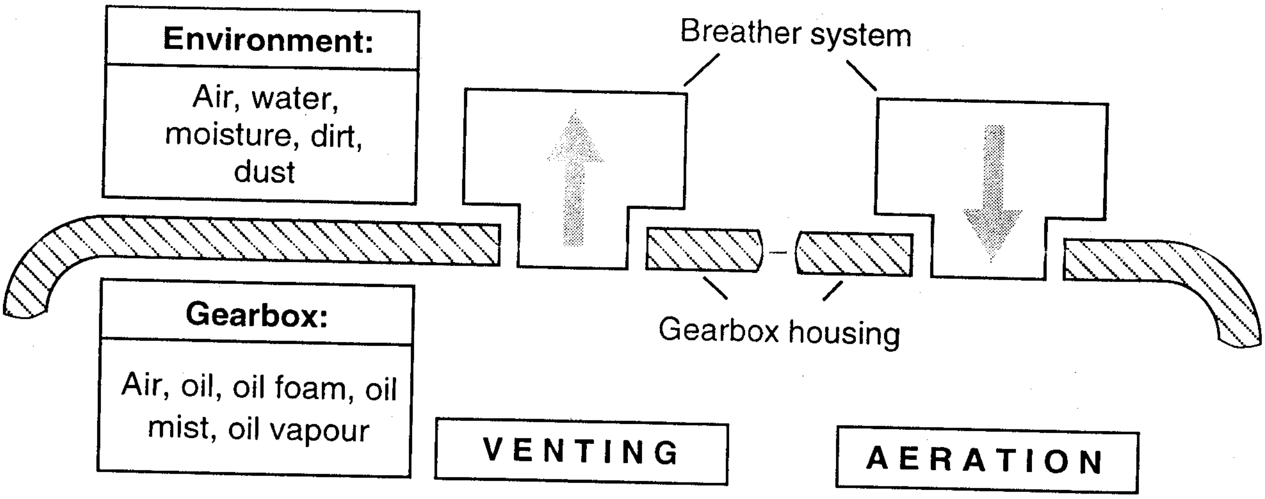


Figure 11.9. Operational modes of a breather system: venting and aeration [11.30]

When non-contact seals are used, e.g. labyrinth seals at the gearbox shaft exits, labyrinth air propulsion can lead to much higher air flows through the breather system.
Non-contact seals should be designed so that they do not accelerate the air flow. Provided this requirement is met, the air flow rate through the breather is restricted to that caused by temperature fluctuations.

Designing Breather Systems

Figure 11.10 shows a typical breather system design. The design of the interior is of particular importance for deflecting oil spray. Many breather systems therefore have a splash guard on the inside. The flow of air is guided through tube extensions, and can be diverted and restricted in several places. Breather systems with valve inserts can also be used. The filters used are made of various materials or forms of material (e.g. flat wire mesh, foam, sintered bronze, filter fabric). Externally there is usually a cap to protect against water splashing. There may also be an externally fitted filter for extreme operating conditions.

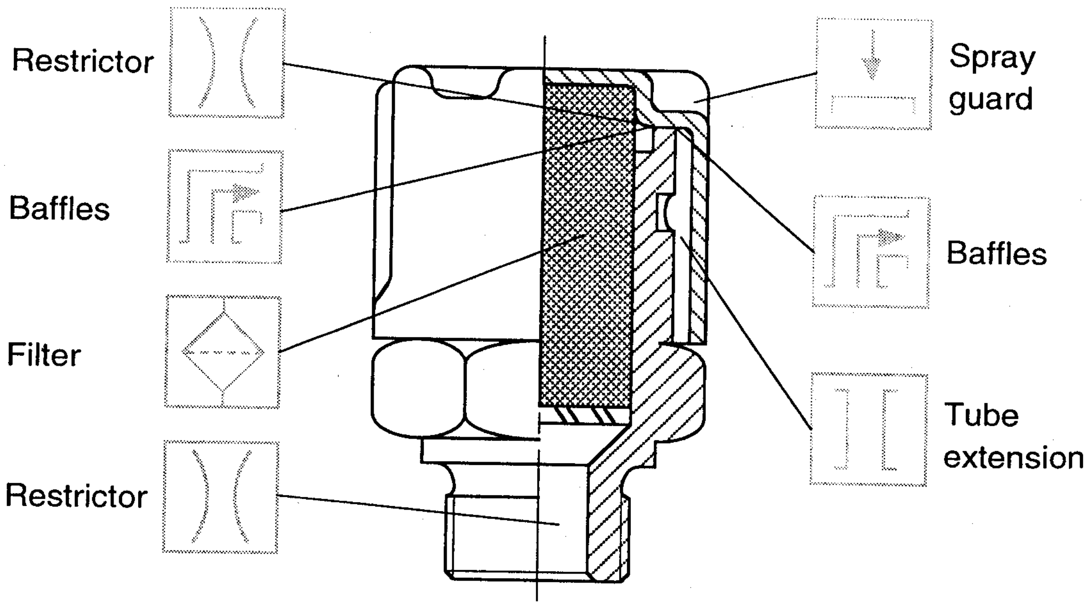


Figure 11.10. Structural elements of a breather system based on the *Hunger 1019 R1/8", shape A*

Most breather systems are made of metal. The various parts of the system are made by machining and metal terming. For complex designs it can be more suitable to use plastic injection moulding. Considerations of weight and cost also play a major role when selecting the material.

Breather systems can generally be fitted externally. There are some special designs in which the breather is combined with other functions.

Structural Elements of Breather Systems

The structural elements listed below are used in current breather systems. The breather system as a whole is made up by mounting these individual elements in series.

External spray guard:

Caps, covers and external deflector surfaces are used to prevent water spray and dirt directly entering the breather system.

Filter:

The function of a filter in the breather system is to retain the dirt in incoming air. The oil mist and oil foam carried by the exhaust air is also deposited on the filter surface. A contaminated filter can lead to an increase in pressure.

Vent valve:

Air is allowed to escape from the gearbox only at a certain pressure, controlled by spring tension or by the weight of a cover. No air can flow in through the vent valve.

Aerating valve:

A spring arrangement with a sealing plate is used to ensure that air can enter the gearbox from outside only when a given negative pressure is reached. Air cannot escape through the aerating valve.

Baffles:

The baffles in the flow path prevent oil spray from being directly emitted, and water spray from directly entering.

Labyrinth:

A labyrinth is made up of various air flow baffles and channels. This has the desired effect of lengthening the flow path in the breather system, and thus increasing the accretion of oil on the internal surface of the labyrinth. This oil can then be returned to the gearbox.

Tube extension:

A tube extension is often necessary to connect the various structural elements. Not only round cross-sections, but also rectangular, square or circular cross-sections can be used. For example the quantity of oil spray reaching the top end of a vertical tube can be decreased using this tube.

Restrictor:

A restrictor in the flow restricts the flow cross-section and thus increases the speed of flow. It is often difficult to distinguish between a restrictor and a tube extension.

Internal spray guard:

Spray guards are fitted to the inside of the breather system to prevent oil spray directly entering the breather system.

Design Guidelines for Breather Systems

Comprehensive bench tests in [11.30] have been carried out to establish the ideal design for various operating conditions (presence of oil spray, oil foam, water and dust). Figure 11.11 shows a possible overall structure of a breather system for these operating conditions.

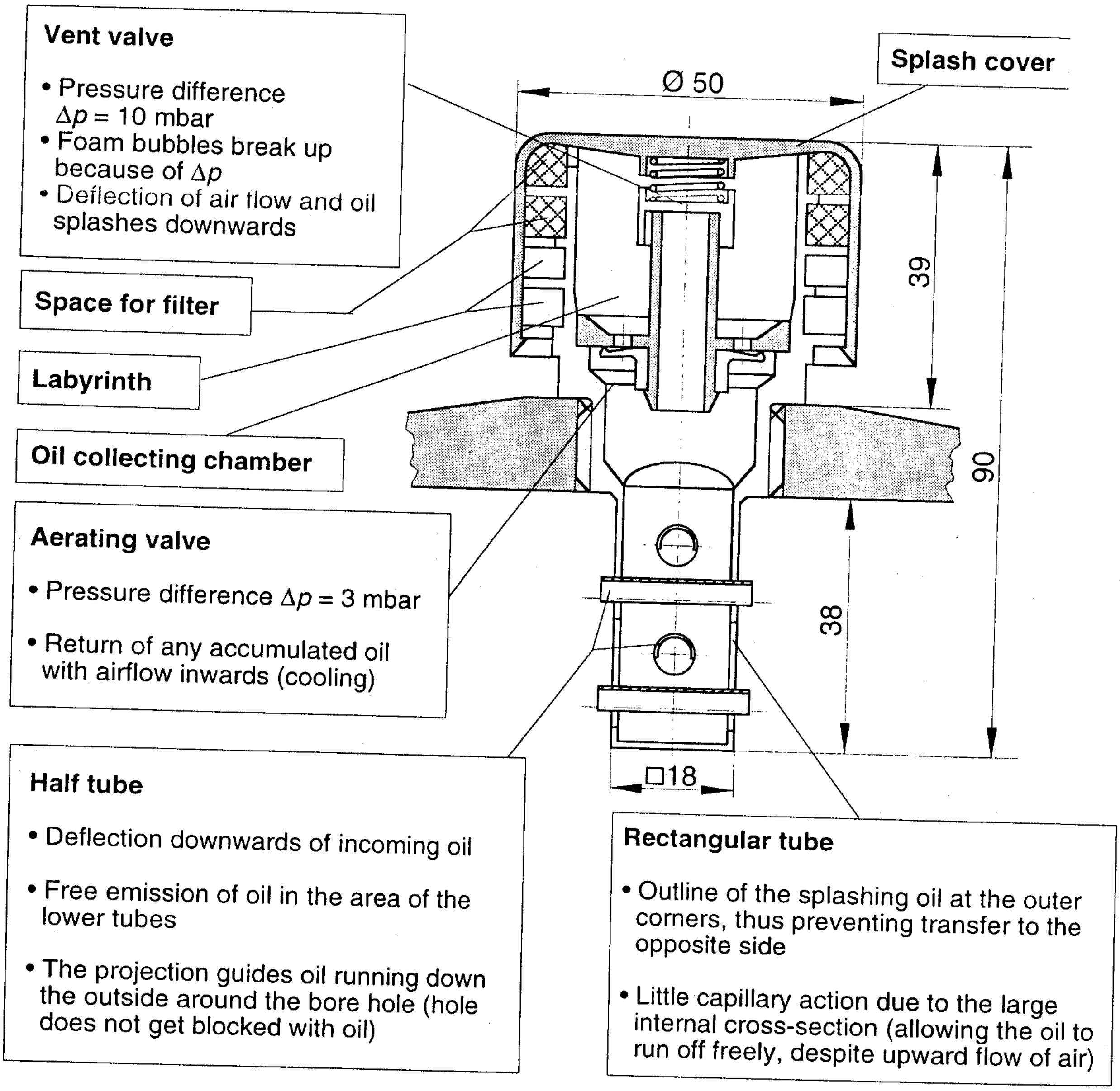


Figure 11.11. Alternative means of venting for different operating conditions [11.30]

If oil splashing against the inner side of the breather cannot be avoided, then the oil spray should be separated from the exhaust air at the inlet on the inner side of the breather. The type of breather system proposed, comprising an intruding rectangular tube with several half tubes, offered the best characteristics for preventing oil escaping when oil splashes against the inner side of the breather.

When there is oil foam on the inner side of the breather combined with an outward flow of air, there is no simple means of preventing an oil leak. In the double valve type of breather, the oil foam emitted through the vent valve, which has dispersed at the vent valve, is held in the breather. When air is drawn in, the oil held in the breather is fed back inside through the aerating valve, thus acting to reduce the oil leak.

Where the ventilation side is subject to external water splashing, the inlet point must not have any small orifices. Especially when there is water running down the outside of the breather system, there is a danger of the orifices being blocked by the retained water. Any incoming flow of air will draw large quantities of water into the gearbox in these circumstances.

Special filter units are a suitable means of preventing dust from entering the gearbox. The filter elements should be fitted in the external upward air intake, but not directly at the point of entry.

The decisive factor for the performance of the breather system is its location in the gearbox. When assessing or designing a breather system, the gearbox system and its environment as a whole must be taken into account. There is no panacea. But it is possible to define the preferred parameters for a breather system on the basis of the information provided.

11.4 Gearbox Sealing

The demands of service life and environmental considerations have made reliable, durable gearbox sealing a major factor. If a seal fails, the repair costs amount to many times the cost of the seal.

There are numerous points to be sealed in vehicle transmissions, such as shaft input and output seals, housing joints, selector shaft output, and speedometer drive. There are three types of seal for these purposes:

1/ seals for static components (e.g. gaskets),
2/ seals for rotating components (e.g. rotary shaft seals),
3/ seals for reciprocating round components (e.g. groove rings).

11.4.1 Seals for Static Components

Radial seals for static components are normally O-rings, radially compressed (Figure 11.12a). Static radial seals (flange seals) can be in the form of O-rings or gaskets. O-ring type flange seals are axially compressed (Figure 11.12b). O-rings are usually made of nitrile butadiene rubber (NBR).

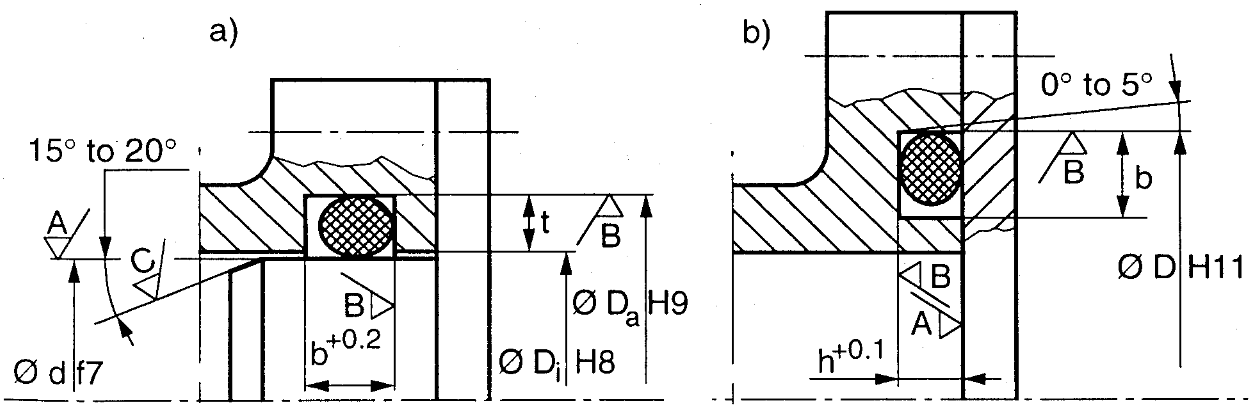


Figure 11.12. Static components sealed with O-rings. **a)** Radial static seal (groove in outer part), O-ring radially compressed; **b)** Axial static sea; (flange seal), O-ring axially compressed

Directions for Installing O-Rings:

- compress as necessary (10 to 20% of the cord thickness),
- use largest possible cord diameter, to compensate for relaxation and production variances,

- ❍ the surface roughness at A, B and C (Figure 11.12) depend on the operating conditions. Further details will be given in the manufacturers catalogues,
- ❍ preferably fit O-rings in the front face to facilitate dismantling (Figure 11.12b),
- ❍ use inserting bevels to avoid damage when installing radially fitted O-rings,
- ❍ avoid twisting radially fitted O-rings when installing,
- ❍ use thrust rings to prevent gap extrusion at high pressures.

Surface seals are used to seal the partition lines of gearbox housings and covers that are bolted together (flange seals) (Figure 11.13). Together with the flanges and the bolted connection, the surface seal forms a *sealing joint*.

Surface seals can be pre-formed seals (*gaskets*) or unformed seals (*jointing compounds*). Jointing compounds assume the required shape only when applied to the sealing faces. Gaskets have to change shape when fitted to the sealing faces. This ensures microscopic matching of the seal surface to the sealing faces, and closes the pores in the sealing material.

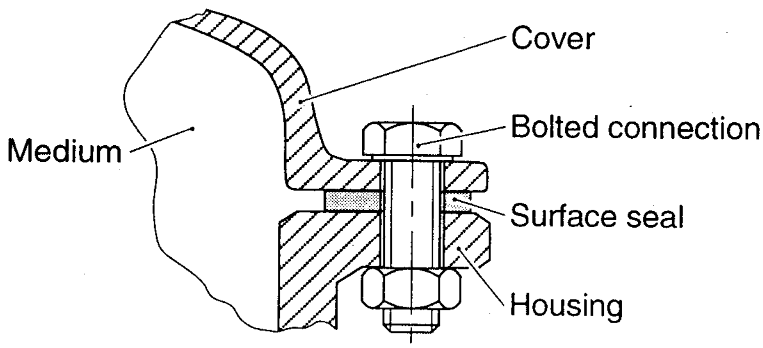


Figure 11.13. Surface sealed joint

Usual materials are:

- ❍ *gaskets* (soft seals):
  - cellulose fibre seals (e.g. paper seals, sealing board),
  - fibre reinforced seals (e.g. aramide fibres),
- ❍ *jointing compounds* (fluid seals):
  - chemical setting (e.g. anaerobic jointing compounds, silicones),
  - non-setting (e.g. glycol compounds).

High demands are made on surface sealed joints of light-weight housings. There are considerable stresses on sealed joints in vehicle transmissions because of the high power density. As well as providing a seal, the surface sealed joint must provide a suitable mechanical bond between the parts of the housing so that force and torque can be transmitted. Together with the housing, the sealed joint must be made as stiff as necessary. There must be no excessive deformation in operation. Deformation generates microscopic relative movements in the sealing faces, which impair the function of the gearbox if they are excessive.

The sealed joint must be capable of withstanding the mechanical stresses arising throughout the life of the gearbox. There must be no breakdown due to loss of pretension in the bolted connections (settling in service), or break-up of the sealing material, or surface wear to the sealing flange [11.31, 11.32].

Guidelines for Using

1/ Gaskets:

- ❍ keep within the recommended contact pressure range, which is normally approximately 2 to 50 N/mm^2. Differential pressure across the seal usually approximately 1 bar (approximately 40 bar in the case of automatic control housings),

- ensure adequate contact pressure,
- ensure contact pressure between the stressed parts is as even as possible,
- select the gasket to suit the macroscopic characteristics of the flange (e.g. undulations). The surface coating of the gasket adapts itself to the microscopic characteristics of the flange (flange roughness),
- create robust sealed joint. Avoid excessive thermal and dynamic stressing, to prevent the seal settling (settling in service).

2/ Jointing Compounds:

- check surfaces before fitting (ensure the surfaces to be joined are clean),
- ensure gaps are bridged as far as possible,
- provide lifting screws for removal,
- allow time for de-aeration when fitting and for setting after fitting,
- there must be a minimum contact pressure in the flange joint between the bolts to prevent permanently plastic (non-setting) sealing material being flushed out.

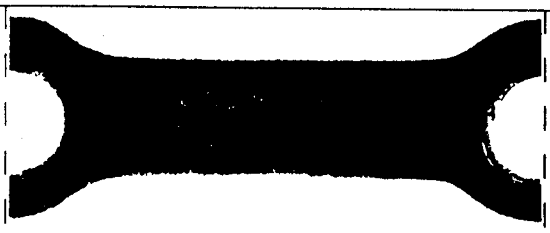
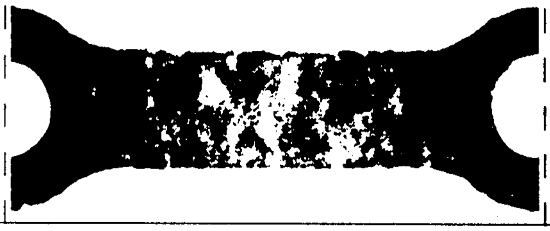
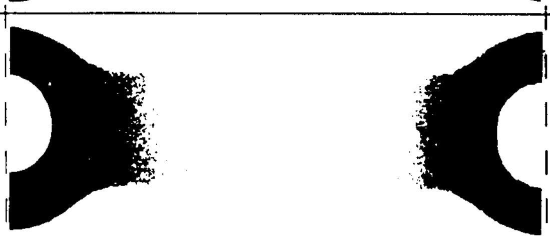
Contact imprint	Characteristic feature
	Very **uniform imprint** with high values in the middle between the bolts. "Very good".
	Marked **inhomogeneities** caused by the structure of the seal material.
	High contact pressure in the **peripheral zones of the seal**, caused by deformations during production of the seal (purching, cutting).
	**Uneven distribution**, no contact pressure in centre between the bolts.
	Effect of the **sheet steel carrier**.

Figure 11.14. Characteristic pressure patterns recorded using Fuji Film Prescale pressure-sensitive film (low pressure range)

Determining the Contact Pressure in the Sealing Gap

It is very important for the designer to know the distribution of contact stress in the sealing gap of a housing seal, since it is an important indicator of the quality of design and the selection of sealing material. There must be a minimum contact stress at each point on the sealing face, to achieve a seal. Contact stress should ideally be as evenly distributed as possible.

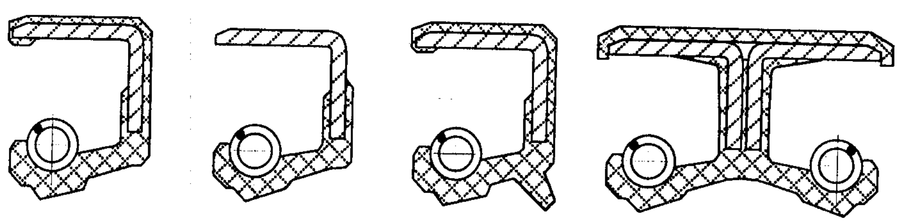
One suitable method of determining the distribution of contact stress in the sealing gap of the test housing is to use the pressure-sensitive Fuji Film Prescale film. Figure 11.14 shows some typical segments of the pressure patterns of a housing. The following characteristic features emerge:

- very *even pressure pattern* with high contact stresses evident in the middle between the bolts,
- marked *inhomogeneities,* due to the structure of the sealing material,
- increased contact pressure in the *seal periphery,* caused by deformation during production of the seal (punching, cutting),
- *uneven distribution of contact stress*; no contact stress in the middle between the bolts,
- effect of *sheet steel carrier* (structure of the sheet steel carrier is visible).

Beside the qualitative appearance described above, the contact stress in the middle between the bolts is the decisive criterion when assessing the pressure pattern, and thus the tightness of the joint.

11.4.2 Seals for Rotating Components

Elastomer *rotary shaft seals* are used in vehicle transmissions wherever rotating shaft through holes are subject to unpressurised fluid or have to be sealed against lubrication oil spray [11.33]. The basic types of rotary shaft seal are shown in Figure 11.15. Figure 11.16 shows the main dimensions and terms. Rotary shaft seals provide a seal in one direction only (oil side).



a) b) c) d)

Figure 11.15. Rotary shaft seals, basic types: **a)** Rubber elastic external shell; **b)** Metallic housing; **c)** With additional dust lip; **d)** Double seal

Guidelines for Using Rotary Shaft Seals

The damage caused by leaking rotary shaft seals is many times greater than the price of a new seal. Guidelines for installing, operating, testing and configuring seals are comprehensively set out in German standard DIN 3760 and 3761, to eliminate seal failures as far as possible.

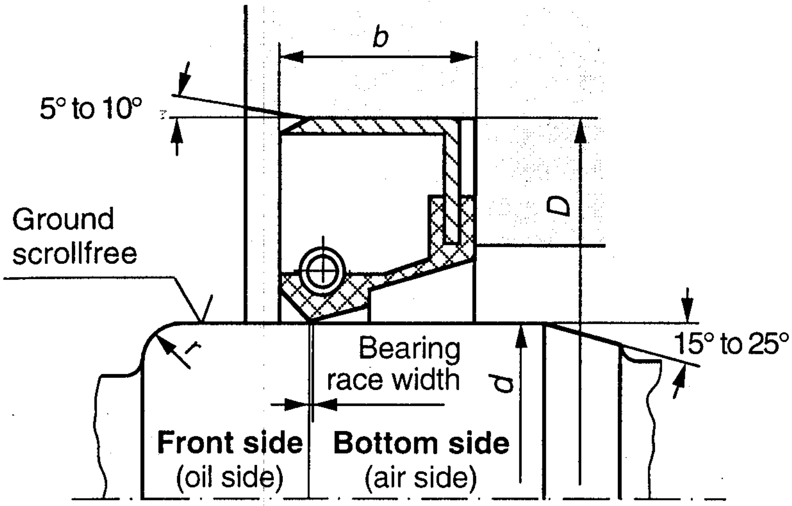


Figure 11.16. Dimensions and terms for rotary shaft seals

- Bearing surface of the shaft, diameter d:
  - tolerance zone h11,
  - surface roughness R_{max} = 1 to 4 µm; the surface must not be too smooth, to ensure the operational wear necessary for proper functioning,
  - surface free of scrolling, i.e. grind or polish without feed, trim grinding wheel scrollfree beforehand,
  - surface hardness: ≥ 45 HRC at a circumferential speed of $v_u \leq 4$ m/s, ≥ 55 HRC at a circumferential speed of $v_u > 4$ m/s,
  - depth of hardness at least 0.3 mm,
  - the grey film must be polished after nitriding,
  - soft bushes made of stainless have however also proved effective in practice.
- Locating bore, diameter D:
  - tolerance zone H8,
  - surface roughness $R_{max} \leq 16$ µm.

Piston rings are used to seal rotating shafts in internal housing parts (automatic transmission: sealing off the servo fluid). The materials used are cast iron and polytetrafluorethylene (PTFE); the coefficient of friction of PTFE is approximately 50% less than that of cast iron.

11.4.3 Seals for Reciprocating Round Components

Seals for reciprocating round components are required at the selector bar through hole for example, and in the case of automatic transmissions, at the pistons of hydraulic clutches and brakes. The seals used are *O-rings*, *plain compression rings* and *groove rings* (Figure 11.17).

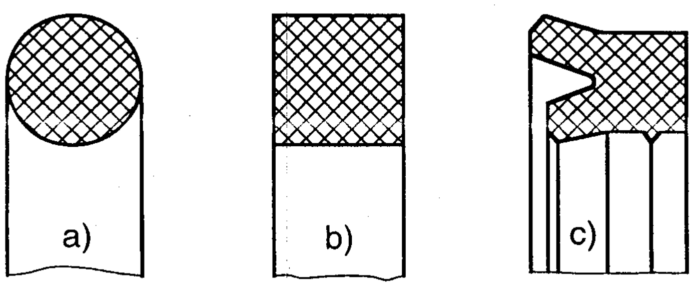


Figure 11.17. Types of seal:
a) O-ring
b) Plain compression ring
c) Groove ring

Guidelines for Using:

1/ O-Rings and Plain Compression Rings

O-rings and plain compression rings are used where running speeds are low, and a little leakage (necessary for lubrication and cooling) is acceptable. Plain compression rings have the advantage over O-rings that they do not twist. The following points should be noted:

- adequate lubrication is essential (no dry running),
- no long periods out of service, since the ring can "stick" to the sealing face,
- select materials with Shore A hardness of 70° to 80°,
- provide thrust rings where there is high pressure, to prevent gap extrusion,
- provide positioning bevels and rounded edges to facilitate fitting,
- permissible initial compression 15%,
- surface roughness must comply with manufacturers specification,
- use plain compression rings where there is a danger of twisting.

2/ Groove Rings

O-rings and plain compression rings can only be used where radial and tilting movements are small. If the design requires considerable freedom of movement between piston and cylinder, groove rings are used. They seal in one direction only, and are available with a great variety of profiles. Further details are given in the manufacturers catalogues.

11.4.4 Practical Examples

The designs discussed in Chapter 12 offer a large number of practical examples of seals. Figure 11.18 shows the seal of a gearbox output shaft. The rotary shaft seal *1* prevents oil spray escaping between the rotating output flange *2* and the housing cover *3*, retaining some of the oil by means of the fluid seal *4* between anti-friction bearing and housing.

To prevent the rotary shaft seal being directly splashed with dirt and water from outside, an oil splasher *5* is mounted on the output flange. The profile of the gearbox output splined shaft is sealed with an O-ring *6*.

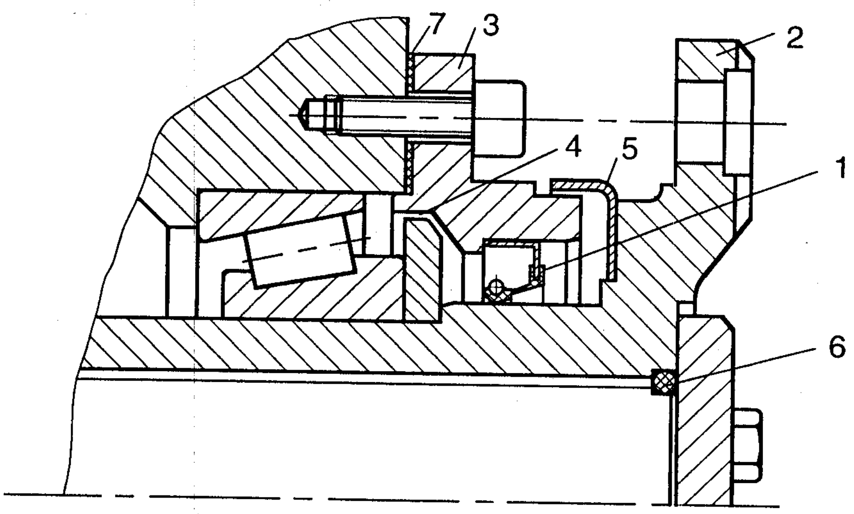


Figure 11.18. Sealing a gearbox output shaft. *1* Rotary shaft seal ; *2* Output flange; *3* Housing cover; *4* Fluid seal; *5* Oil splasher; *6* O-ring; *7* Gasket

Figure 11.19 is a sectional view of a multi-disc clutch of a passenger car automatic transmission. The seal between the rotating clutch cage *4* and the housing *5* or between the rotating shaft *6* and the housing *5* is formed by piston rings *1*, mostly butt jointed. The rings are tensioned outwards, and run in the groove of the respective inner part. The piston *7* moves linearly, and is sealed by the groove rings *2*.

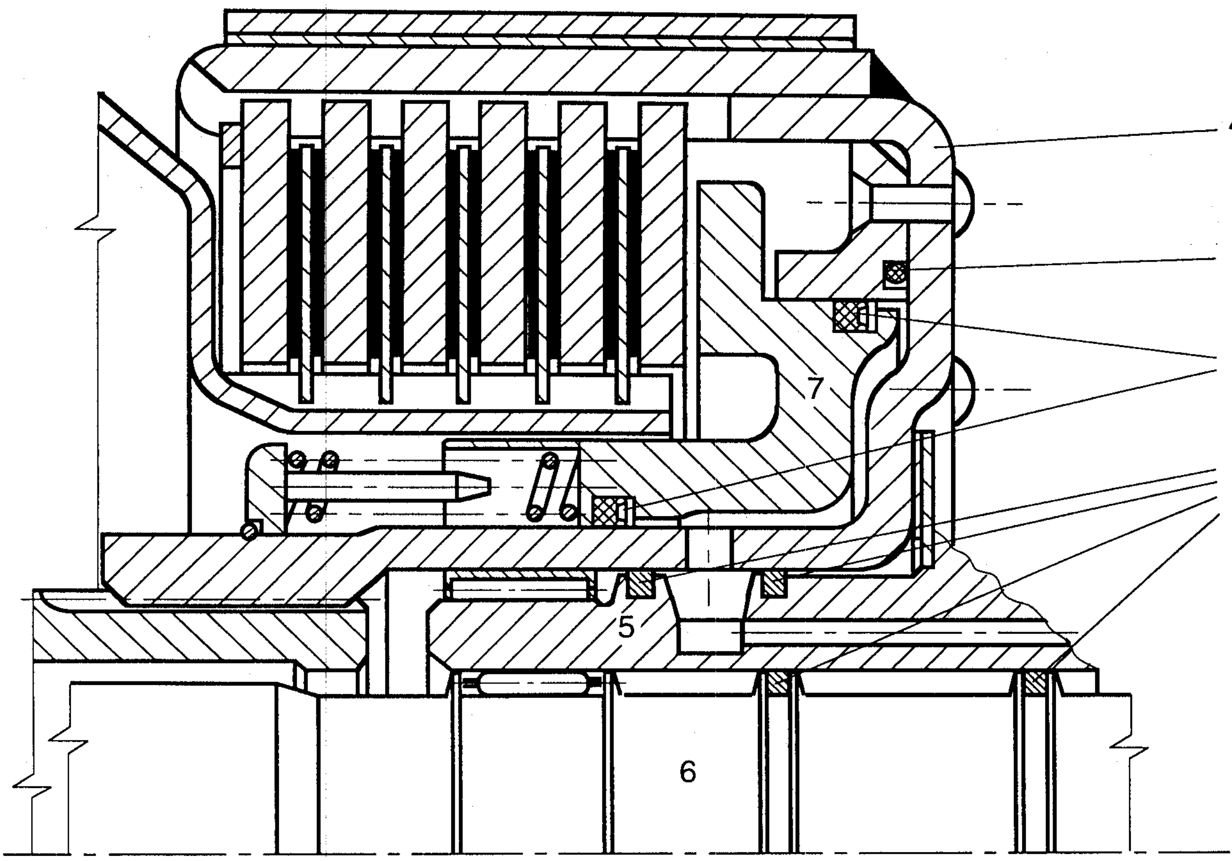


Figure 11.19. Sealing a multi-disc clutch. *1* Piston ring (rotating); *2* Groove ring (linear); *3* O-ring (static); *4* Clutch cage; *5* Housing; *6* Shaft; *7* Piston

11.5 Vehicle Continuous Service Brakes

You can drive faster with good brakes

Engine power has increased significantly relative to overall vehicle weight in recent years. Vehicles are thus capable of travelling at higher average speeds. Maintaining high speeds even when travelling downhill, requires steady-state braking, sometimes with a high energy content, especially in the case of vehicles with a high laden weight (see also Section 5.1.2 "Engine Braking Force").

Heavy commercial vehicles normally have compressed air operated service brakes. These are designed to be capable of safe deceleration braking. They are not designed for continuous operation. Friction service brakes can be subject to thermal overload on long downhill runs with permanent steady-state braking, which leads to impairment of the braking effect. This effect is called "fading".

Equation 5.7 demonstrates that a commercial vehicle with a laden weight of 40 t needs some 360 kW of braking power to negotiate a downhill gradient of 7% at a constant speed of 60 km/h. This braking power must be dissipated in the form of heat. The service brake can effectively carry out this function only for brief periods, because of poor heat dissipation. By way of comparison, the motive power rating of such a commercial vehicle would also be some 360 kW.

Continuous service brake systems can increase the economic efficiency of commercial vehicles by enabling higher average speeds, especially on long downhill runs, and by reducing brake wear. But above all they contribute to improving active safety by reducing the load on the service brake.

There are two types of continuous service brakes in general use, namely engine brakes and retarders, which in turn have different basic concepts (Figure 11.20).

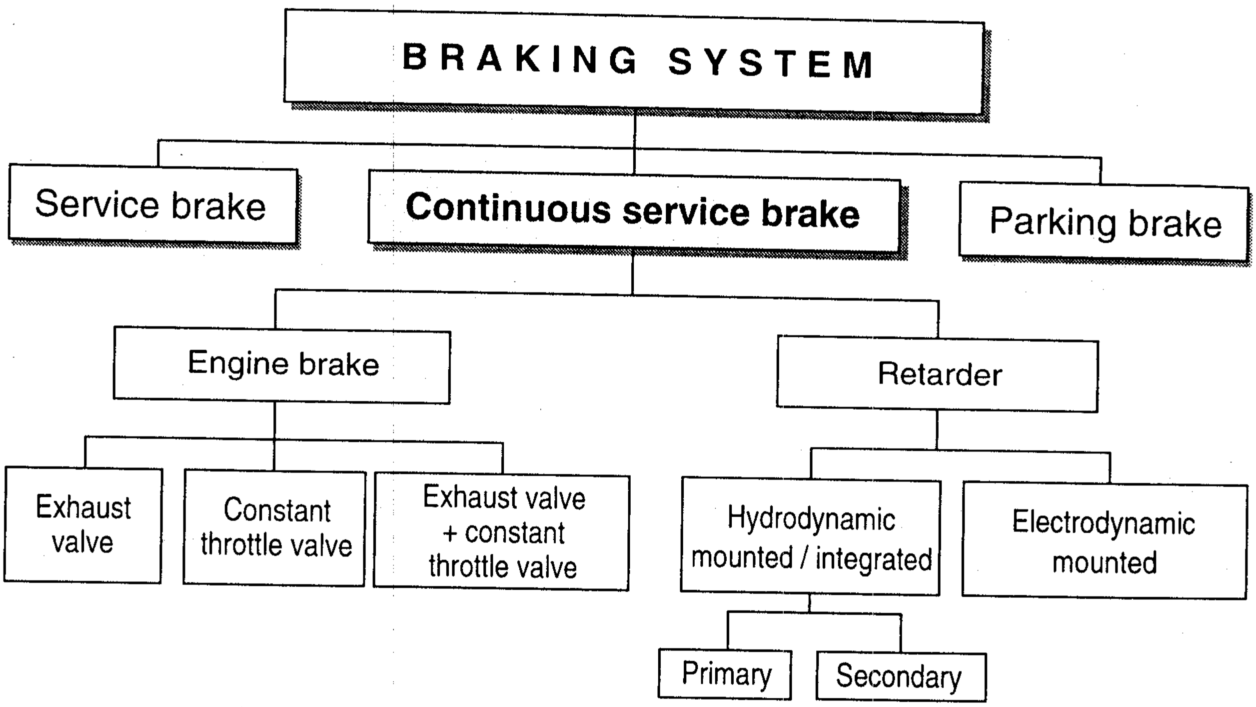


Figure 11.20. Overview of braking systems

11.5.1 Definitions

A *continuous service brake* is a supplementary braking system capable of producing and maintaining braking force over a long period without any noticeable decline in performance. The continuous service brake must therefore function reliably regardless of the condition and effectiveness of the other braking system. The requirements a motor vehicle braking system has to fulfil in the EU are set out in the ECE regulation 13 and in EU directives [11.34]. Whereas the service brake acts on all wheels, the continuous service brake brakes only on the vehicle's connected wheels (drive wheels). There are also continuous service brake systems for trailers and semi-trailers.

The German Motor Vehicle and Use Regulations (Section 41) requires continuous service brakes to be fitted to all commercial vehicles and trailers with a gross weight rating of more than 9 t, and to all buses of more than 5.5 t. The continuous service brake fitted must enable the vehicle to sustain a steady-state speed of 30 km/h on a 7% downhill gradient for a distance of 6 km.

11.5.2 Engine Braking Systems

Combustion engines generate braking torque when coasting (Figure 3.12). Engine braking torque arises principally from pumping work. The engine braking force F_B depends on the gear (Figure 5.6). Additional engine brakes are mounted directly in the exhaust section or in the cylinder head.

In the case of engine brakes with exhaust valve, a butterfly valve installed in the exhaust section closes, creating dynamic pressure in the exhaust system (Figure 11.21). The dynamic pressure reduces the engine speed by inhibiting the charge cycle, thus braking the vehicle. Exhaust valves are not used in coaches because of the valve noise. The exhaust valve is also known as "exhaust throttle valve" or simply "throttle valve".

The constant throttle valve is an additional exhaust valve in the cylinder head of the engine. Opening the constant throttle valve when braking dissipates the unwanted expansion energy during the second cycle in the combustion chamber, amplifying the engine braking effect.

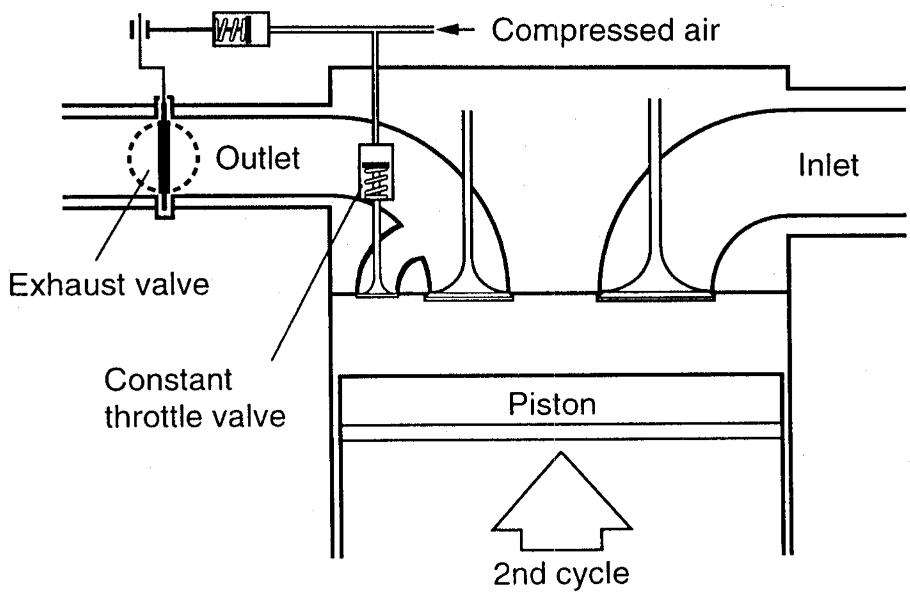


Figure 11.21. Engine brake with exhaust valve and constant throttle valve

The braking effect of the constant throttle valve is much better than that of the exhaust valve in the lower engine speed and velocity range, which in turn provides better braking performance in the upper speed range. Figure 11.21 shows a combined solution comprising both exhaust valve and constant throttle valve [11.35].

11.5.3 Retarders

Retarders are virtually non-wearing continuous service brakes. They are capable of converting kinetic energy arising over long periods into thermal energy, which they store and dissipate without overheating. They are used in commercial vehicles and buses. Retarders used in practice vary principally in the way they convert energy. In the case of hydrodynamic retarders, the energy is converted by fluid friction, whereas electrodynamic retarders use a magnetic field.

Hydrodynamic Retarders

In hydrodynamic retarders the hydraulic energy of a fluid is used to brake the vehicle. The physical principle of operation corresponds to that of a hydrodynamic clutch with a fixed turbine (see Chapter 10 "Hydrodynamic Clutches and Torque Converters"). The rotor R (impeller) is located in the power flow. The stator S is fixed to the retarder housing (Figure 11.22).

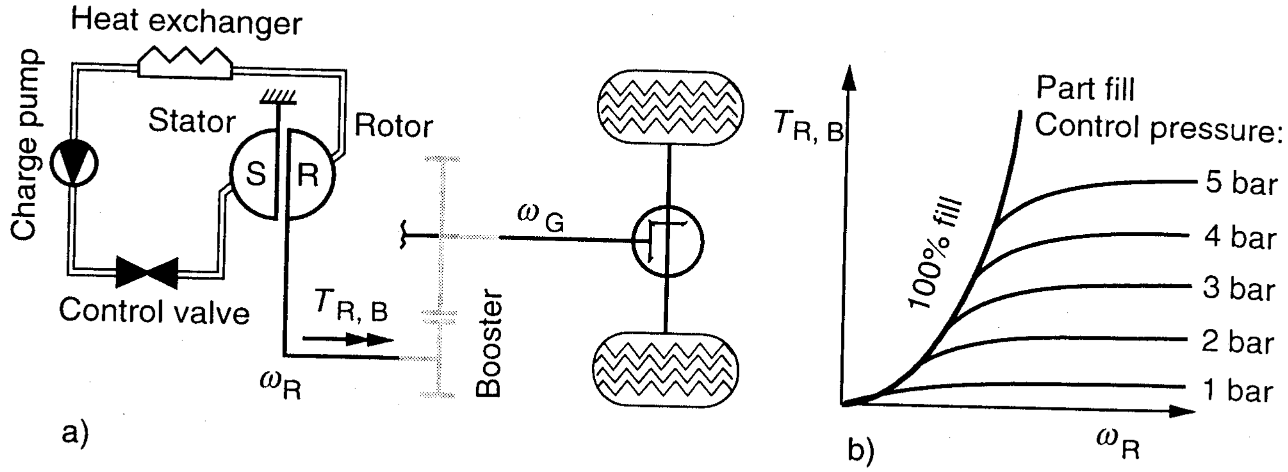


Figure 11.22. Structure of a hydrodynamic retarder. **a)** Secondary retarder with optional booster; **b)** Rotor brake torque $T_{R,B}$ related to control pressure

When the hydrodynamic retarder is activated, a quantity of oil proportionate to the brake position is fed into the blade chamber. The braking torque is controlled by the retarder fill level. The rotating rotor carries the oil that reacts on the stator, thus producing a braking effect on the rotor shaft. Hydrodynamic retarders can produce no braking torque when the vehicle is at rest, and very little when the rotor is rotating slowly. Hydrodynamic retarders are therefore not suitable as service brakes.

The relationship of retarder torque or rotor brake torque $T_{R,B}$ to the rotor speed is given by the theory of hydrodynamic machines (Equation 10.14), thus:

$$T_{R,B} = \lambda\, \rho\, \omega_R^2\, D^5 \,. \tag{11.13}$$

The performance figure λ is a function of the speed ratios ν (Equation 4.2), $\nu = n_S / n_R$. In the retarder the stator S stands still, making the speed ratio ν of stator to rotor zero. This corresponds to the stall point of a hydrodynamic clutch.

The performance figure λ is a function of the rotor speed, the blade position and the fill level. The fill level of the retarder, and thus the rotor braking torque, is a function of the control pressure. The oil seeking to escape because of the centrifugal force in the retarder is retained in the retarder by means of the control pressure (also known as dynamic pressure), depending on the braking torque required. This gives a brake torque range of adjustment $T_{R,B}$ (Figure 11.22b).

The symbol for oil density is ρ, and the symbol for retarder diameter is D. The angular velocity of the rotor shaft ω_R is either equal to the angular velocity of the gearbox input shaft, or equal to the angular velocity of the propeller shaft ω_G (Figure 11.23), depending on where the hydrodynamic retarder is installed.

Primary retarders are located on the engine side, and secondary retarders on the gearbox side in the power train (Figure 11.23). There is thus gear-dependent braking torque at the wheels for the primary retarder, which increases in proportion to the transmission ratio as the gear becomes lower. Primary retarders are therefore effective even at low road speeds.

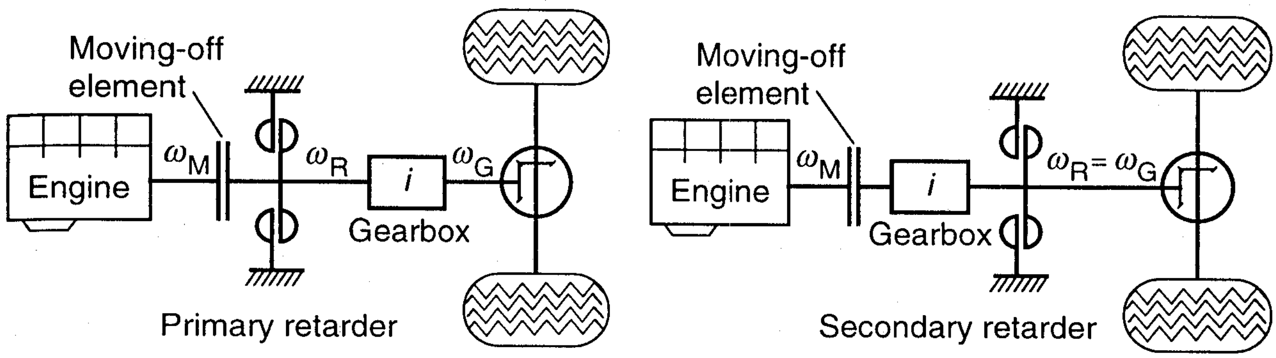


Figure 11.23. Alternative positions for mounting hydrodynamic retarders

They generate relatively high braking torque levels at the drive wheels, which fall substantially as the road speed increases (Figure 11.24). The effectiveness of this system is impaired by the interruption of the power flow and thus the braking effect when changing gear in manual gearboxes. In commercial vehicle transmissions with torque converters, primary retarders are often integrated into the transmission. Please refer in this connection to the production examples in Figure 12.10 "16-speed semi-automatic manual commercial vehicle gearbox", and Figure 12.14 "6-speed automatic gearbox".

The braking torque of the secondary retarder is only dependent on its characteristic. The secondary retarder is advantageous for dimensioning the transmission, since its braking torque is not an additional load on the transmission.

A high level of braking torque is virtually constantly available over a wide velocity range (Figure 11.24). This design is particularly suitable for commercial vehicles with a high laden weight and high speed. A booster step can also be fitted before the rotor shaft (Figure 11.22a). The booster is one way of increasing the braking torque of secondary retarders at low road speeds, and fitting twin rotors is another. The secondary retarder with booster is flange mounted to the side of the gearbox at the rear, and driven by a spur gear stage – a ratio of 1 : 2 is usual.

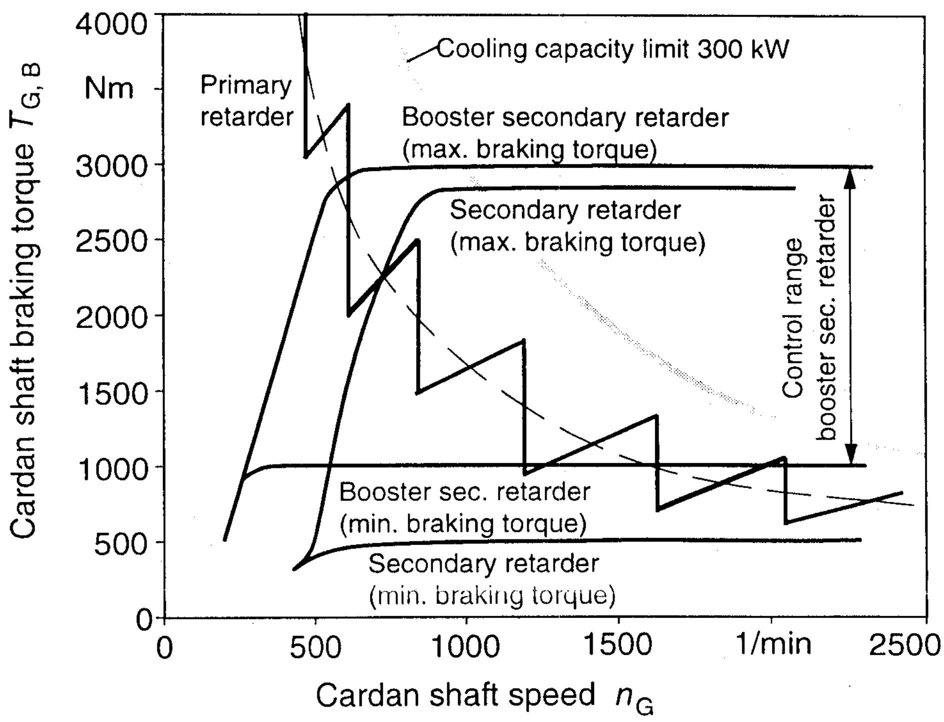


Figure 11.24. Braking torque levels of hydrodynamic retarders as a function of the propeller shaft speed, i.e. the road speed

The achievable braking power is limited primarily by the cooling circuit rather than by the continuous brake itself. The cooling capacity can be increased by fitting an auxiliary oil/air heat exchanger (Figure 11.22a). It is possible to use a dedicated oil circuit for the retarder, or to share the transmission oil circuit. Cooling capacity is currently limited to around 300 kW.

With a shared oil circuit, the additional heat exchanger of the retarder can be used to cool the transmission when the retarder is switched off. This increases the service life of toothing and bearings, and slows down oil ageing. When the retarder is switched on, the oil circuit is separated from the transmission. The transmission oil temperature is thus not affected during braking.

If there is a retarder in the power train, then there will be power loss under power when the retarder is not full, due to air re-circulation (fan losses). The losses of the unfilled retarder can be minimised reducing the air circulation by means of annular slide valves and restrictor slide valves pivoting between rotor and stator [11.36] (see also Figure 12.14 and associated discussion).

Electrodynamic Retarders

Electrodynamic retarders – often also referred to as eddy current retarders – are of much simpler design than hydrodynamic retarders. They are generally designed as mounted secondary retarders. The braking effect is based on the physical principle of the dynamic effect in electromagnetic fields. The stator, in the form of a disc, is fitted with several field coils, and fixed to the gearbox housing. On the transmission and rear axle side there are rotors linked to the propeller shaft (Figure 11.25). The rotors are air-cooled, and have fins to facilitate heat dissipation to the environment.

The field coils are fed with current from the battery or the generator during braking. Eddy currents are induced in the rotors when they pass through the magnetic field, which impede the rotation of the rotors. The level of braking torque depends on the excitation of the stator coils and of the air gap between the rotor and the stator. The braking power can be activated in several stages by passing current through the field coils in pairs.

The features of electrodynamic retarders are their simplicity of construction, and their rapid response; on the other hand they are relatively heavy and can require an adequate electrical power supply to function properly. In contrast to hydrodynamic retarders, they have relatively high braking torque even at low rotor speeds. They can thus be used until the vehicle is at rest, and are therefore particularly common in city buses.

But as the rotor speed increases, the temperature of the electrodynamic retarder increases too. The number of active coils, or the power supply to the coils, is accordingly limited in order to protect against thermal damage.

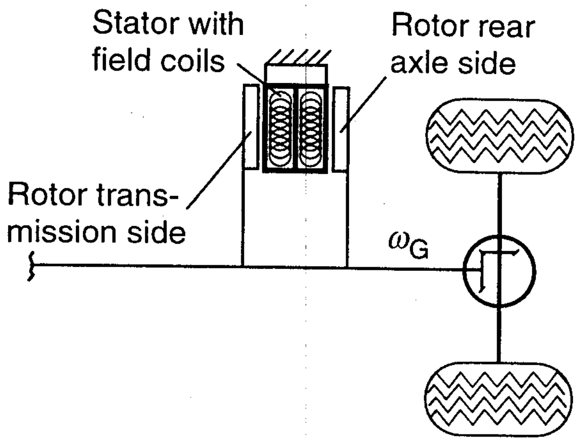


Figure 11.25. Basic structure of an electrodynamic retarder (eddy current retarder)

11.5.4 Actuation and Use

The engine brake (exhaust valve, constant throttle valve) is usually actuated by means of a foot switch. Retarders are actuated either with the brake pedal or with a hand lever, enabling various braking steps can be selected.

New design concepts are seeking to make the controls easier to operate, with braking systems actuated just by the brake pedal, and cutting in as required using smart logic. When the brake pedal is operated, first the retarder is engaged. If the braking force is not sufficient, the engine brake is automatically activated, providing additional braking power. The service brake only cuts in finally if needed.

Electronic engine brake and retarder control combined with automatic speed control can keep road speed constant on downhill gradients. The retarder is then combined with the antilock braking system, to prevent the wheels locking. The driver only needs to operate the brake pedal until the vehicle has slowed down to the desired speed; the speed set is then maintained. Only when the accelerator pedal is pressed does the braking system switch off (see Section 13.4 "Electronically Controlled Braking and Traction Systems").

Whereas primary retarders and engine brakes have advantages in the lower speed range and on steep downhill stretches, secondary retarders are suitable for higher speeds. Electrodynamic retarders perform consistently throughout the speed range. There is normally a hydrodynamic primary retarder integrated in commercial vehicle automatic transmissions, since various essential peripherals such as the charge pump have to be fitted anyway.

Both primary and secondary retarders brake only through the drive wheels. If the drive wheels are only partly loaded, or the coefficient of friction of the wheels to the road surface is reduced, the drive wheels can lock. Retarders should therefore preferably be used in combination with antilock braking systems.

12 Typical Designs of Vehicle Transmissions

The process of human thought proceeds from the concrete to the abstract, then back to the concrete /J. DEWEY, 1910/

This chapter examines some particular transmission designs, and considers their structural design. You may refer to the gearbox diagrams in Chapter 6 with regard to the gear-wheel configurations in the transmissions examined. This applies particularly to multi-range transmissions.

Sections 12.1 to 12.4 are devoted to manual passenger car and commercial vehicle gearboxes. Sections 12.5 to 12.7 cover final drives, differential gears, transfer boxes and all-wheel power trains. Table 12.1 gives an overview of the selector boxes discussed. Each gearbox is allocated a serial number. At the beginning of Sections 12.5 and 12.6 there are summaries of the designs covered in the sections.

Table 12.1. Vehicle transmissions examined in Sections 12.1–12.4. *TCC* Torque converter clutch; *CC* Torque converter lock-up clutch

Serial No.	Vehicle	Gears	Characteristic	Manufacturer	Designation	Diagram Fig. No.	Design Fig. No.
1	Passenger car	5	Two-stage manual shift	ZF	S 5-31	6.19b	12.1–3
2	Passenger car	6	Two-stage manual shift	Getrag	6-speed	6.20a	12.4
3	Passenger car	5	Single-stage manual shift	VW	MQ	6.18b	12.5
4	Passenger car	6	Single-stage manual shift	Opel	F28-6	6.20b	12.6
5	Commercial vehicle	6	Single-range transmission	ZF	S 6-66	6.36b	12.7
6	Commercial vehicle	16	Three-range transmission	ZF	16 S 109	6.44	12.8
7	Commercial vehicle	13	Two-range, two countershaft	Eaton	Twin Splitter	6.45	12.9
8	Commercial vehicle	16	TCC, with CC, with retarder	ZF	Transmatic	6.47	12.10 12.11
9	Passenger car	5	Conv. automatic, without CC	MB	W5A 030	–	12.12
10	Passenger car	∞	Continuously variable transmission	Ford	CTX	6.32	12.13

Serial No.	Vehicle	Gears	Characteristic	Manufacturer	Designation	Diagram Fig. No.	Design Fig. No.
11	Commercial vehicle	6	Conv. automatic with CC	ZF	6 HP 600	6.48	12.14
12	Commercial vehicle	6	Conv. automatic with CC, modulation clutch, transfer box	Renk	HSRM 226.22	–	12.15
13	Passenger car	5	Single-stage manual shift automatic master clutch	MB LuK	SG 150 EKM	–	12.16a
14	Passenger car	5	Countershaft-type automatic with CC	MB	W5A 180	–	12.16b
15	Passenger car	5	Conv. automatic with CC	MB	W5A 580	–	12.16c
16	Passenger car	5	Conv. automatic	ZF	5 HP 18	6.29	12.17
17	Commercial vehicle	16	Three-range, two countershaft, semi-automatic	ZF	16 AS 2600	–	12.18

12.1 Manual Gearboxes

12.1.1 Manual Passenger Car Gearboxes

4-speed manual gearboxes were standard for passenger cars in Europe until the early 1980's. As engine power and vehicle weight increased and c_W ratings improved, larger overall gear ratios became necessary. Large overall gear ratios facilitate moving off, provide good acceleration, and also reduce engine speed and hence fuel consumption at high speeds. Manual gearboxes therefore now usually have five speeds. Sometimes 6-speed gearboxes are now used in "high-performance" passenger cars.

1/ Two-Stage 5-Speed Manual Passenger Car Gearbox; ZF S 5-31

Figure 12.1 shows a two-stage 5-speed passenger car countershaft-type manual gearbox with direct drive in fifth gear. In this design, first and second gear are roughly in the middle of the main shaft. This contravenes the principle whereby gear steps with higher torque conversion should be located as close as possible to a main bearing (Section 8.2 "General Design Guidelines"). But the resultant shaft deflection can be controlled by appropriate gearing geometry. The advantage of this structural design is that the more frequently used gear steps of third and fourth gear are near a bearing point, making them run more quietly.

In contrast to *in-line gearboxes,* where all synchronisers are mounted on the gearbox main shaft, in this gearbox the synchronisers for third and fourth gear are moved to the countershaft. This arrangement means the idler gears and shift gears for third and fourth gear are no longer on the countershaft itself, but linked to the output side.

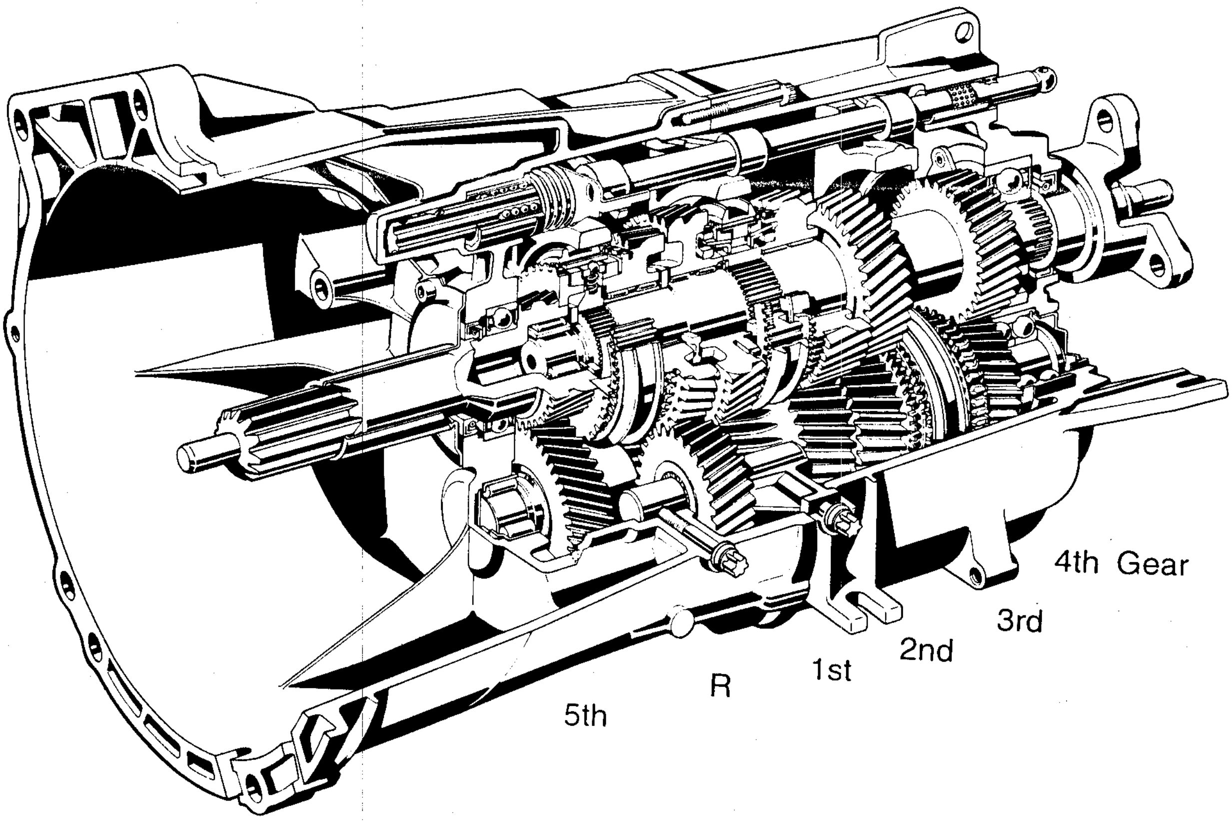


Figure 12.1. 5-speed manual passenger car gearbox ZF S 5-31; *gearbox diagram Figure 6.19b*

Their speeds thus no longer have to be matched to the output speed during synchronisation, which reduces the frictional work and shift effort required to change gear. It is not possible to move any more idler gears to the countershaft, since the differential rotational speeds are too great. The synchronisers for first and second gear are of double-taper design, significantly reducing gearshift effort. All other gear steps, including reverse gear, have single-taper synchronisers.

Since the idler gears in third and fourth do not rotate when the vehicle is stationary in neutral, they do not produce as much rattle as idler gears mounted on the main shaft (see also Section 7.5).

The kinematics of the engagement process is shown in Figure 12.2. Instead of separate selector bars for each individual selector fork, a central selector shaft *1* is used with gear shift forks *5–7*. This has weight advantages, and is more cost-effective. The central selector shaft runs in linear ball bearings *4*, which reduces gear shift effort. The gear shift forks have a pivot *9* supported in the housing, around which they pivot on the lever principle. This enables gear shifting effort to be reduced by selecting a suitable lever ratio. In this type of design the gear shift forks (or gates) are changed by the shaft turning.

The countershaft runs in a cylindrical roller bearing on the input side and in a double-row angular ball bearing on the output side, instead of the more usual tapered roller bearings or single-row deep groove ball bearings, making shifting more precise and the gearbox quieter. No tapered roller bearings were used anywhere in the gearbox, since they entail a number of disadvantages, particularly changing the bearing clearance as a result of temperature-related changes in length in the gearbox, and increased play caused by bedding in of the housing.

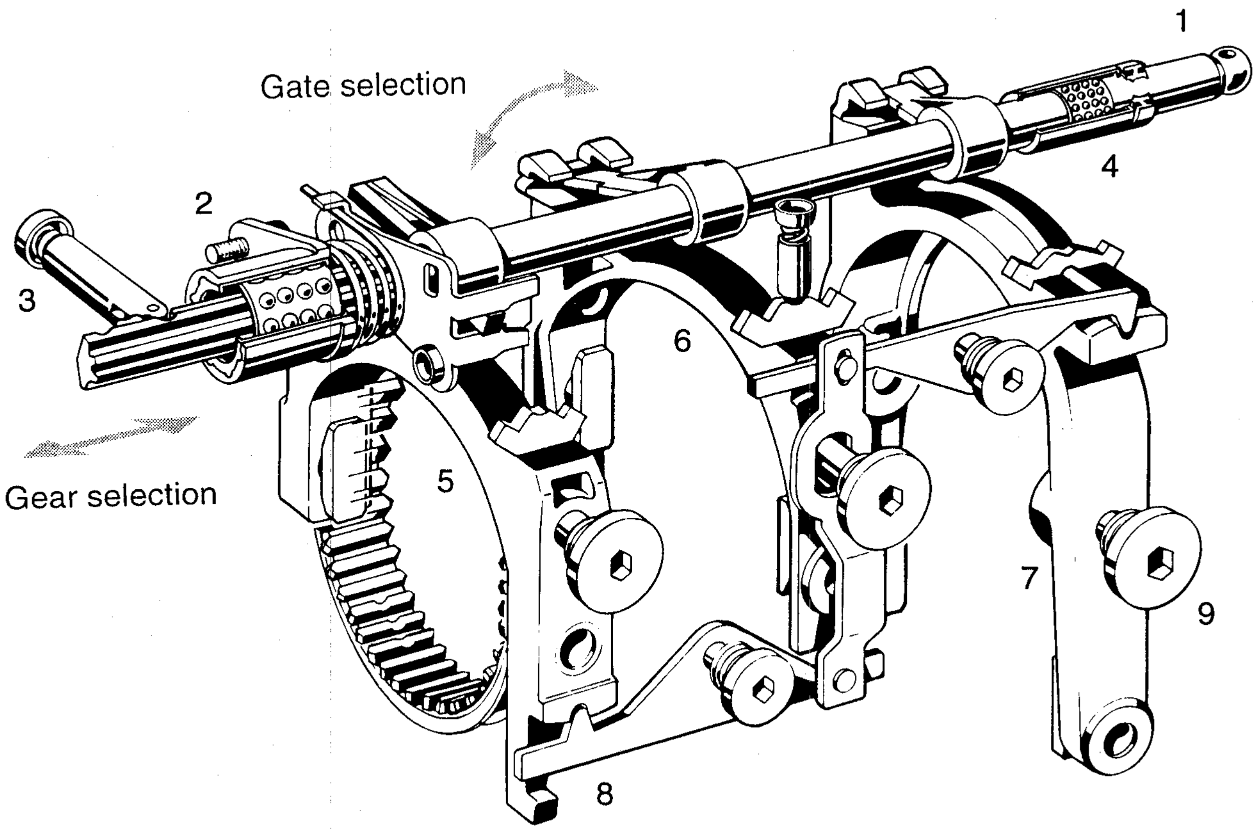


Figure 12.2. Operation of the shifting elements in the gearbox in Figure 12.1.
1 Central selector shaft; *2* Selector fork; *3* Indexing pin; *4* Ball sleeve; *5–7* Gear shift forks; *8* Catch; *9* Pivot

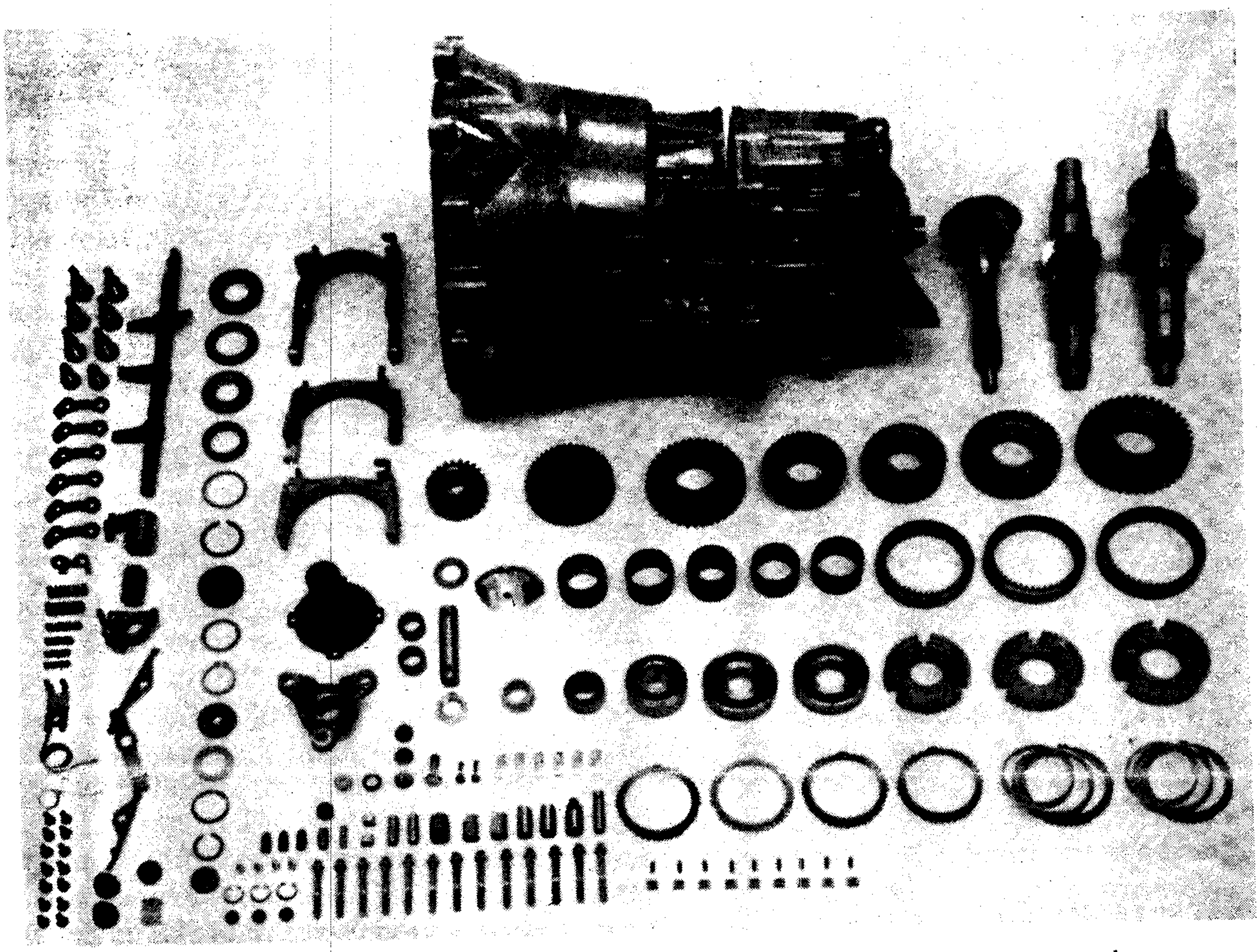


Figure 12.3. The components of the ZF S 5-31 5-speed manual passenger car gearbox

Tapered roller bearings also make it impossible to use "clean bearings" that are sealed against dirt by a sealing collar. Using them makes it possible to extend the oil change intervals.

The gearbox housing itself is of end-loaded design with an integrally cast clutch bell housing (Figure 12.3), which ensures the housing is very rigid [12.1].

2/ Two-Stage 6-Speed Manual Passenger Car Gearbox; Getrag

Changes in automotive engineering technology have lead to the development of manual gearboxes with six speeds. These changes were improved c_W ratings enabling higher top speeds, increased vehicle weight due to more comprehensive equipment, and the desire for improved flexibility [12.1].

An example of 6-speed manual gearboxes for passenger cars is shown in Figure 12.4. This is a two-stage countershaft gearbox with head set mounted on the input side. The transmission shafts are longer than the gearbox shown in Figure 12.1. This is firstly because of the additional gear pair, and secondly because of the larger face widths needed for vehicles with powerful engines. In order to minimise the resultant shaft deflection under load, the gear pairs for first and second gear are located near the output side main bearing.

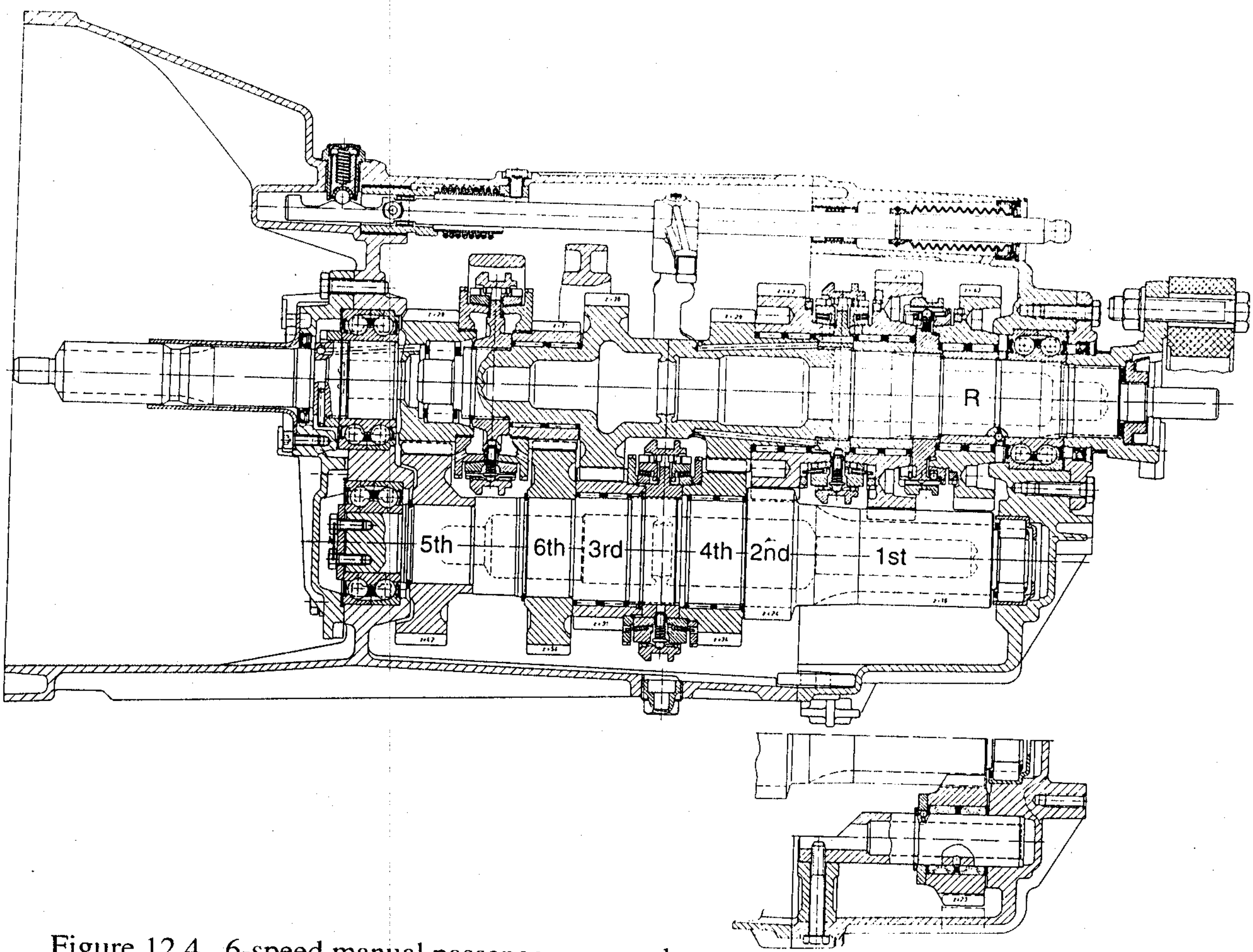


Figure 12.4. 6-speed manual passenger car gearbox, built by Getrag; *gearbox diagram Figure 6.20a*

The gear pairs in third and fourth gear are mounted in the centre of the gearbox. The gear pattern resulting from the configuration of the gear pairs, with a gate each for first and second gear, third and fourth gear, and fifth and sixth gear, represents a logical extension of the familiar gear pattern of 5-speed gearboxes.

To reduce rattling noise, the third and fourth gear idler gears are moved onto the countershaft. To reduce the gear shift effort, the first and second gear synchronisers are three-cone synchronisers, and the third and fourth gear synchronisers are double-taper synchronisers. The fifth and sixth gear and reverse gear synchronisers have a single cone. Figure 12.4 shows the central selector shaft running in ball bearings. This view does not show the four additional selector bars for the respective gates. Shifting between these selector bars is controlled by turning of the central selector shaft, acting through lever mechanisms. The gear selection lock is located on the input side end of the central selector shaft. The torsion spring mounted on the right of the selector shaft defines the initial position for the gate selection rotary movement.

The countershaft runs in a cylindrical roller bearing on the output side, and in a double-row angular ball bearing on the input side, which makes the gearbox run quieter. Bearings sealed to exclude dirt (clean bearings) increase bearing service life, and can enable smaller and therefore lighter bearings to be used.

The main shafts and countershafts of the gearbox are hollow drilled to contour. This is achieved using two-part shafts, which are friction welded after the internal contour has been bored. This confers weight advantages. The gearbox housing itself is of end-loaded design with integrally cast clutch bell housing, and is thus very rigid [12.1].

3/ Single-Stage 5-Speed Manual Passenger Car Gearbox; VW MQ

A large proportion of vehicles are fitted with front-wheel drive and transverse-mounted gearbox. Single-stage countershaft transmissions with integral final drive are always used in this drive configuration.

This format is compact, and reduces the space needed to accommodate the power train, importantly by eliminating the propeller shaft. It has the major advantage over the standard drive format that the complete power train including the engine can be preassembled as a "package" and fitted to the body shell. The 5-speed gearbox in Figure 12.5 is a typical example of the front/transverse format.

The gearwheels of first gear to fifth gear are mounted on the input shaft in sequence, starting from the clutch side. A striking feature is the fifth gear located outside the cast gearbox housing, which is encased against the environment by a separate sheet steel pan. The reason for this feature is that this gearbox is developed from a 4-speed gearbox, with the fifth gear added on. The distance between bearings is kept small by the resultant location of a main bearing between fourth and fifth gear. This has a beneficial effect on shaft deflection under load, although the fifth gear is overhung.

The tapered roller bearings of the input shaft, as also that of the output shaft (= countershaft), have a collar at the bearing outer ring to counteract the axial forces against the housing. These special bearings allow the wall of the housing to be thinner, since no collars are required on the housing, and no grooves for circlips.

The toothing of the reverse gear is located on the first and second gear sliding sleeve. All forward gears have a single-taper synchroniser. The synchronisers are activated by gear shift forks, whose plain bearings are clearly visible over the input shaft. The gear shift forks are in turn activated by means of a central selector shaft (mounted vertically in the middle of the gearbox).

The gearbox housing is open to both sides and has an integrally cast half of the axle gearbox housing. The clutch bell housing, which carries the other half of the axle gearbox, is bolted to the gearbox. The other side of the gearbox housing is closed off with a metal cover, as already mentioned. The bevel gear differential is lubricated by the gearbox oil circulation. The worm gear of the speedometer drive can be seen at the drive cage of the differential. The clutch throwout bearing with its operating lever is located in the clutch bell housing, sitting on the gearbox input shaft. It is designed as a pressed steel part.

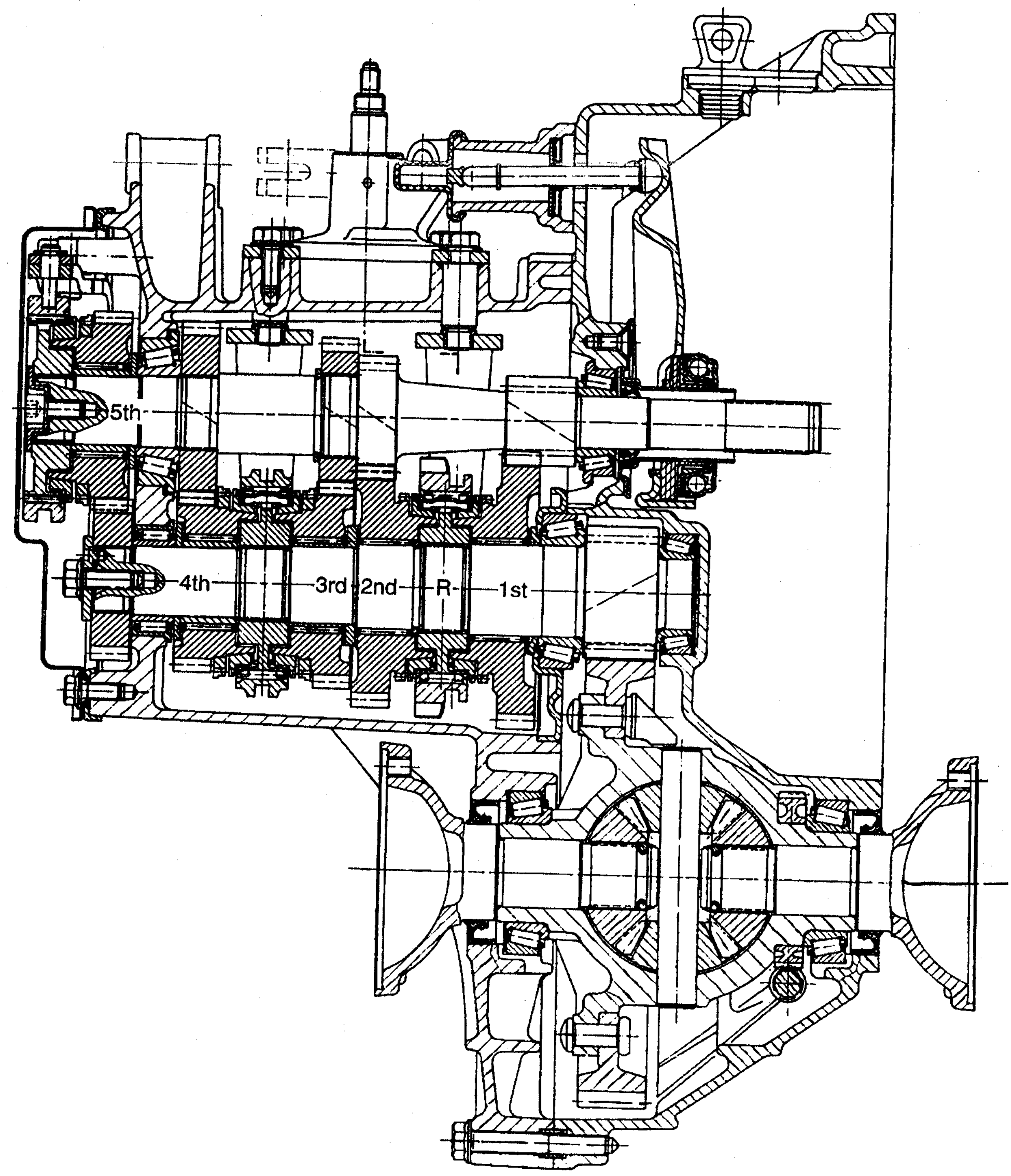


Figure 12.5. VW MQ 5-speed manual passenger car gearbox;
gearbox diagram Figure 6.18b

4/ Single-Stage 6-Speed Manual Passenger Car Gearbox; Opel F28-6

More and more powerful models of passenger cars with front-wheel drive are being produced. This has led to the development of 6-speed gearboxes for single-stage passenger car gearboxes too [12.2].

The gearbox in Figure 12.6 is an example of this type. The gearwheels of the individual gear steps are mounted on the input shaft as follows, starting from the clutch side.

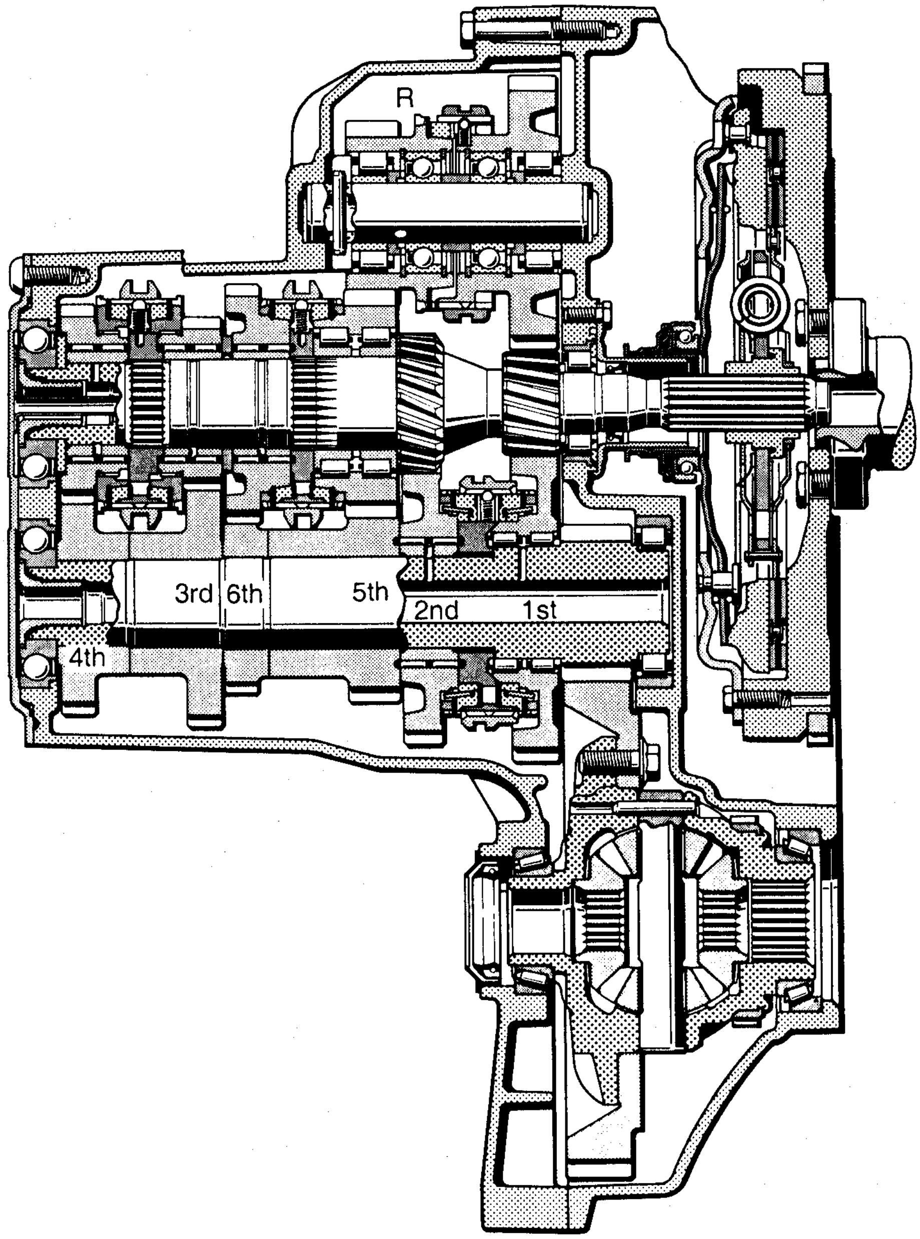


Figure 12.6. Opel F28-6 6-speed manual passenger car gearbox; *gearbox diagram Figure 6.20b*

The gearwheels of first and second gear are located at the bearing, and their idler gears and synchronisers are located on the countershaft. Then come gearwheels of fifth and sixth gear, and finally those of third and fourth gear. Their idler gears are mounted on the input shaft. The design solution adopted for reverse gear is of interest, which dispenses with additional gearing on the input shaft. It is shifted on its own countershaft. This is not located in one plane with the other shafts, as shown in the Figure, but is spatially displaced, like the axle configuration in Figure 6.21. The power flow in reverse gear is from the fixed gear of first gear to its idler gear, from there to the countershaft of the reverse gear, and from there on to the fixed gear of fifth gear via its idler gear. This design makes it possible for the overall length of the gearbox to be very short. In this case, it was actually possible to reduce the width compared to its predecessor with five gears.

All gears are synchronised. First and second gear are fitted with a double-taper synchroniser. The operating elements are not shown in the figure; they have anti-friction bearings at all bearing points. The structure of the gearbox housing is similar to the gearbox presented in Figure 12.5, so the comments in that case apply here too.

12.1.2 Manual Commercial Vehicle Gearboxes

As discussed in Chapter 6, most commercial vehicle gearboxes with more than six speeds are of multi-range design. If the gearbox is so designed that the range units are only flange-mounted at defined interfaces, it is possible to produce a gearbox kit with a uniform basic gearbox.

5/ Two-Stage 6-Speed Manual Commercial Vehicle Gearbox; ZF S 6-66

The 6-speed manual gearbox (Figure 12.7) for gearbox input torques of up to 660 Nm for mid-range trucks (110–210 kW), becomes a 12-speed gearbox by flange-mounting the splitter unit that has two constant ratios, and "compressing" the gear sequence of the gearbox.

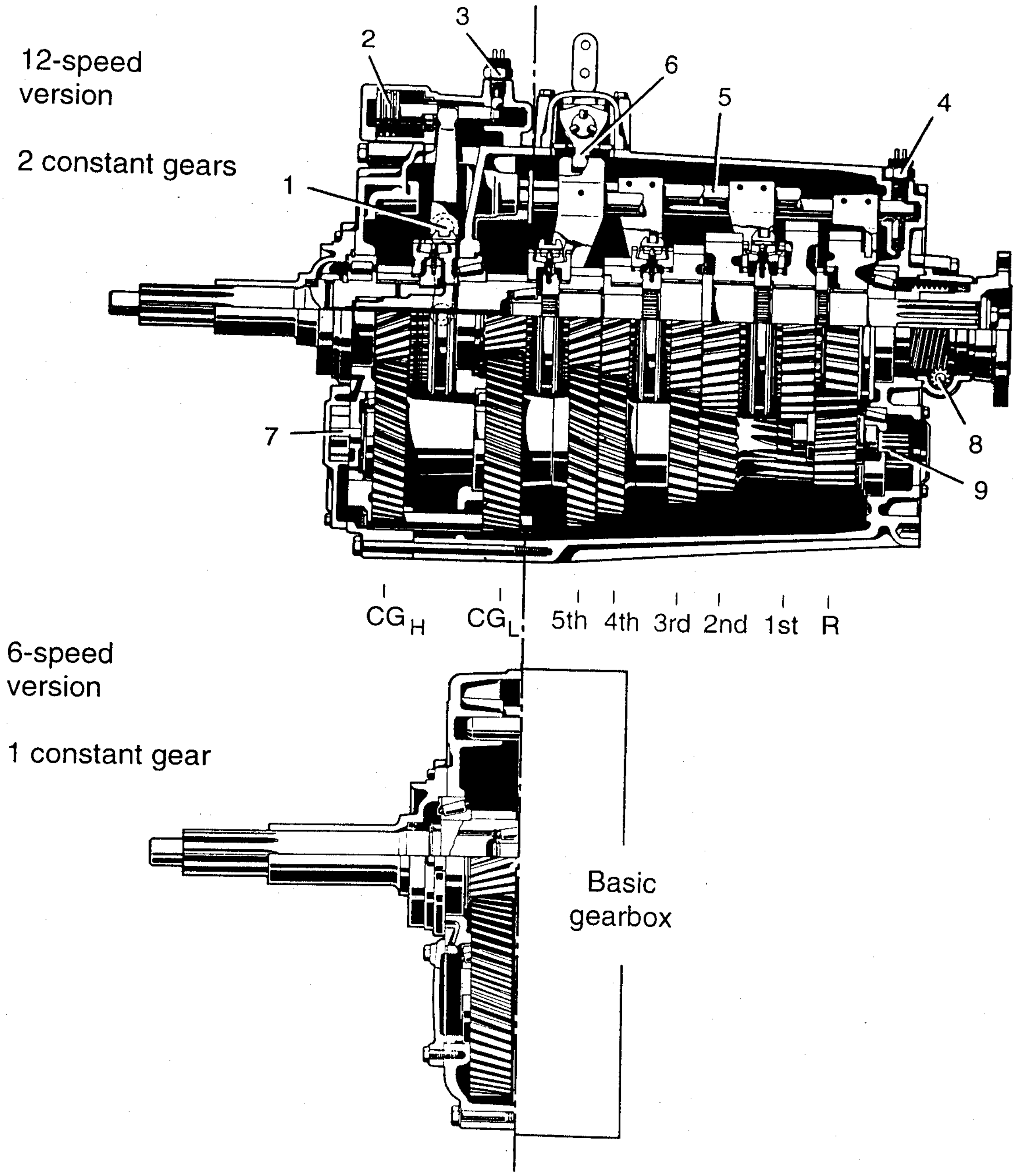


Figure 12.7. ZF S 6-66 6/12-speed manual commercial vehicle gearbox with ZF GV 66 splitter unit; *gearbox diagram Figure 6.36b*. *1* Gear shift fork; *2* Compressed-air cylinder; *3* and *4* Switches; *5* Selector bar; *6* Selector finger; *7* Oil pump; *8* Speedometer drive; *9* Power take-off shaft; CG_H Constant gear ratio high (high speed); CG_L Constant gear ratio low (low speed)

There is no design change to the basic gearbox. Please refer to Figure 6.36b to compare the gearbox diagrams.

In 6-speed basic gearboxes, the reverse and first gears are located directly at the output side bearings, to minimise the shaft deflection caused by the large tooth forces arising from the high transmission ratios (see Section 8.2). The gear pairs for second gear to fifth gear are mounted in sequence from left to right. The sixth gear of the basic gearbox is direct drive (direct drive gearbox). Two different constant ratio gear pairs are available in the optional splitter unit. The front-mounted splitter unit makes the 6-speed direct drive gearbox into a 12-speed overdrive gearbox (CG_H constant gear ratio high = high speed). The transmission shafts run in tapered roller bearings and cylindrical roller bearings.

The synchroniser for first and second gear is a double-taper synchroniser, reducing the shifting force required. Reverse gear is unsynchronised, and shifted by means of a dog clutch. All the other gears have single-taper synchronisers. The splitter unit is shifted by means of a gear shift fork *1*, in contrast to the selector forks in the basic gearbox. The shifting effort is provided by the compressed-air cylinder *2* located above the gear shift fork. This is operated by the driver using a pilot valve mounted on the gearshift lever.

The electric switch *3* mounted on the right-hand end of the splitter stage selector bar gathers information on the splitter stage selected. The selector bars *5* with their selector forks and the selector finger *6* moving the selector forks are shown in section in the top part of the gearbox. Depending on the gate selected, the selector finger grips one of the selector bars with an axial movement along its axis of rotation. In the top right-hand part of the gearbox the electric switch *4* can be seen, which gives the signal for the reversing lights. The locking pins that define the end of the selector bar travel are located on the opposite side of the shaft.

The gearbox housing is of rigid end-loaded design. The worm gearing *8* of the speedometer drive is located on the output flange. The serration gear *9* located on the output end of the countershaft enables a clutch-driven power take-off to be attached (Section 6.8). This also requires no design change to the basic gearbox, as for connecting the splitter unit. An oil pump *7* can optionally be mounted at the input end of the countershaft, which supplies the input side bearings with lubricating oil, and pumps the gearbox oil through a separate oil cooler if required.

6/ Three-Range 16-Speed Manual Commercial Vehicle Gearbox; ZF 16 S 109

The 16-speed commercial vehicle three-range gearbox in Figure 12.8 consists of a 4-speed basic gearbox with flange-mounted splitter unit and range-change unit (cf. also the gearbox diagram in Figure 6.44). The gearbox with a maximum input torque of 1100 Nm is used in upper mid range trucks (180–240 kW).

In the basic gearbox, reverse gear and first gear are located close to a bearing, in order to minimise shaft deflection. The come the second and third gear pairs in order. The fourth gear is direct drive. There are two different constant-ratio gear pairs available in the splitter unit. This increases the number of selectable gears to eight, "compressing" the gear sequence. The rear-mounted range-change unit is of planetary design, and has two gear steps: one direct step in which the planetary gear-set revolves as a block (toothed clutch), and a second step with a ratio larger than the overall gear ratio of the basic gearbox plus the gear step of the main gearbox (see Figure 6.40). This serves to double the gear sequence from 8 to a total of 16 selectable gears. The transmission shafts run in tapered roller bearings and in deep groove ball bearing. The central shafts of the planetary step (sun, ring gear, spider) are interlinked by means of a single-taper synchroniser, located on the output side beside the gearwheels of the planetary step.

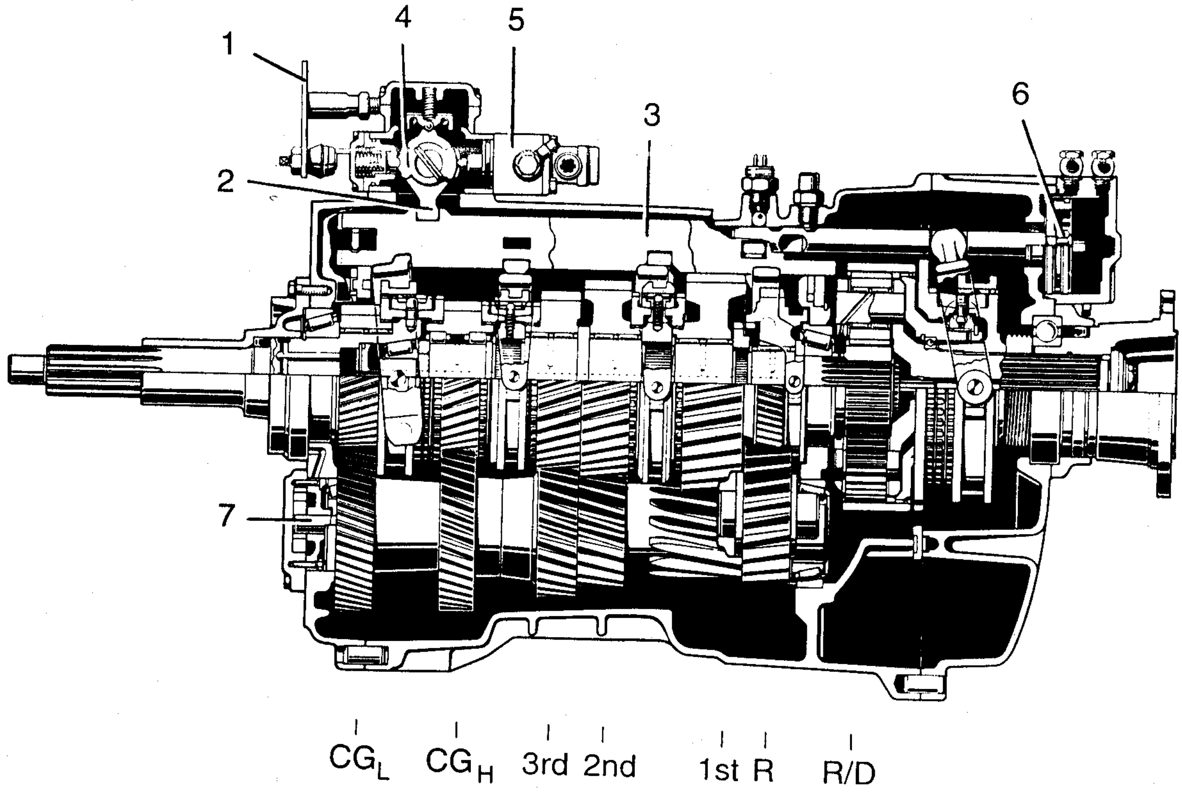


Figure 12.8. ZF 16 S 109 16-speed (2 x 4 x 2) manual commercial vehicle gearbox; *gearbox diagram and power flows in the gears Figure 6.44.* *1* Connection for the turning shaft remote control; *2* Selector finger; *3* Selector bars; *4* Cam plate; *5* Pilot valve; *6* Shift cylinder; *7* Oil pump; CG_H Constant gear ratio high; CG_L Constant gear ratio low; Range-change unit: *R* Range; *D* Direct

The output shaft of the gearbox runs on a fixed type ball bearing on the one side, and on the toothing of the planetary gears on the other side. Since no major axial forces arise in the planetary step, the ball bearing of the output shaft is primarily required to counteract external forces from the propeller shaft in the gearbox housing.

All the gears except reverse have single-taper synchronisers. The gearshift sleeves are shifted by gear shift forks. The connection *1* for the turning shaft remote control can be seen in the upper gearbox housing. This moves the selector finger *2*, which in turn operates the selector bars *3*. A cam plate *4* can be seen on the rotational axis of the selector fingers, which activates the pilot valve *5* located on the right when the selector finger is moved axially (changing from gate 3–4 to gate 5–6), causing the shift cylinder *6* mounted above the output flange to automatically change the range-change unit. The splitter unit is also shifted pneumatically. The pneumatic cylinder that performs this function is physically located behind that of the range change unit, and cannot therefore be seen in this illustration. It is operated by the driver by means of a pilot valve mounted on the gear lever. In contrast to earlier designs, the pneumatic cylinders were integrated in the gearbox housing to save cost and space.

At the input end of the countershaft there is an oil pump *7* that supplies the bearings on the input side with lubricating oil, and pumps the gearbox oil through a separate oil cooler when necessary, since cooling the oil makes the gearwheels more resistant to pitting. The gearbox housing is of three-part, end-loaded design.

7/ Two-Range 12-Speed Manual Commercial Vehicle Gearbox; Eaton Twin Splitter

The gearbox in Figure 12.9 is an interesting type of countershaft selector gearbox (compare also with Figure 6.45). This is a 12-speed selector gearbox with a 4-speed basic gearbox and a 3-speed rear-mounted splitter unit. The special feature of the rear-mounted splitter unit is that it has a low-speed and a high-speed splitter stage in addition to the direct drive. Splitter mode is pre-selected by the driver at the gearshift lever, and implemented by means of pre-tensioned shifting elements when power is interrupted (clutch operation or deceleration).

The basic idea behind both the basic gearbox and the rear-mounted splitter unit is to reduce the overall length of the gearbox by reducing the face widths required. To achieve this, the rolling contact power is transmitted by two opposed countershafts. These two countershafts are located in the same plane as the gearbox main shaft for reasons of symmetrical loading. This distributes the load in one gearwheel stage to two meshings, thus halving the theoretical contact stress arising. In practice face widths could be reduced by 40%. This serves to achieve the desired reduction in overall length, at the expense of making the whole gearbox wider.

The load on the main shaft gearwheels is balanced by precise "float" on the main shaft in a radial direction. They are centred between their respective countershaft gearwheels under load. The rear-mounted splitter unit works on the same principle. Since the main shaft gearwheels are only guided and do not run in bearings, they are not capable of transmitting substantial axial forces. Helical gearing is therefore not appropriate for them. The noise level of spur gearing is improved by contact ratios and high contact gearing.

All gears except the friction-synchronised splitter stages are constant mesh. The splitter stages are shifted pneumatically. The gearbox housing is of end-loaded design, and has an internal bulkhead for the bearings of the main shafts and countershafts. The clutch bell housing is bolted in place.

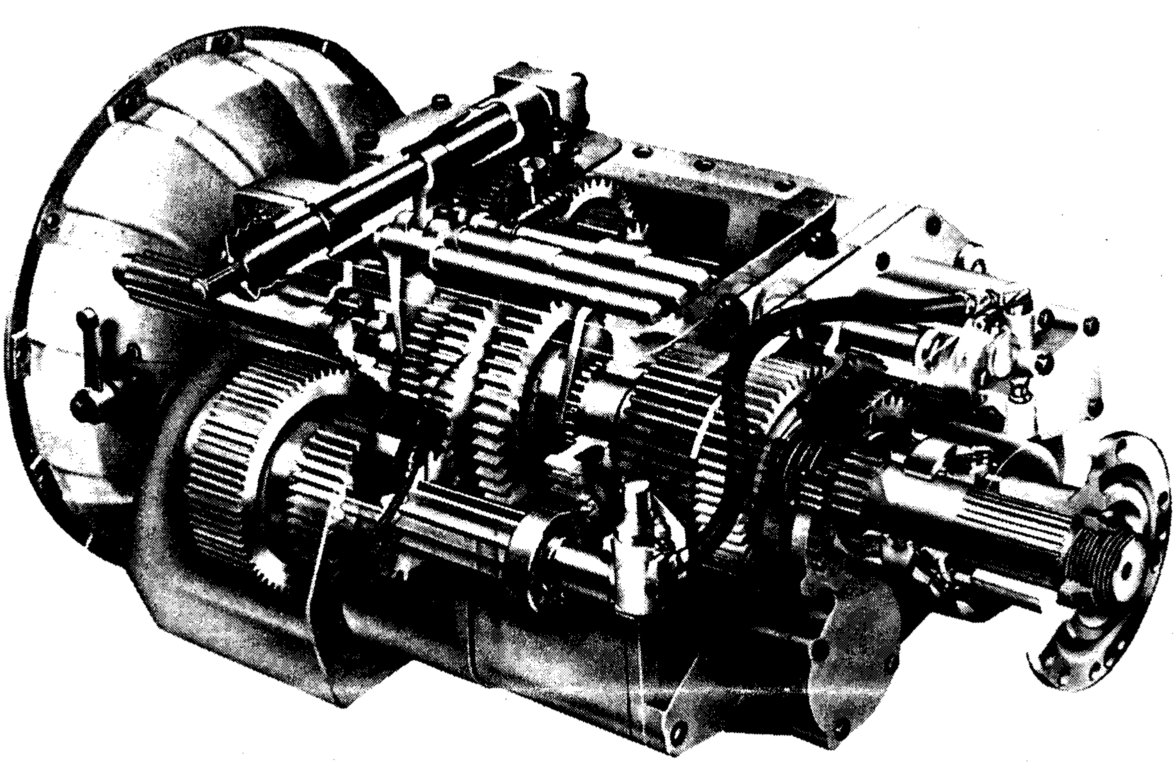


Figure 12.9. Eaton Twin Splitter 12-speed (4 x 3) manual commercial vehicle gearbox, of twin countershafts design; *gearbox diagram Figure 6.45*

12.2 Semi-Automatic Manual Gearboxes

Semi-automatic manual gearboxes are gearboxes that assist the driver when moving off and/or when changing gear, yet enable him to intervene when changing gear. The full range extending from entirely manual gearboxes to semi-automatic and fully automatic transmissions is principally evident in the case of commercial vehicles. All the manual gearboxes reviewed in Section 12.1 can in principle be automated in whole or in part by suitable moving-off elements and operating mechanisms (see also Sections 6.6.2 and 6.7.4).

12.2.1 Semi-Automatic Manual Passenger Car Gearboxes

Electronically controlled clutches are available, with product designations such as EKS (Electronic Clutch System, by Mannesmann Sachs) or EKM (Electronic Clutch Management, by LuK) for automating moving off and changing gear [12.40]. See production design Figure 12.16a.

12.2.2 Semi-Automatic Manual Commercial Vehicle Gearboxes

To fulfil the customer requirements highly integrated semi-automatic transmissions for commercial vehicles have been developed. The systems are known as "AS TRONiC" (ZF), "Geartronic" (Volvo), "Opticruise" (Scania), "SAMT B" (Eaton), "Telligent" (Mercedes-Benz) [12.41, 12.42].

The gearbox ZF 16 AS 2600 is shown in Figure 12.18. A master clutch pedal is no longer required. Fully automatic modes and manual drive modes are available.

8/ 16-Speed Semi-Automatic Manual Commercial Vehicle Gearbox; ZF Transmatic

The torque converter clutch transmission has secured a market niche alongside the manual transmission semi-automated by electronic systems and actuators. Torque converter clutch transmissions are particularly suitable for heavy transporters and special-purpose vehicles rated at 220 to 500 kW.

The gear unit in Figure 12.10 is a combination of torque converter clutch and 16-speed synchromesh gearbox (see also gearbox diagram, Figure 6.47). The gearbox as a whole is made up of standard parts, and can therefore readily be adapted to particular requirements. The synchromesh gearbox is a three-range gearbox with split and range-change unit. Its construction and arrangement of gears correspond to those of the gearbox in Figure 12.8, which can also be combined with the torque converter clutch shown here. Please refer to the discussion relating to Figure 12.8. The 16 S 220 selector gearbox shown here allows gearbox input torques up to 2200 Nm.

The torque converter clutch assembly provides the driver with the following controls and facilities:

- foot operated gearshifting clutch,
- manual gear change,
- automatic moving off using the torque converter,
- kick-down instead of shifting down if necessary.

The gear shifting clutch is foot operated by the driver in the conventional manner. The clutch pedal acts hydrostatically on the clutch release lever 2. Since a lot of force is required to disengage the clutch, the action is pneumatically servo assisted.

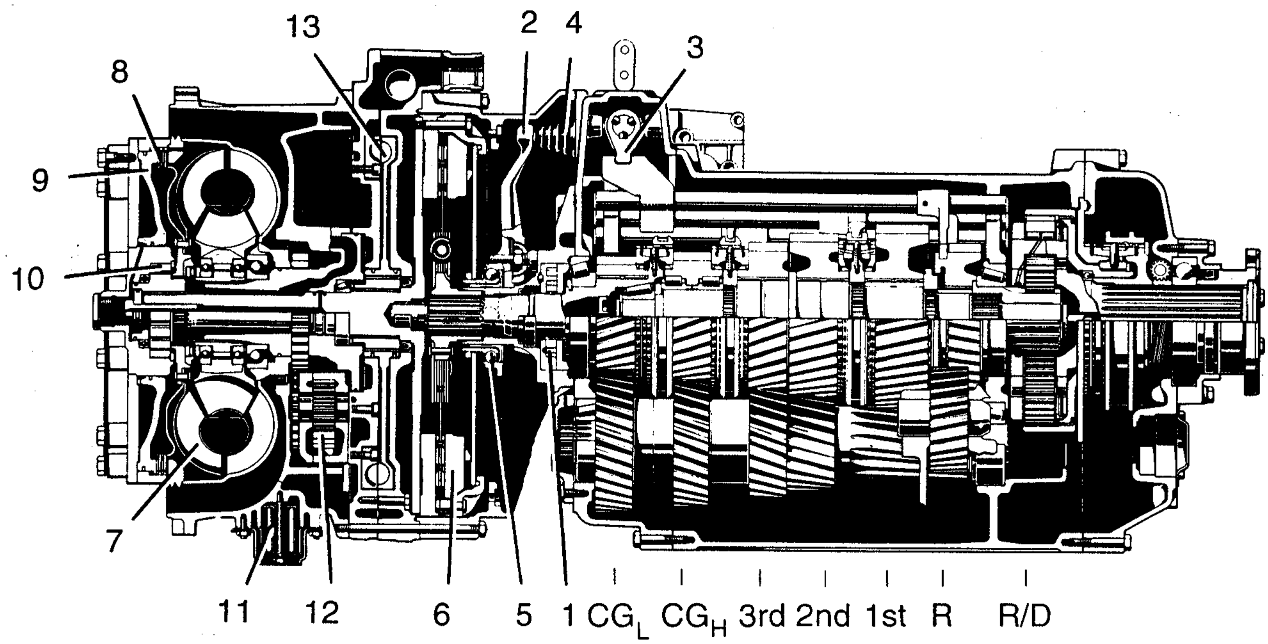


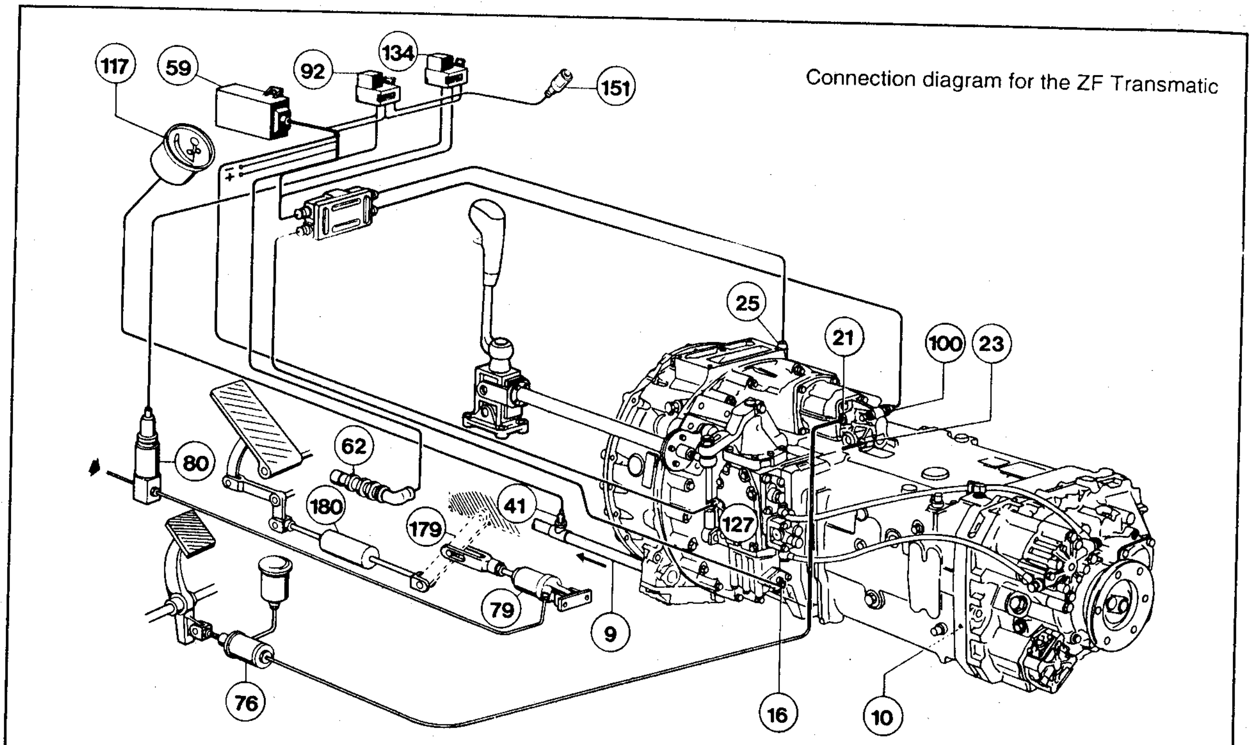
Figure 12.10. ZF-Transmatic 16-speed semi-automatic manual commercial vehicle gearbox (consisting of ZF WSK 400 + ZF 16 S 220); *gearbox diagram Figure 6.47*. *1* Oil pump; *2* Clutch release lever; *3* Selector finger; *4* Bellows; *5* Clutch release bearing; *6* Gearshifting clutch; *7* Torque converter; *8* Lock-up clutch; *9* Hydraulic piston; *10* Coasting freewheel; *11* Oil filter; *12* Oil pump; *13* Retarder; CG_H Constant gear ratio High; CG_L Constant gear ratio Low; Range-change unit: *R* Range; *D* Direct

Beside the selector finger *3* the bellows *4* is to be seen, which protects the piston rod of the compressed-air cylinder. The cylinder acts through the clutch release lever and the clutch release bearing *5* by opening the cup spring to disengage the gearshifting clutch *6*. It is now possible to change gear manually. The Trilok torque converter *7* is used for moving off. This increases the starting torque to 2 to 2.5 times the nominal torque. The torque converter makes the whole drive train very largely free of shocks and jolts. To prevent moving off with the gearshifting clutch slipping and the torque converter not locked up, this state is eliminated by an accelerator interlock (Figure 12.11). The increase in torque in the torque converter would destroy the clutch.

The torque converter has a lock-up clutch *8*, which is engaged by means of a circular hydraulic piston *9*. The cylindrical rollers of the coasting freewheel *10* can be seen below the torque converter lock-up clutch. This freewheel is located between the impeller and the turbine wheel of the torque converter, and grips when coasting. This harnesses the engine braking torque, and enables the vehicle to be tow-started.

The oil filter *11* of the torque converter clutch oil circuit is located below the torque converter. Above the oil filter is the oil pump *12*. The vacant space beside the torque converter is occupied by an engine-driven power take-off, if fitted. An integral hydrodynamic retarder (continuous brake, Section 11.5) *13* is fitted between the torque converter and the gearshifting clutch. It works in principle like a hydrodynamic clutch, with one side of the clutch (the stator) fixed to the housing. The braking torque of the retarder is controlled by the oil fill level.

The function of the retarder is to relieve the service brake in heavy vehicles. The torque converter clutch does not have to be flange mounted directly on the selector gearbox as in Figure 12.10. They can both be mounted separately from each other in the vehicle, and linked by a cardan shaft. The same applies to the link between the engine and the torque converter clutch.



9 Connection to heat exchanger
10 Connection from heat exchanger
16 Electrical connection for the neutral position switch
21 Connection for the clutch operating hydraulic line
23 Compressed air connection for clutch power assistance
25 Electrical connection for the lock-up clutch control
41 Temperature probe
59 Electronic module
62 Kick-down switch
76 Pick-up cylinder for activating the separator clutch
79 Stop cylinder for accelerator interlock
80 Solenoid valve for accelerator interlock
92 Relay for controlling the lock-up clutch
100 Electrical connection for pressure switch
117 Temperature display unit
127 Electrical connection to inductive pickup for turbine speed
134 Relay for accelerator interlock
151 Pilot lamp for lock-up clutch
179 Throttle lever on the injection pump
180 Throttle linkage spring loading

Figure 12.11. Connection diagram of the electrical, pneumatic and mechanical control components for the ZF Transmatic, model without retarder

If changing gear is automatically controlled rather than being controlled manually by the driver, the semi-automatic gearbox becomes a fully automatic selector gearbox, although with power interruption.

12.3 Fully Automatic Gearboxes

This section considers conventional automatic transmissions, comprising a torque converter and rear-mounted planetary type gearbox, and continuously variable automatic transmissions.

Conventional automatic transmissions currently have one standard for passenger cars and commercial vehicles. The transmission is fully automatically controlled, with the driver intervening only to the extent of inhibiting certain driving positions. Some types have manual selector levers for the driver to direct the transmission control unit to change up or down. The normal arrangement is to have control units that work entirely hydraulically or electro-hydraulically. There are also "adaptive controls" that adapt the shift program to the driver's driving style (performance or economy oriented), which it detects by means of sensors (see also Section 13.3.3).

12.3.1 Fully Automatic Passenger Car Gearboxes

9/ 5-Speed Conventional Automatic Passenger Car Gearbox; Mercedes-Benz W5A 030

The Mercedes-Benz W5A 030 5-speed transmission in Figure 12.12 is an example of a passenger car automatic transmission. This transmission was derived from the W4A 040 4-speed automatic transmission by adding a planetary step [12.3].

The gearbox housing with integrally cast clutch bell housing is divided into a front and a rear chamber by a bulkhead at the level of the breather unit *1*. The front chamber contains basically the same assemblies as in the 4-speed transmission, and the fully hydraulic control of the 4-speed part is also taken from the 4-speed transmission. The planetary gear-set *2* in the rear chamber rotates as a block in the first four gears, providing a speed increasing ratio in fifth gear, with the 4-speed part rotating as a block. The 4-speed part comprises a Ravigneaux set *3* with a rear-mounted simple planetary gear-set *4*. The front chamber is separated from the torque converter by a cover *5*, which also accommodates the primary oil pump *6*. This cover has channels for the oil pump, and also serves as the cylinder for the actuating piston *7* of the multi-disc brake *8*. Some of the brakes are multi-disc and belt brakes *9*. The belt brakes in particular are very compact, using the outer surface of a clutch disc carrier *10* as brake surface.

There is no lock-up clutch in the torque converter *11* (relatively "hard" torque converter; cf. Section 10.4), the rationale being as follows: lock-up clutches are only engaged for a small proportion of the time in normal operation. Since the torque converter requires a given amount of space, fitting a lock-up clutch reduces the space available for the hydrodynamic circuit, thus impairing the efficiency of the torque converter.

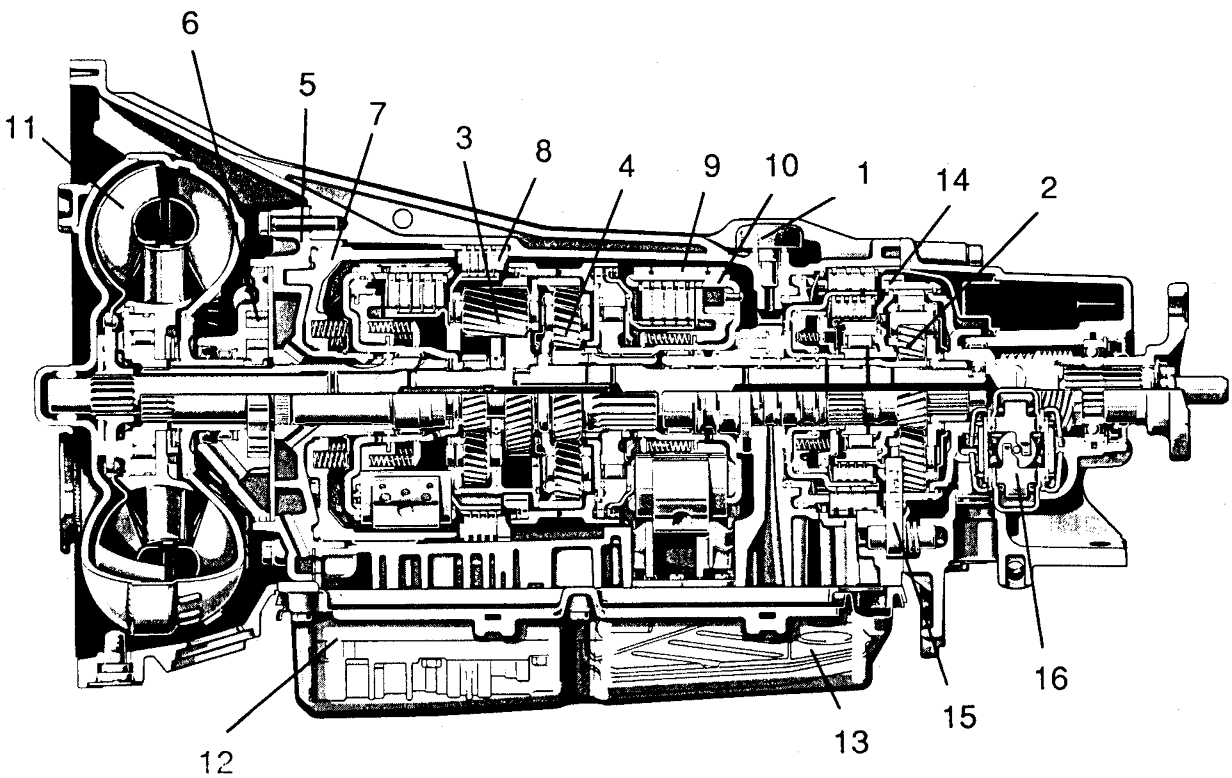


Figure 12.12. Mercedes-Benz W5A 030 5-speed automatic gearbox.
1 Breather unit; *2* and *4* Planetary gear-set; *3* Ravigneaux set; *5* Cover; *6* Primary oil pump; *7* Actuator piston; *8* Multi-disc brake; *9* Belt brake; *10* Clutch disc carrier; *11* Torque converter; *12* Control plate; *13* Sump; *14* Parking interlock gear; *15* Parking lock detent; *16* Centrifugal governor

As already mentioned, the power flow in the transmission is controlled by brakes, clutches and freewheels, linking the sun gears to each other and to the housing. This arrangement is examined in detail in Section 6.6.3, and is in principle identical in all conventional automatic transmissions. The brake and clutch disc sets are clearly shown in the whole transmission. They are shifted by circular hydraulic pistons. There are coil springs or a cup spring to return each piston. The oil activating the piston passes through holes in the transmission shafts or the housing walls. The numerous sealing rings between the rotating parts, and between them and the housing, serve to seal off spaces through which this oil passes (see also Section 11.4 "Gearbox Sealing"). The control plate *12* of the hydraulic transmission control unit has valves directing the flow of oil into the various channels to control the transmission, and is located on the bottom of the transmission. The control plate is enclosed by the sheet steel oil pan *13* containing the oil sump.

The parking interlock gear *14* is located in the rear chamber, and locks the transmission output shaft by means of the parking interlock detent *15*. The gearing of the parking interlock gear is mounted externally on the ring gear of the planetary gear-set, saving axial space. The parking interlock is necessary in automatic transmissions because when the engine is at rest and the oil pump is therefore not providing power, the transmission cannot provide force locking through its shifting elements (see Section 9.1.3). There can thus be no transmission of torque between the input and output shaft of the transmission, and the idle engine cannot prevent the vehicle rolling away. Since this also makes it impossible to tow the vehicle, this transmission has a secondary oil pump mounted on the shaft of the centrifugal governor *16*, which provides the control pressure in the oil when the vehicle is moving and the engine is at rest. The centrifugal governor actuates a pressure control valve, and depends on the velocity of the vehicle, which supplies a speed-dependent control pressure.

10/ Continuously Variable Passenger Car Gearbox; Ford CTX

The desire to equip a vehicle with a continuously variable automatic transmission is almost as old as the automobile itself. After numerous developments throughout the history of automotive engineering, none of which proved viable, continuously variable transmissions were again developed by various manufacturers in the 1980's to the point where they could go into production. Please refer to Section 6.6.4 for a discussion of their principle of operation. All mechanical CV transmissions transmit torque by means of friction. There will therefore inevitably be a goal conflict at all operating points in the transmission between ensuring sufficient contact pressure for the friction elements, and minimising pump power. This is one of the transmission's two main functions, the other being controlling the transmission ratio, which affects the driveability and fuel consumption of the vehicle [12.4].

Figure 12.13 presents the Ford CTX continuously variable transmission, used in sub compacts with an engine rating of around 50 kW. The heart of this transmission is the thrust link chain *1* that runs between two axially adjustable taper discs.

There is a flywheel *2* with vibration damper at the transmission input. Since the continuously variable part of this transmission is not capable of a ratio of zero (in contrast to geared-neutral transmissions), a master clutch is necessary for moving off from rest. The clutch fitted is a wet running multi-disc clutch *3*, and appears in the figure on the left beside the planetary gear-set *4* at the transmission input. It links the input shaft via the spider of the planetary gear-set to the primary side of the belt drive transmission. There are also some CV transmissions that use a Trilok converter for moving off (ZF Ecotronic CFT20E transmission). The oil pump *5* is driven by a shaft in the hollow transmission input shaft dependent on the engine speed, and provides the control pressure for actuating the shift elements and the taper disc adjustment. The planetary gear-set has planet gear pairs, enabling a change in direction of rotation. This is achieved by operating the multi-disc brake *6*, and acts as the reverse gear of the transmission [12.5].

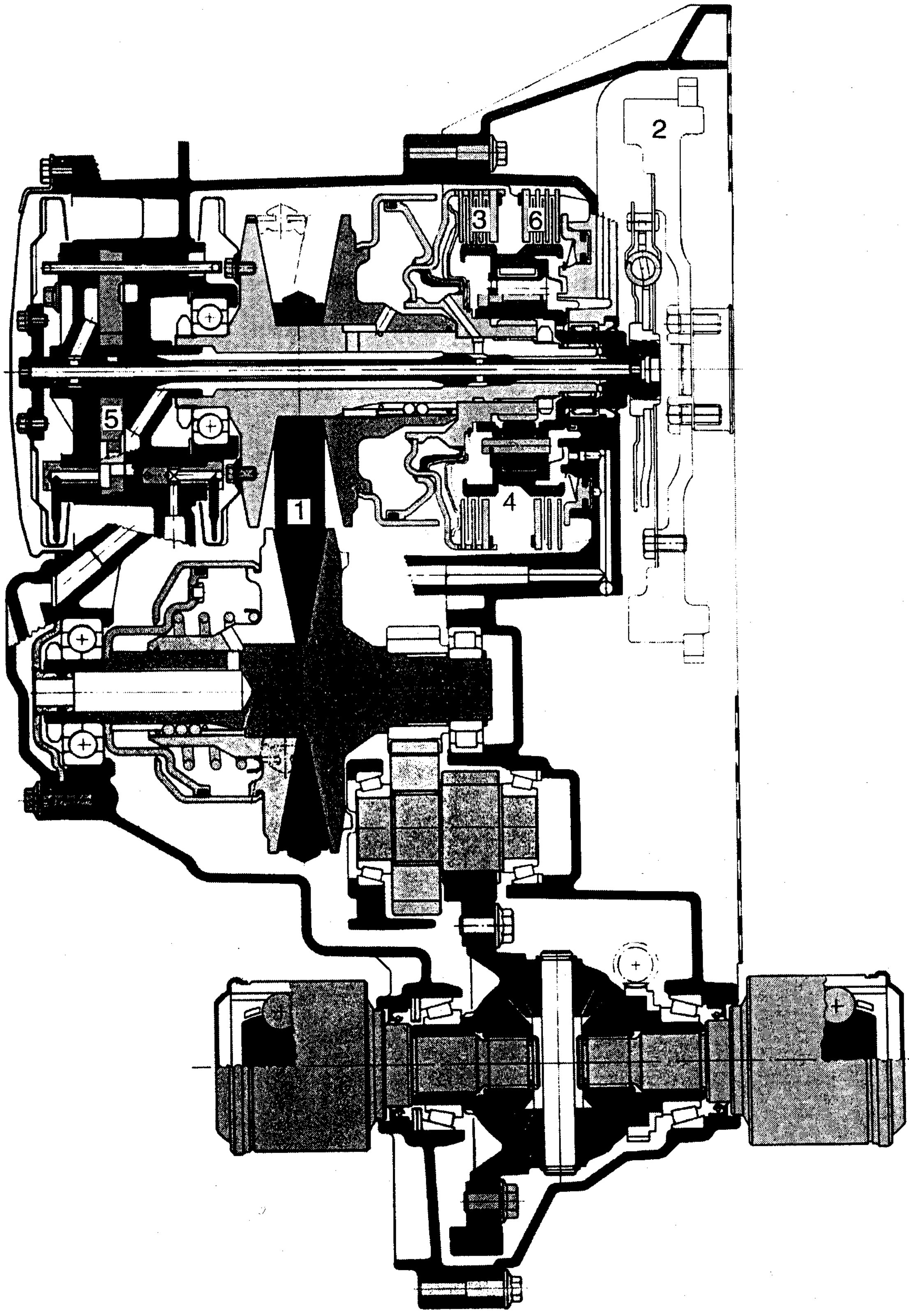


Figure 12.13. Ford CTX continuously variable passenger car gearbox; *schematic diagram 6.32b*. *1* Thrust link chain; *2* Flywheel; *3* Multi-disc clutch; *4* Planetary gear-set; *5* Oil pump; *6* Multi-disc brake

The transmission control is fully hydraulic, and related to rotational speed and load. The throttle valve position is fed into the control as a load signal. The speed signals are detected hydraulically using pitot chambers that produce a pressure signal through the oil movement. The transmission ratio is set by means of the pressure on the adjustable primary side taper disc. The effective radius on the secondary side is thus derived from the fixed belt length.

The axle gearbox is integral to the gearbox housing, giving the transmission its name (CTX - **c**ontinuously variable **trans**a**x**le; in this case, transaxle = transmission + axle). CV transmissions for more powerful rear-wheel drive vehicles are also under development.

12.3.2 Fully Automatic Commercial Vehicle Gearboxes

11/ 6-Speed Automatic Commercial Vehicle Gearbox; ZF 6 HP 600

The 6 HP 600 automatic transmission for commercial vehicles, buses and special-purpose vehicles shown here (Figure 12.14) is one of the ZF AG Ecomat range. There are 4- to 7-gear versions of this transmission for various power classes (6 HP 600: to 1600 Nm). The gearbox diagram and the shifting elements are shown in Figure 6.48.

The torque converter *3* has a lock-up clutch *2*, actuated by a separate hydraulic piston. After the torque converter housing with its pump blades is the engine-driven primary oil pump drive *8* connected by a gear interstage.

The next assembly before the selector gearbox part is the retarder *4* (see also the description of the Transmatic, Figure 12.10). The retarder used here is a refinement of that used in the Transmatic, principally improving the fan losses of the retarder when running without oil.

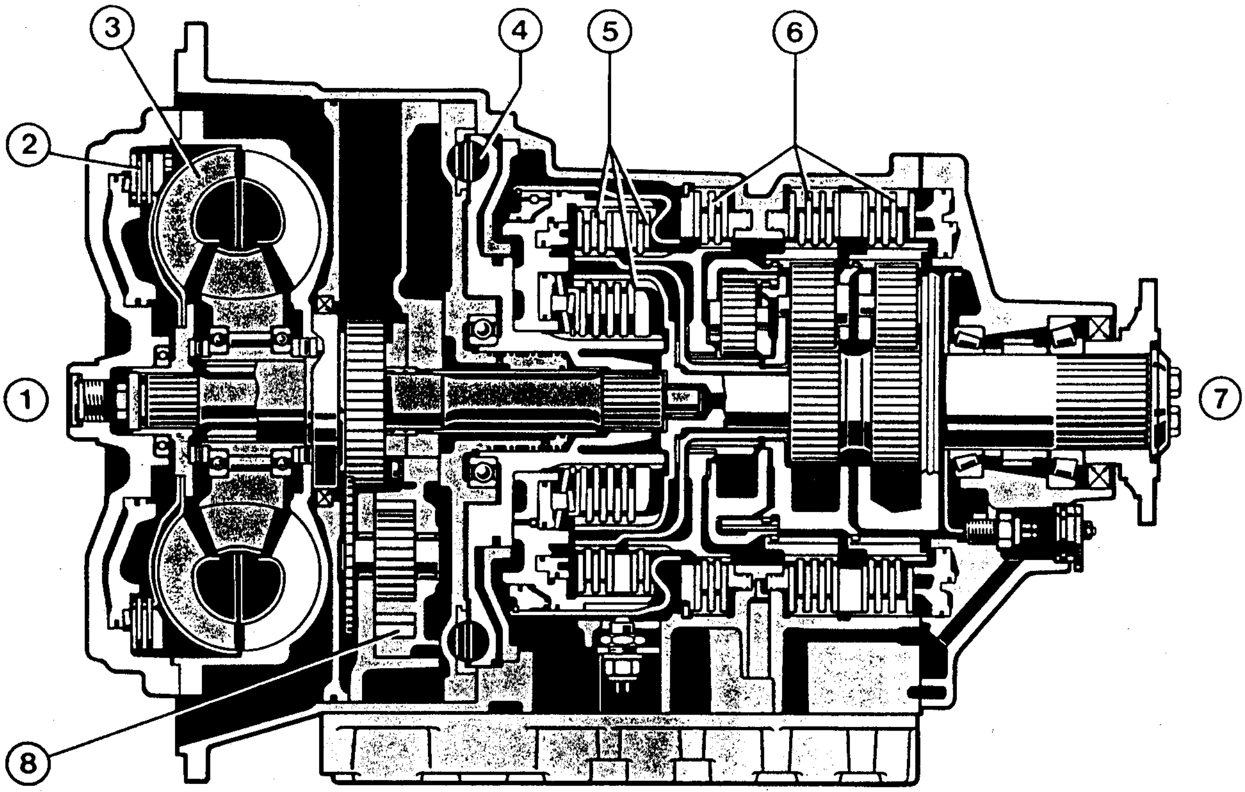


Figure 12.14. ZF 6 HP 600 6-speed automatic gearbox;
gearbox diagram and power flows, Figure 6.48. 1 Input; *2* Torque converter lock-up clutch ; *3* Torque converter ; *4* Retarder; *5* Clutches; *6* Brakes; *7* Output; *8* Oil pump

The air constantly present in the retarder generates braking torque that impairs the efficiency of the transmission. To reduce these losses, the stator fixed to the housing is split to form two discs, one of which can be rotated through half a blade increment relative to the other. This rotation opens the stator blades, allowing the air impelled by the rotor to flow past them. The rotating force is provided by springs mounted at the periphery, which are pressed together by the moments arising at the turning disc when the retarder is filled with oil, closing the stator blades and providing maximum braking torque.

The selector gearbox proper can again be subdivided into two sections, the space for the clutches, and the space accommodating the brakes and planet gear-sets. The three gearshifting clutches *5* are grouped together on one clutch carrier. (The sequence of shifting actions in automatic transmissions is described in Section 6.6.3). The inner clutch, characterised by its large number of discs, is engaged by the annular piston located beside it on the input side. The two outer clutches share the same disc external gearing. The block required to engage the clutches is located between the two disc sets of the clutches. The left-hand one of the two clutches is activated directly by the annular piston beside it acting on it. The piston of the right-hand clutch is located at the left-hand outermost diameter of the clutch carrier, and acts on the disc set on the opposite side through a pressed steel part enclosing the entire clutch carrier.

The brakes *6* and the planet gear-sets are located on the output side. The disc sets of the brakes with their actuating pistons and blocks are mounted on the inner wall of the housing. The application force of the hydraulic pistons and of the clutches must be adapted to the torque to be shifted for the particular change of ratio. This is carried out by an electronic shifting pressure control, and by means of step pistons with different effective piston areas.

The speed sensor for the output speed or the velocity signal can be seen on the output side, and that of the turbine speed signal in the lower central area. Both signals are fed to the electronic transmission control unit.

Since this transmission has one single oil circuit for the torque converter, retarder, shift control, lubrication and for heat dissipation, a heat exchanger is flange mounted on the transmission output (not shown) to use the cooling water circuit to cool the oil.

12/ 6-Speed Automatic Gearbox for Heavy Special-Purpose Vehicles; Renk HSRM 226.22

The HSRM 226.22 6-speed automatic gearbox is part of the REMAT family of power-shift transmissions with 5 to 7 speeds for input power ratings of up 735 kW. Figure 12.15 shows a 6-speed gearbox with primary retarder *9*, modulation clutch *19* and flange-mounted transfer box *17*.

The Renk/SRM type torque converter is locked up by the lock-up clutch *3*. The modulation clutch *19* provides controlled power distribution between the drive system and the engine-driven power take-off *20*. This enables airport fire appliances for example to have a combined drive for propulsion and for the water pump ("pump and roll"), eliminating the need for a dedicated engine for the water pump [12.43].

12.4 Further Examples

This section presents further examples of transmission designs, without commenting on them explicitly.

1 Input flange
2 Adaptor element for cardan shaft connection
3 Converter lock-up clutch
4 Impeller
5 1st turbine stage
6 Reactor
7 2nd turbine stage
8 Free-wheel roller
9 Retarder
10 Multi-disc brake
11 Multi-disc clutch Splitter unit
12 Multi-disc clutch High gear stages
13 Multi-disc brake Mean gear stages
14 Multi-disc brake Low gear stages
15 Multi-disc brake Reverse gear
16 Trailing oil pump drive, tachometer drive and PTO for emergency steering pump
17 Drop box with longitudinally lockable differential, torque distribution 1:1
18 Output flange
19 Modulation clutch
20 Powershiftable PTO

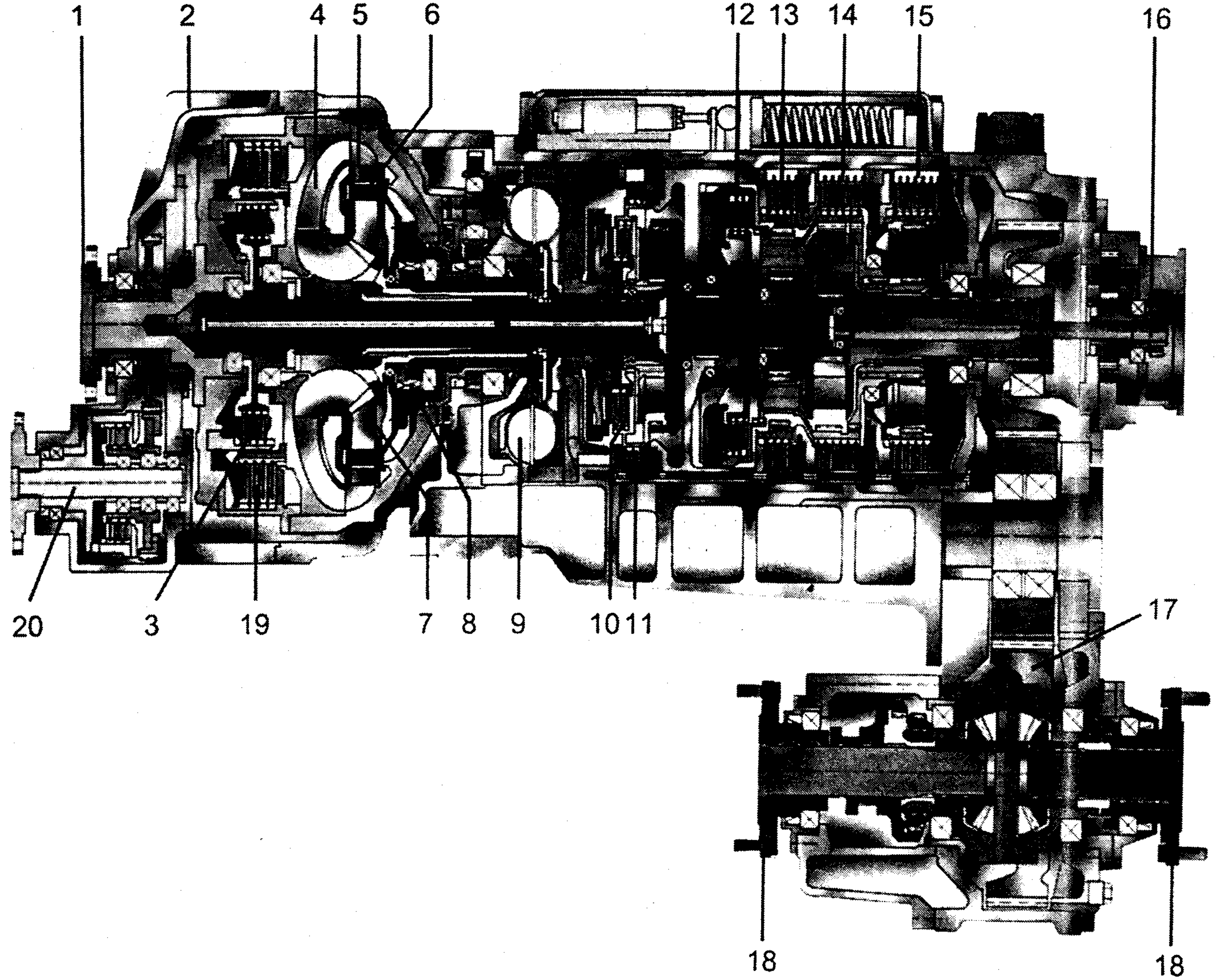
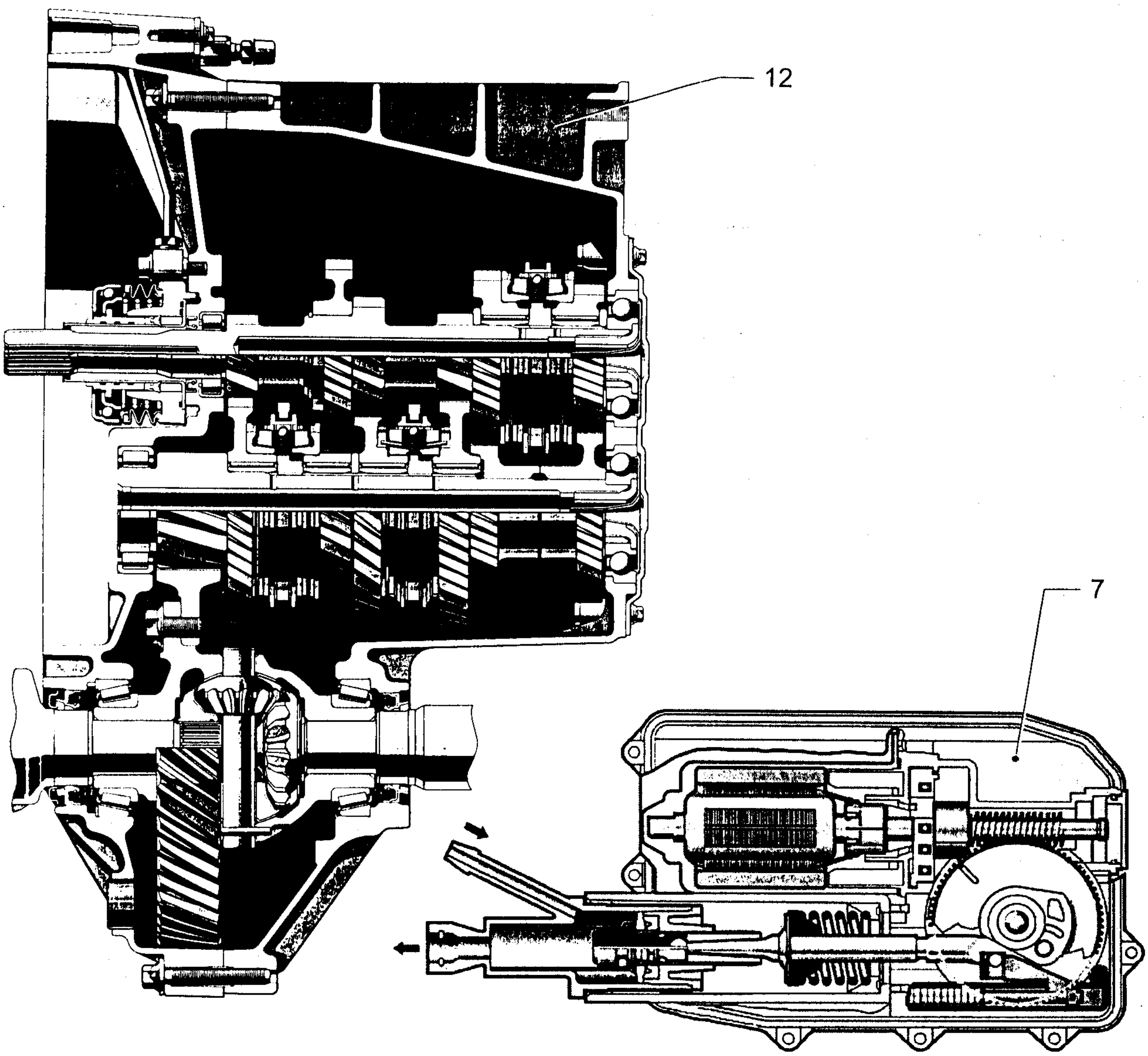


Figure 12.15. Renk HSRM 226.22 (REMAT-type): 6-speed automatic gearbox



1 Gearshift lever
2 Gearshift intention identification
3 Gearshift cable control
4 Selector cable control
5 Self-adjusting clutch (SAC)
6 Gear identification
7 Master clutch actuator
8 Clutch travel sensor
9 Master cylinder
10 Axle drive shaft, right
11 Flywheel mass
12 Manual gearbox
13 Axle drive shaft, left

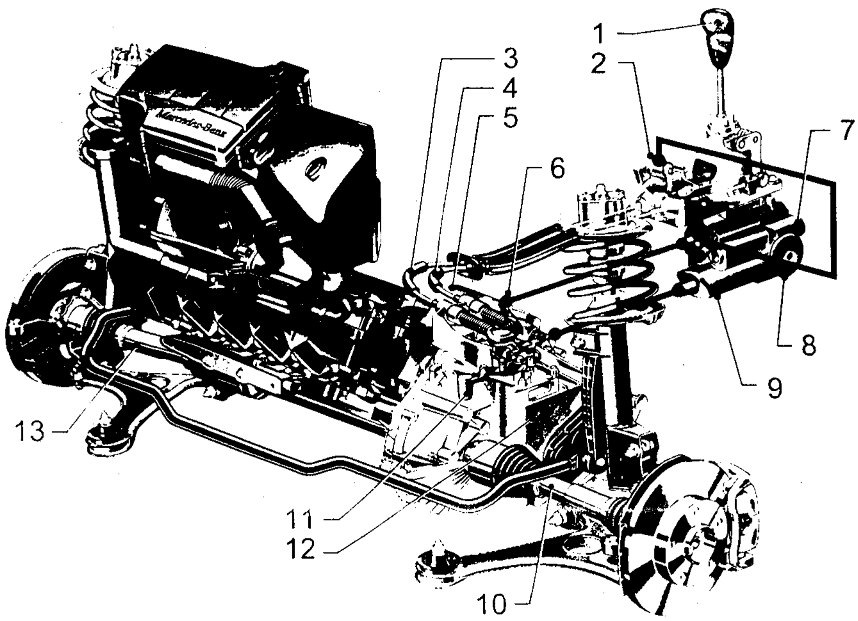


Figure 12.16a. 13/ 5-speed semi-automatic manual passenger car gearbox with automatic master clutch Mercedes-Benz SG 150 (A-Class) [12.44]

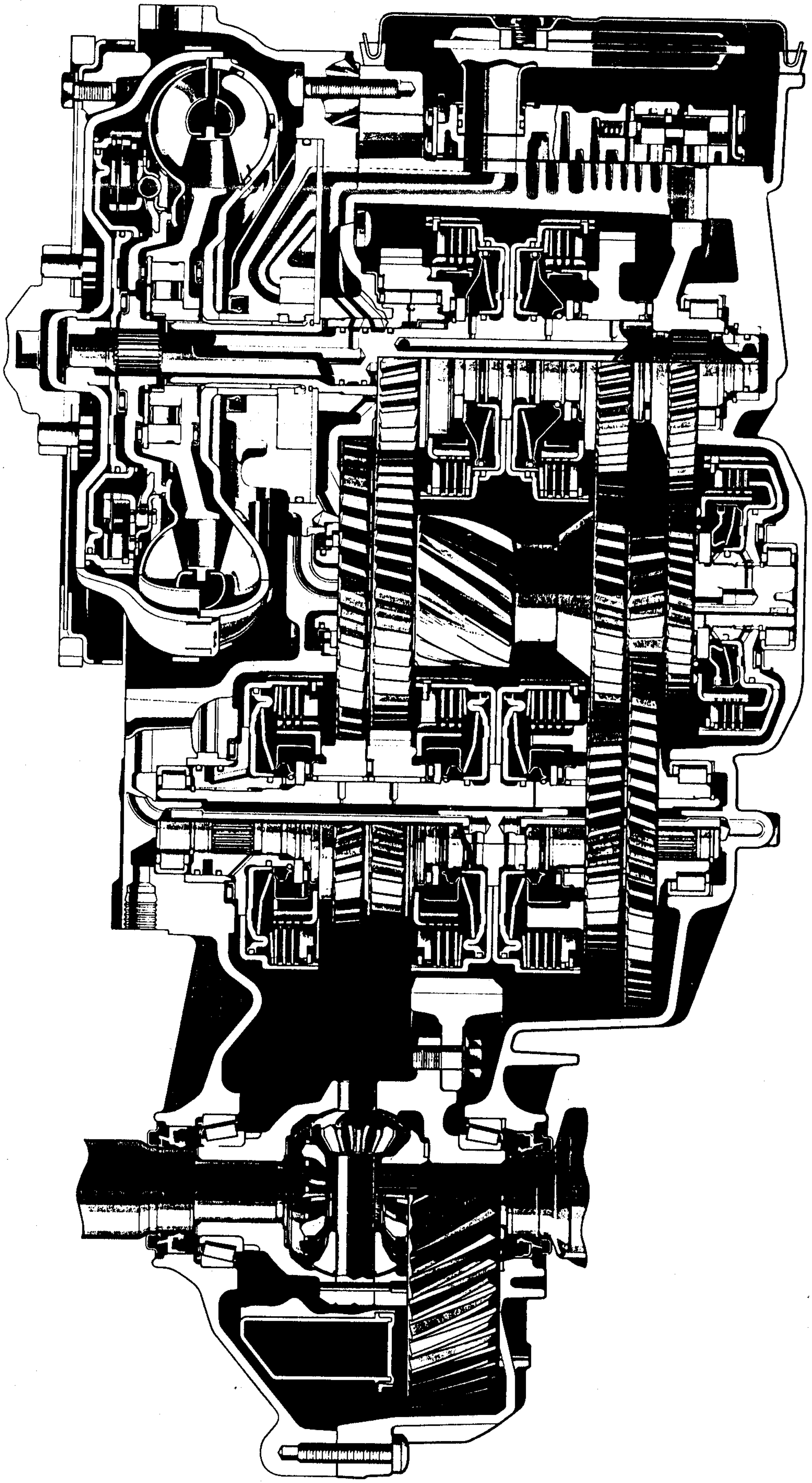


Figure 12.16b. 14/ 5-speed automatic passenger car gearbox, countershaft type Mercedes-Benz W5A 180 [12.45]

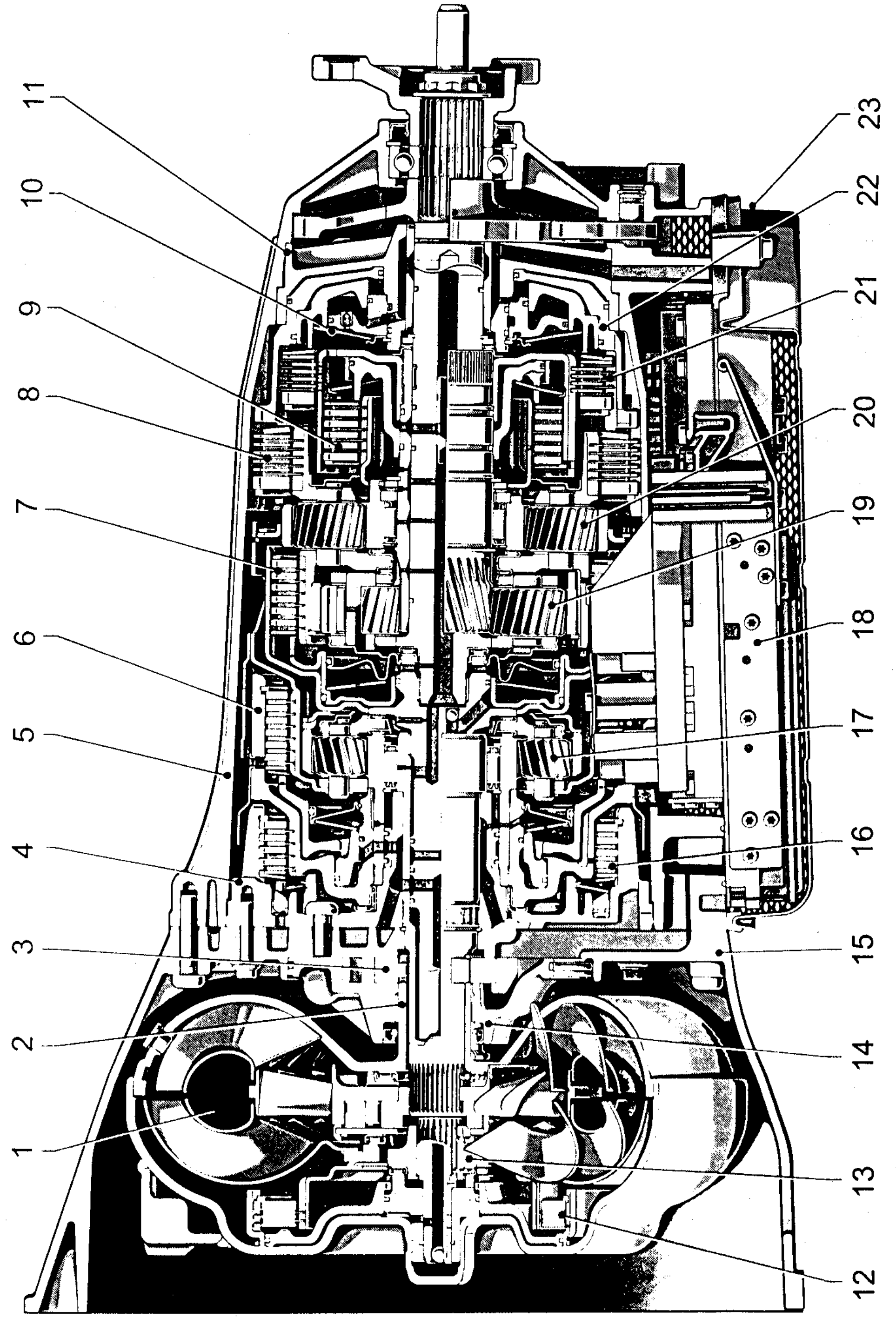


Figure 12.16c. 15/ 5-speed automatic passenger car gearbox Mercedes-Benz 5WA 580 [12.46]

1 Torque converter
2 Input flange oil pump
3 Oil pump
4 Front cover
5 Gearbox housing
6 External disc carrier
7 Multi-disc clutch C2
8 Multi-disc brake BR
9 Multi-disc clutch C3
10 Recompression spring B2/BR
11 Cylinder BR
12 Lock-up clutch
13 Flange turbine wheel
14 Oil pump housing
15 Torque converter housing
16 Multi-disc brake B1
17 Input planetary gear-set
18 Control unit
19 Centre planetary gear-set
20 Output planetary gear-set
21 Multi-disc brake B2
22 Piston BR
23 Oil sump

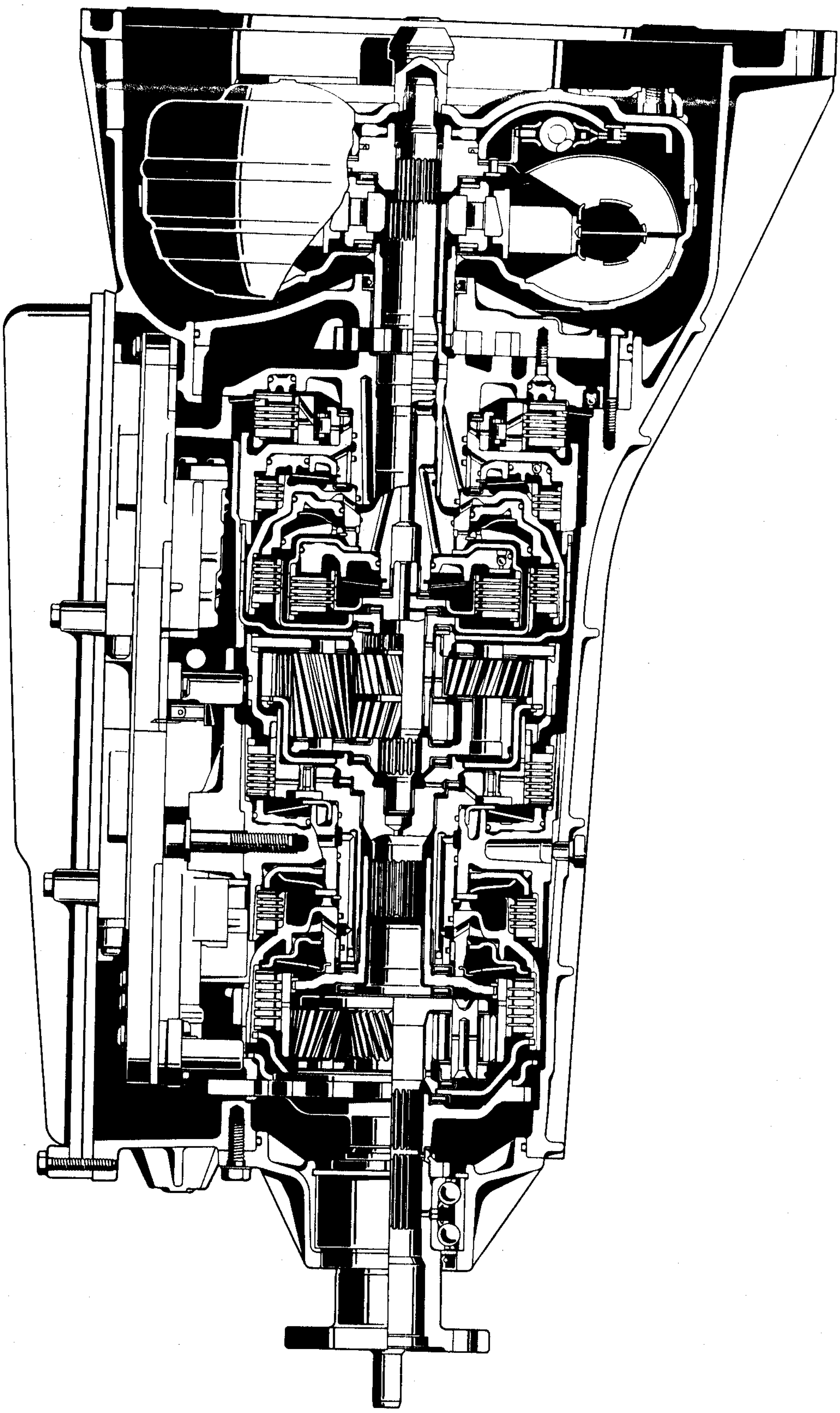


Figure 12.17. 16/ 5-speed automatic passenger car gearbox ZF 5 HP 18; *gearbox diagram and power flows, Figure 6.29*

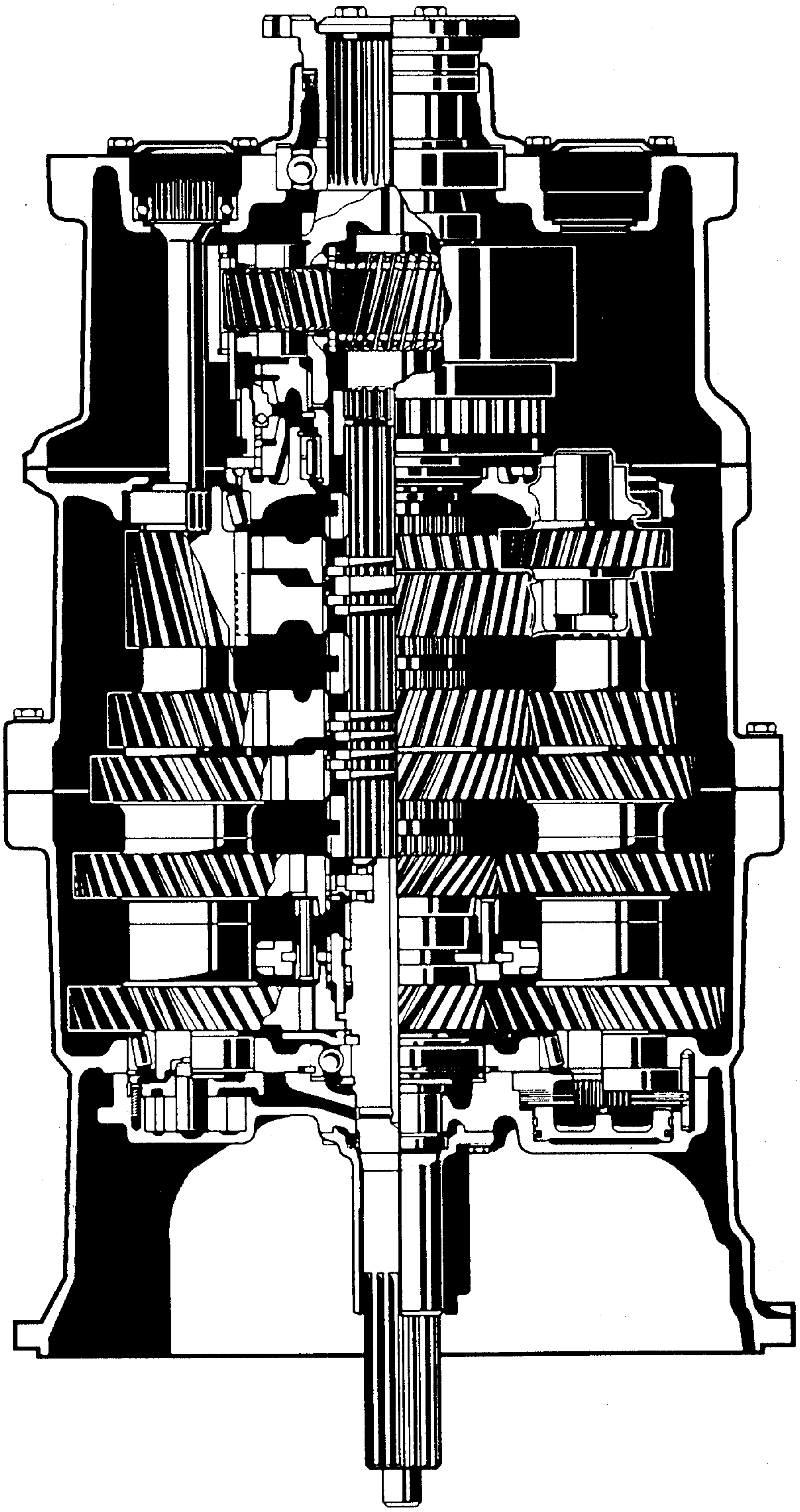


Figure 12.18. 17/ Three-range 16-speed manual commercial vehicle gearbox, double countershaft type ZF-AS TRONiC 16 AS 2600 [12.47]. Range gearbox 2×4×2. Front-mounted splitter and rear-mounted range-change unit shifted by synchronisers. Main gearbox constant-mesh gearshift with speed matching by means of the engine and a gearbox brake

12.5 Final Drives

Vehicle power trains were systematically considered in Section 6.9. The gearbox diagrams used there are reproduced in the following examples for the sake of completeness.

12.5.1 Typical Designs, Passenger Cars

From the multiplicity of production designs, the following final drives are considered:

- spur gear final drive, Figure 12.19,
- bevel gear final drive, independent assembly with hypoid drive, Figure 12.20,
- bevel gear final drive, integral to axle gearbox housing, Figure 12.21,
- worm gear final drive, Figure 12.22.

The final drive of the Opel Kadett D (1984 model) is shown in Figure 12.19 as an example of a spur gear final drive. The torque is transmitted by the output shaft of the selector gearbox *1* by means of a helical cut spur gear stage *2* to the cage of the differential gears *3*. The spur gear and the differential gear cage are bolted together.

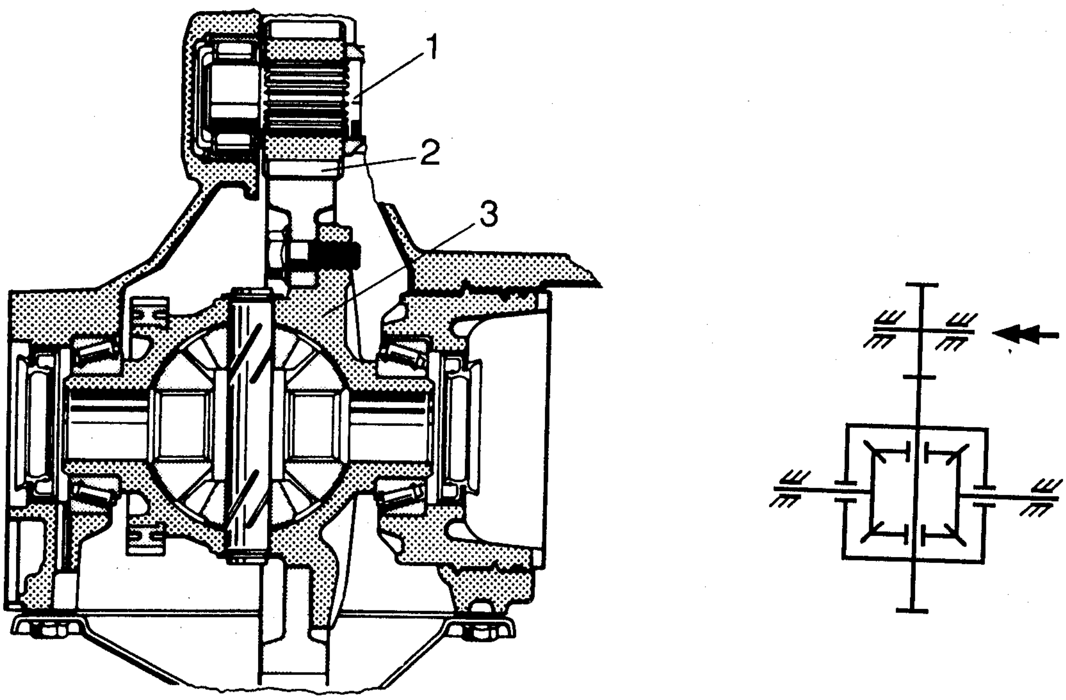


Figure 12.19. Spur gear final drive for vehicles with transverse mounted engine, Opel Kadett, 1984 model. *1* Output shaft; *2* Spur gear stage; *3* Differential gear cage

The Mercedes Benz mid-range W 124 axle gearbox (1988 model) is shown in Figure 12.20 as a representative example of a vehicle with bevel gear final drive. The input torque is transmitted to the axle gearbox from the cardan shaft via an elastic clutch. The drive pinion shaft and the differential cage run in tapered roller bearings. Differential bevel gears and the flanges for connecting the propeller shafts to the rear wheels run in plain bearings. Output is via propeller shafts with constant-velocity joints. The ratio in this example is $i_E = 3.07$. The crown gear has a diameter of $d_m = 185$ mm.

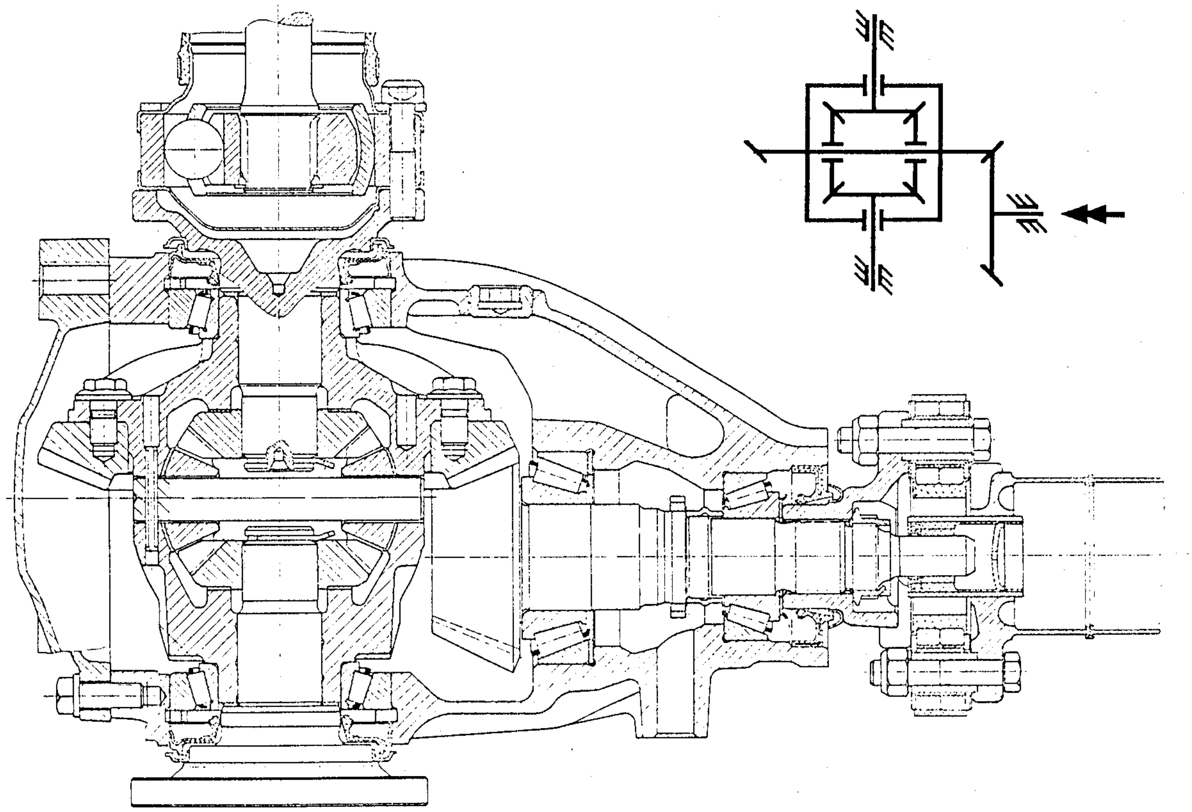


Figure 12.20. Example of an axle gearbox as an independent assembly for standard drive or all-wheel drive, Mercedes-Benz mid range, W 124, 1988 model

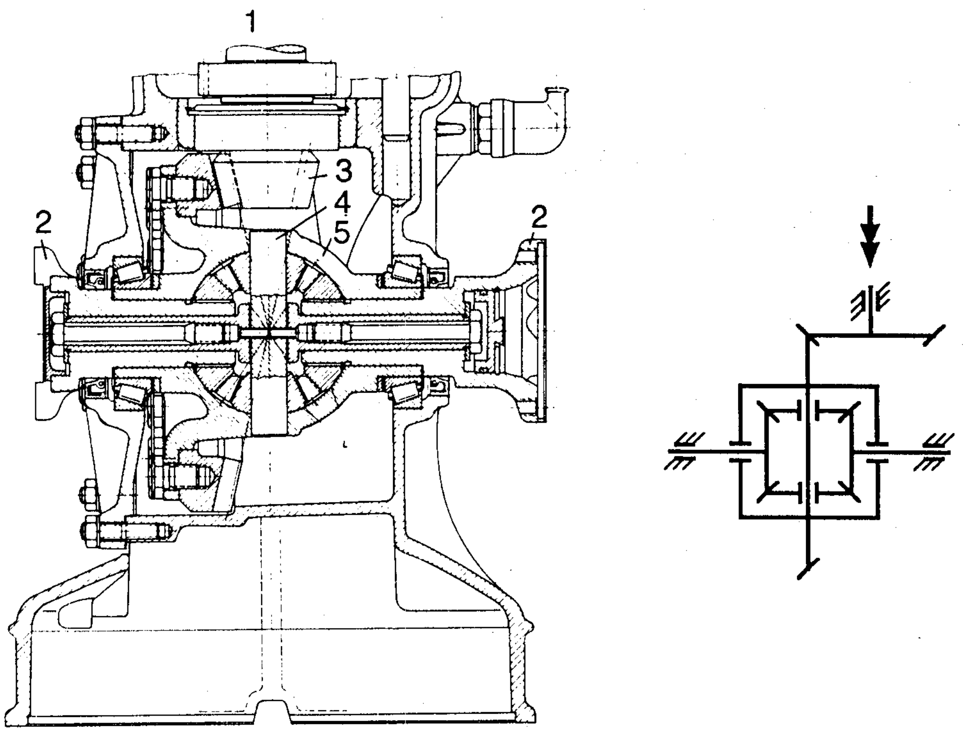


Figure 12.21. Bevel gear final drive with helical gearing, flange mounted directly to the transmission (Porsche 911). *1* Output shaft; *2* Flanges for propeller shafts; *3* Helical gear bevel drive; *4* Differential pins; *5* Differential gear cage

Figure 12.21 shows the axle gearbox of the Porsche 911. It is attached to the selector gearbox, and has a helical gear bevel drive.

A typical example of the worm gear drive is the Peugeot 404. Figure 12.22 shows its final drive. In this case the worm shaft *1* runs in angular ball bearings *2*. The differential cage *3* is of two-part construction to accommodate the worm gear *4*. Speed compensation between the wheels is provided by the bevel gear differential *5* in the differential cage. Differential bevel gears and axle bevel gears run in plain bearings.

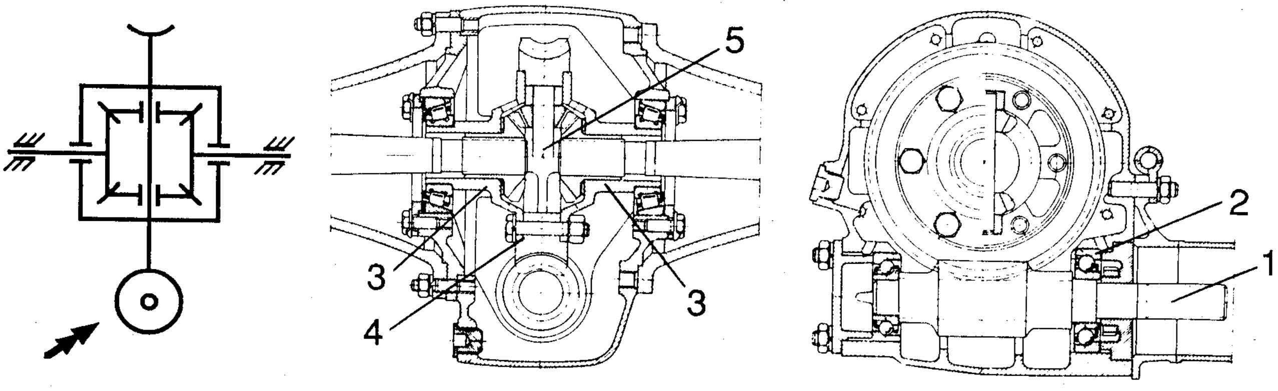


Figure 12.22. Axle gearbox of the Peugeot 404 with worm gear drive. *1* Worm shaft; *2* Angular ball bearing; *3* Differential gear cage; *4* Worm gear; *5* Bevel gear differential

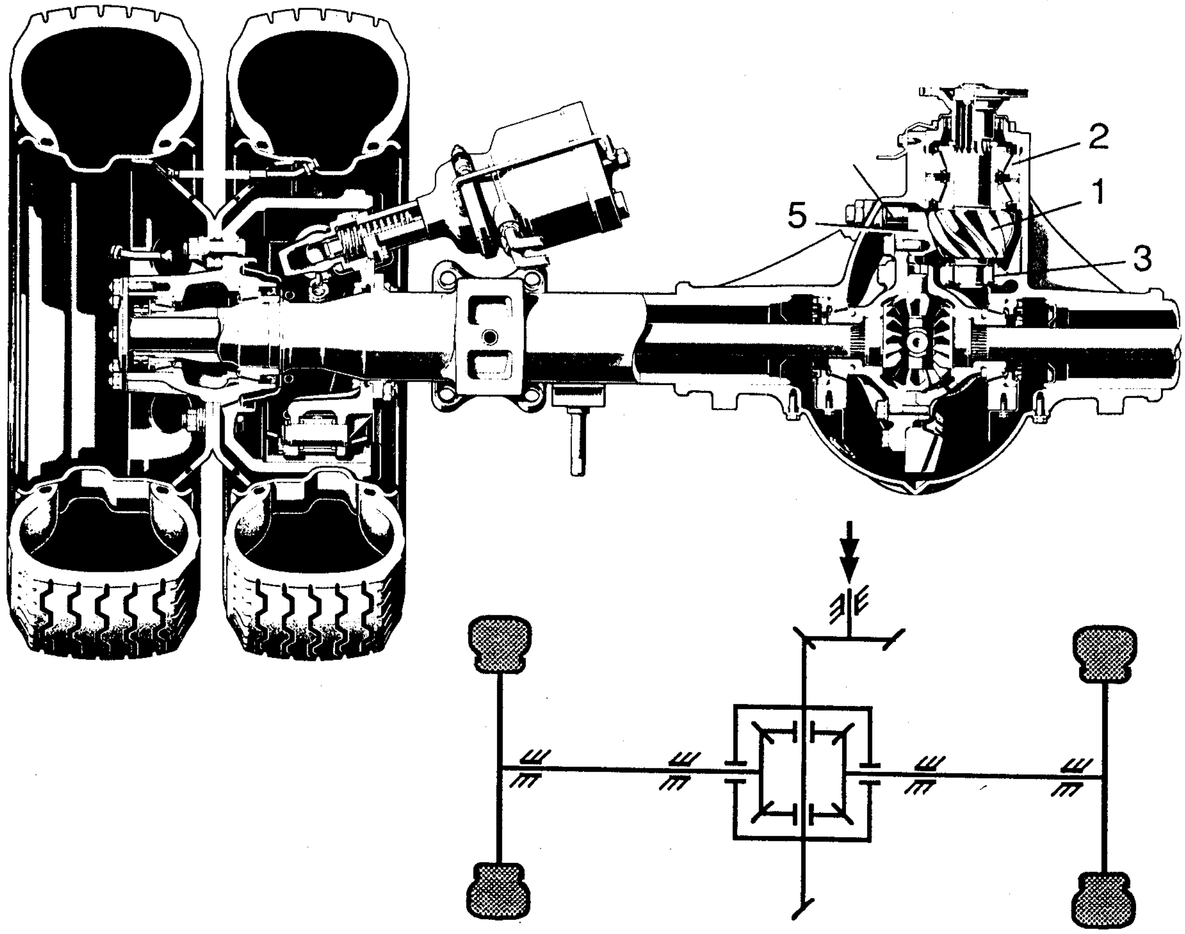


Figure 12.23. Axle with single-ratio, Mercedes-Benz HL 2/11, centre gearbox shown rotated through 90°. *1* Pinion; *2* Tapered roller bearing; *3* Cylindrical roller bearing; *4* Fulcrum pad; *5* Crown gear

12.5.2 Typical Designs, Commercial Vehicles

Numerous possible axle designs can be composed from the centre gearboxes and hub gearboxes described in Section 6.9.2 [12.6–12.8]. Four commonly used drive axles in commercial vehicles are:

1/ axle with single-ratio, Figure 12.23,
2/ two-speed axle, Figure 12.24,
3/ pinion axle (offset axle), Figure 12.25,
4/ outer planetary axle, Figure 12.26.

These axle designs are described below.

1/ Axle with Single-Ratio

These axles are generally used in light to medium-duty commercial vehicles, and offer ratios of up to approximately $i_E = 7.0$. Increasing the ratio or the power rating would involve enlarging the crown gear. This would further reduce the ground clearance underneath the centre gearbox.

Where high levels of power and torque are to be transmitted, a high ratio in the centre gearbox means that all the following parts have to be adapted to the increased torque, which militates against light-weight design and efforts to minimise unsprung weight. Figure 12.23 shows the Mercedes Benz HL 2/11 single-ratio axle.

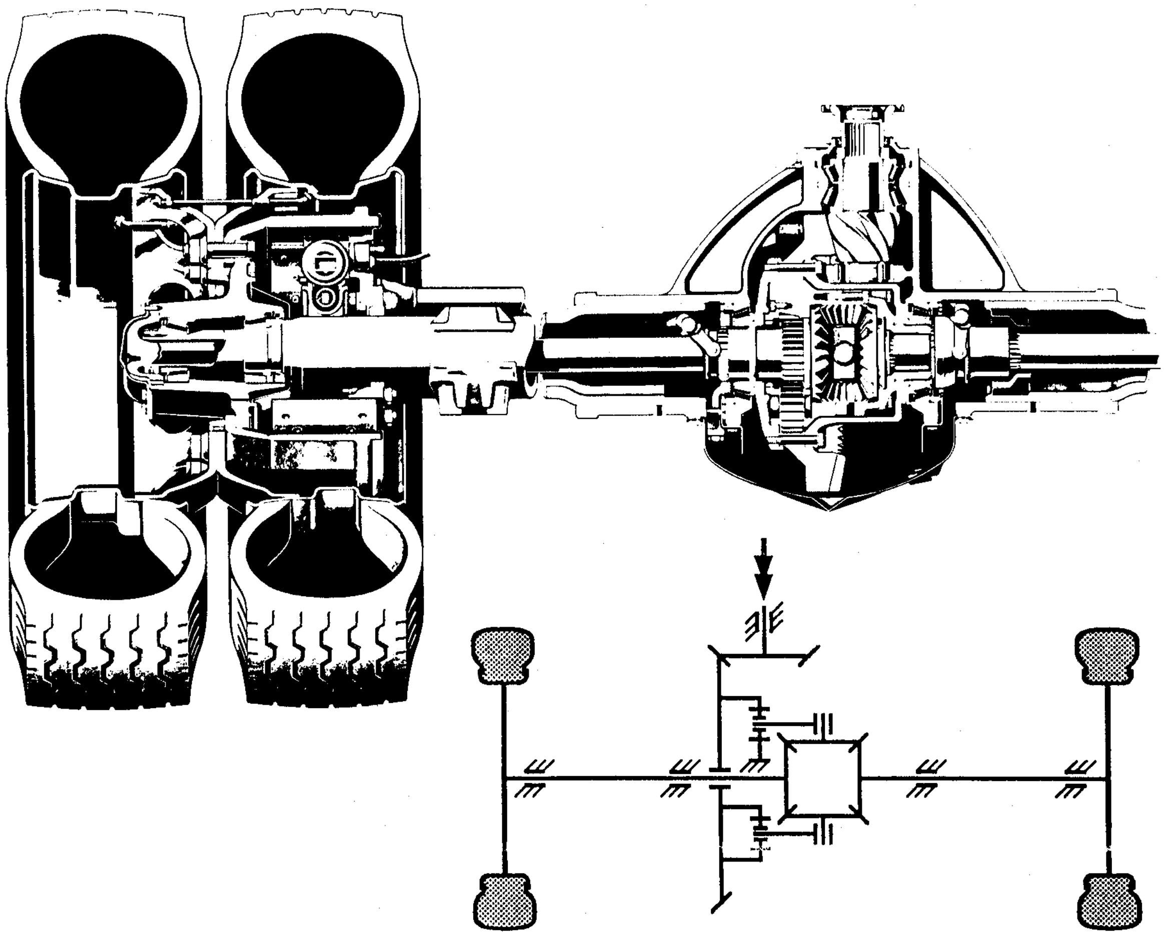


Figure 12.24. Mercedes Benz HL5/2Z-10. Two-speed axle. Schematic diagram top: Transmission engaged. Schematic diagram bottom: Transmission not engaged. Centre gearbox turned through 90°

Because of the large axial and radial forces of the hypoid bevel gears, the pinion *1* has bearings on both sides. Tapered roller bearings *2* are used in an O configuration with the largest possible angle of contingence. A cylindrical roller bearing *3* is used as supporting bearing. The adjustable thrust piece with fulcrum pad *4* prevents the crown gear *5* being deformed excessively under load. This enables the full capacity of the bevel gears to be exploited.

2/ Two-Speed Axles

The type of axle shown in Figure 12.24 is encountered in commercial vehicles with one driven axle a requirement for large final ratios, especially buses. The axle is broadly similar to the single-ratio version. For the larger ratio, a spur gear stage or a planetary gear-set in a crown gear is engaged as well. This enables the final ratio to be increased to approximately $i_E = 9.0$. Some of the disadvantages of the single-ratio axle can be offset with this design. The high inertia torque of the power train creates shiftability problems where ratio changes are too great.

3/ Pinion Axle (Offset Axle)

In contrast to the axle designs described above, the driving torque is increased not only in the centre gearbox but also in the wheel drives (Figure 12.25).

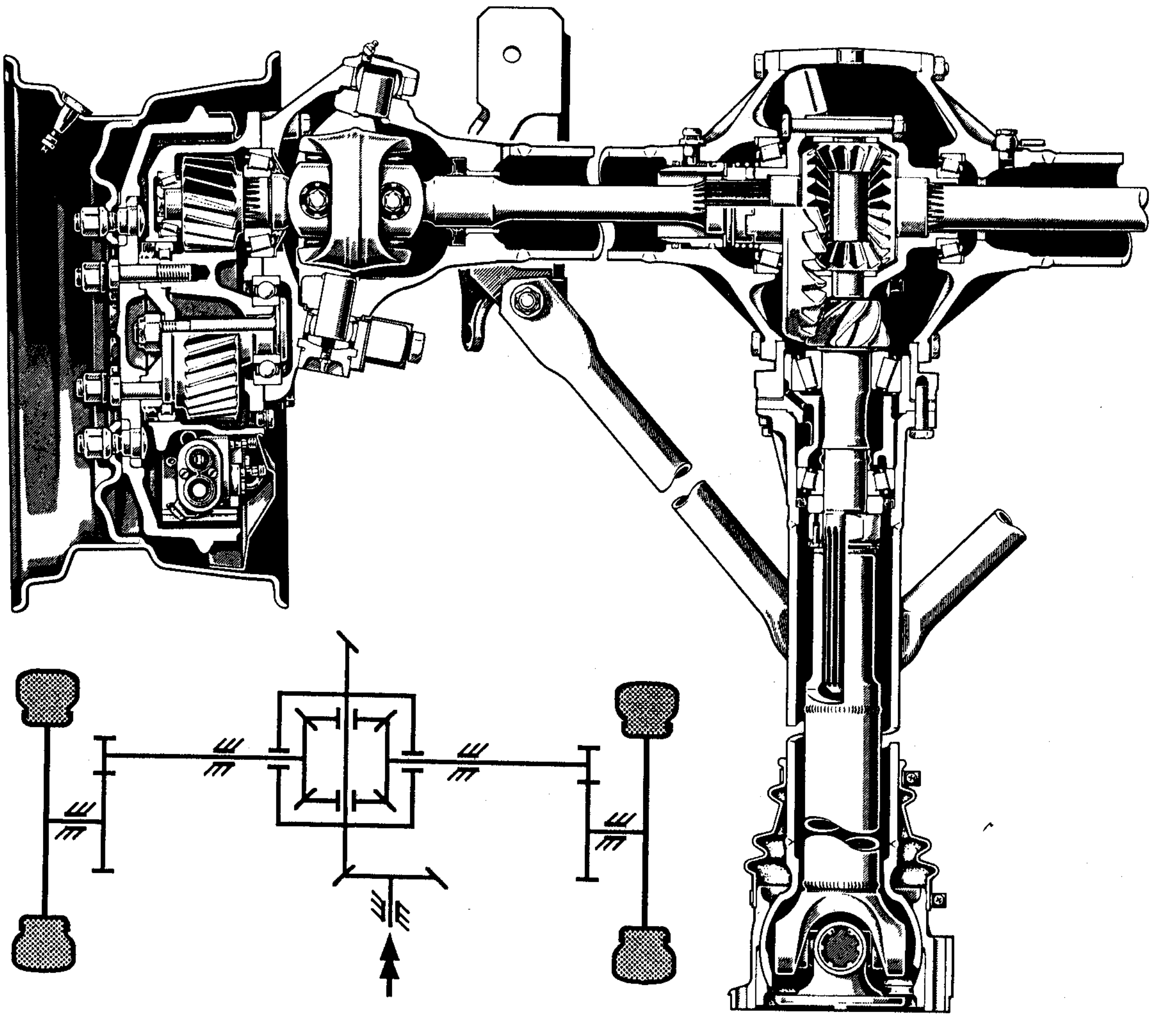


Figure 12.25. Pinion axle (offset axle) AU 2/2S-2.6 of the Unimog 407 as driven steering axle. Centre gearbox shown turned through 90°

The centre gearbox and the axle shafts can be smaller if the ratio is divided up. This gives adequate ground clearance even with high outputs. The spur gear drive is in most cases located inside the brakes. This means the brake drums have to be very large.

If the output rating were further increased, it would become difficult to transmit the relatively large forces with the single meshing of the spur gear drive. The gearwheel bearings would also need to be correspondingly sophisticated. Figure 12.25 shows the pinion axle (offset axle) of the Mercedes-Benz Unimog as a typical example of this axle design. The helical cut wheel drive limits the maximum ratio possible in the wheel hub $i_{N,\,max} \approx 2.5$.

Internal gearing provides improved transmission of large forces, with the disadvantage of less ground clearance. See also Figure 6.56c.

4/ Outer Planetary Axle

In the outer planetary axle the torque also only arises at the wheel, where it is converted directly into traction. This has a number of advantages. The centre gearbox can be kept relatively small, as with the pinion axle (offset axle), allowing good ground clearance. The planetary gear is located outside the wheel brake, so there are no problems when designing the brake. The power to be transmitted is distributed in the planetary gear to several gear meshes (up to 5 planetary gearwheels). This is what makes this type of gear unit compact. Low rolling and running speeds at the tooth flanks lead to a high level of efficiency. Floating mounting is possible due to the balance of static forces within the planetary gear. The outer planetary axle can easily provide virtually any desired ratio. Figure 12.26 shows a ZF outer planetary axle.

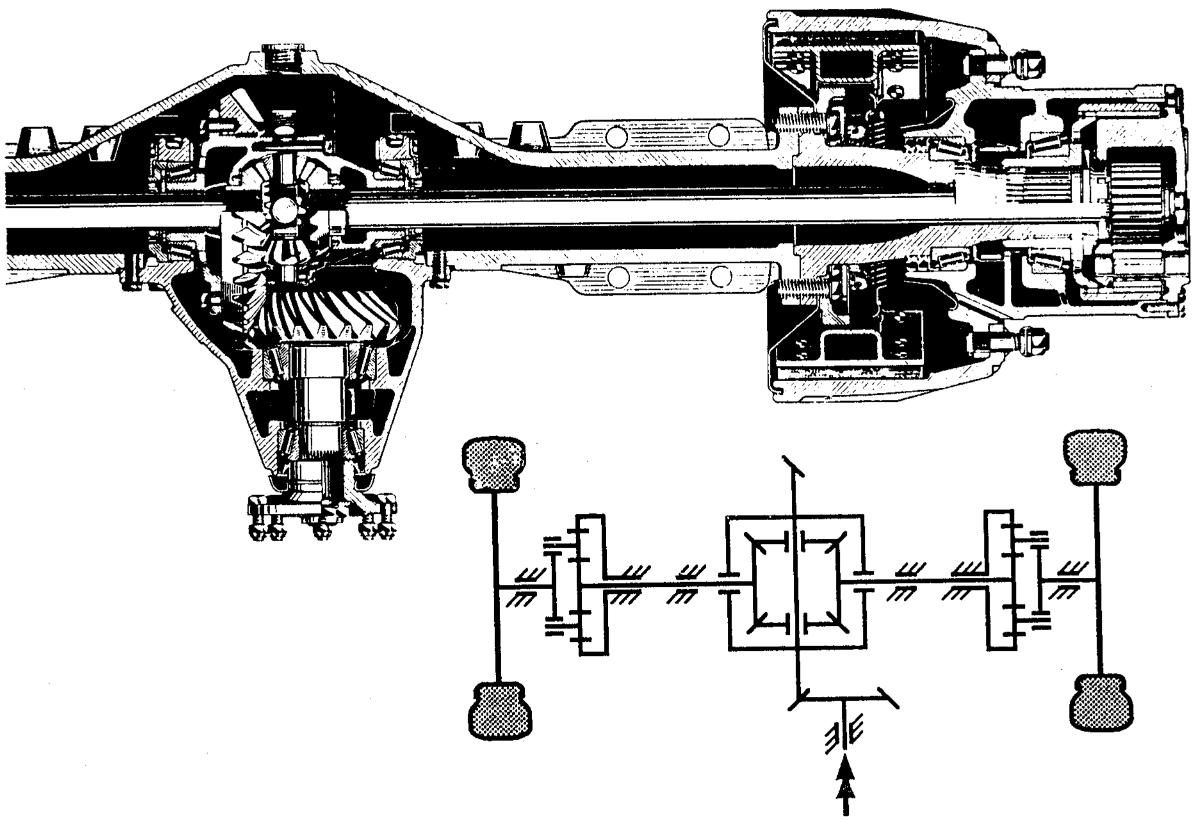


Figure 12.26. Outer planetary axle with spur gear planetary drive, ZF. Centre gearbox shown turned through 90°

12.6 Differential Gears, Locking Differentials

As discussed in Section 6.8, differential mechanisms can be divided into transfer boxes (longitudinal splitting of speed and torque) and differential gearboxes (transverse splitting). Some examples of differential gears and locking alternatives are examined below, selected from numerous production designs. Examples of form-locking mechanisms (dog locks) are given in Section 12.7.

The following differential gears and limited-slip differentials are examined below:

1/ bevel gear differential, Figure 12.27,
2/ self-locking differential with multi-disc clutches, Figure 12.28,
3/ self-locking differential with worm gears (TORSEN), Figure 12.29,
4/ self-locking differential with fluid clutch, Figure 12.30,
5/ slip-controlled self-locking differential, Figure 12.31,
6/ cam-locking differential,
7/ automatic limited-slip differential.

1/ Bevel Gear Differential

Rear axle gearboxes, such as the one shown in Figure 12.27, are used in practically all vehicles with longitudinal engine and rear-wheel drive (standard drive).

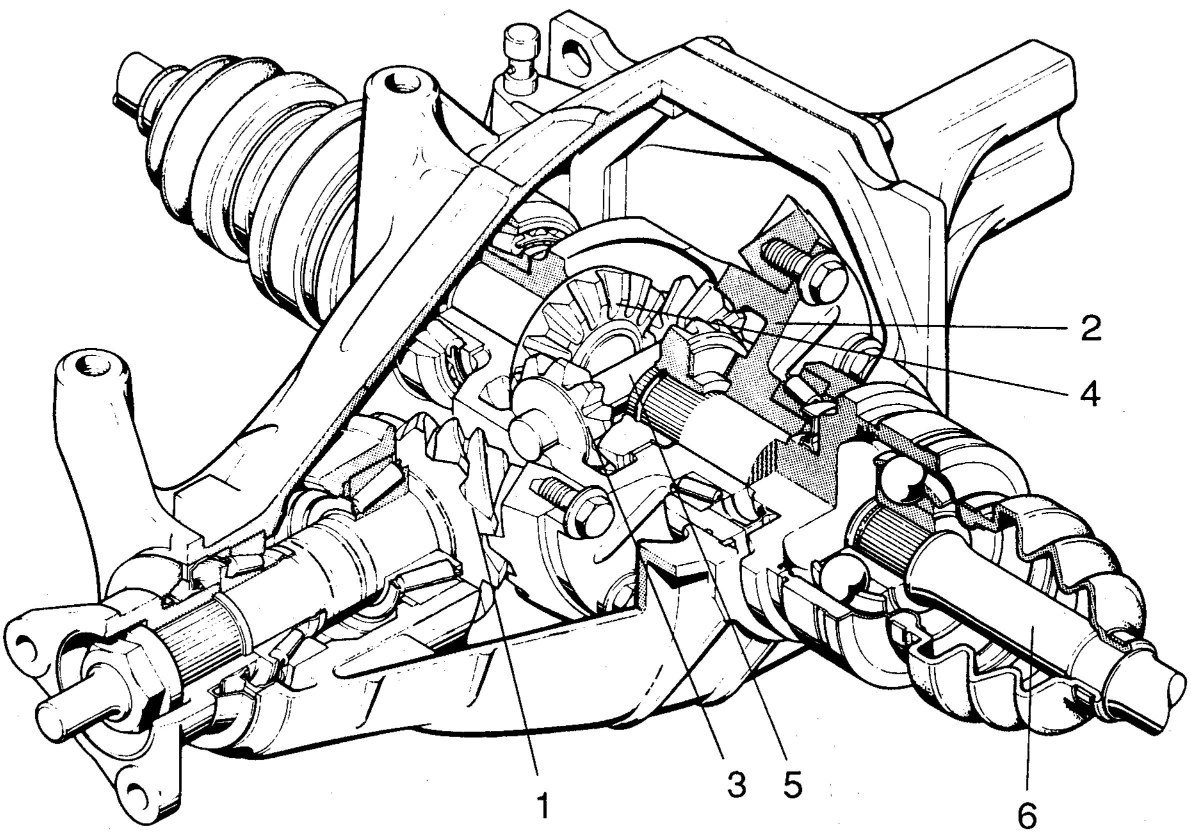


Figure 12.27. Non-locking Opel rear axle differential. *1* Bevel gear final drive; *2* Differential cage; *3* Differential bevel gears; *4* Axle bevel gears; *5* Differential pin; *6* Axle shafts

The torque applied through the bevel gear final drive *1* is transmitted through the differential cage *2* and the differential pin *5* to the differential bevel gears *3* and from there to the axle bevel gears *4*, which are torsionally locked to axle shafts *6*.

When driving in a straight line, the differential cage *2* the axle bevel gears *4*, the axle shafts *6* and the differential bevel gears *3* rotate inside the cage as a block. There is no relative movement between the differential pin *5* and the differential bevel gears resting on it. One axle shaft has to turn faster than the other when cornering. Axle bevel gears and differential bevel gears seesaw on each other. There can be speed compensation between the wheels.

2/ Self-Locking Differential with Multi-Disc Clutches

The locking effect of a self-locking differential with multi-disc clutch relies on the torque-dependent internal friction generated in two multi-disc clutches mounted symmetrically in the differential cage. The self-locking action results from a combination of the load-dependency and spring loading of the multi-disc clutches. The load-dependent locking effect (Figure 12.28, top) relies on the input torque T_1 applied to the differential cage *1* being transmitted via the differential pin *2* to two pressure rings *3* in the differential cage *1* that are torsionally locked but slide axially. Under load, independent locking forces arise at the surfaces of the prism-shaped recesses *8* in the pressure rings (see detail in 12.28), pressing the clutch discs together. The outer discs *4* are torsionally locked to the differential cage *1*, and the inner discs *5* are torsionally locked to the axle bevel gearwheels *6*.

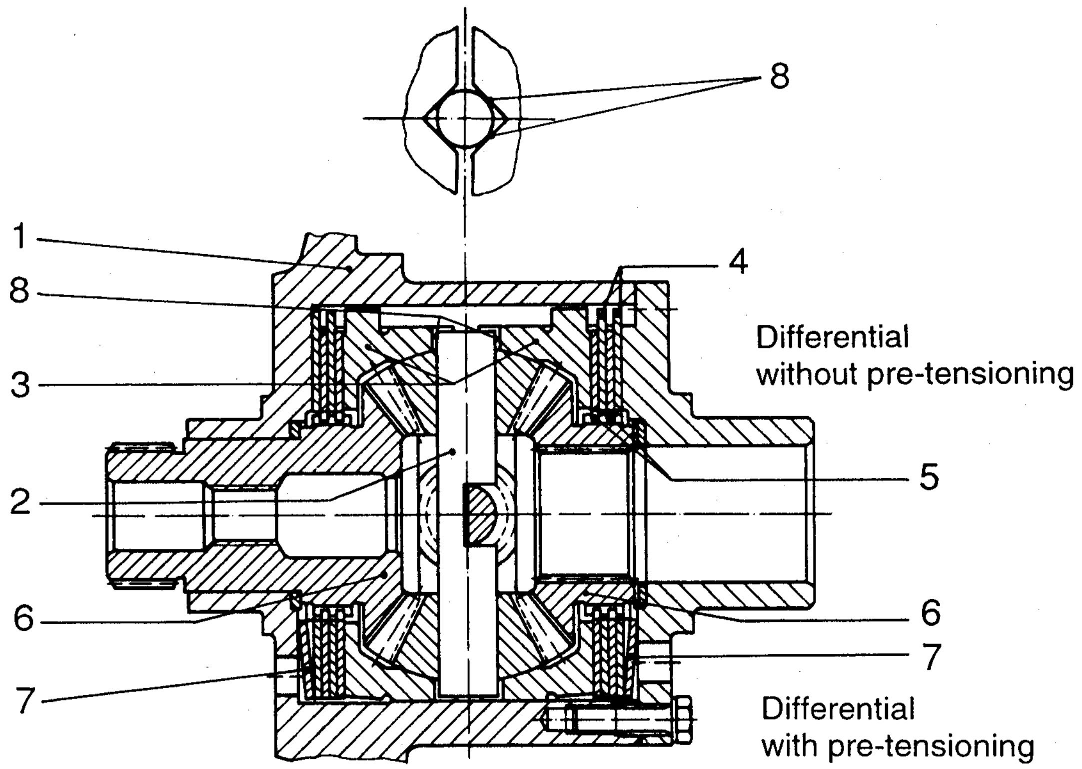


Figure 12.28. Limited-slip differential with pre-stressed multi-disc clutches, Lok-O-Matic. Top section: differential without pre-tensioning. Bottom section: differential with pre-tensioning. *1* Differential cage; *2* Differential pin; *3* Pressure rings; *4* Outer discs; *5* Inner discs; *6* Axle bevel gearwheels; *7* Cup springs; *8* Recesses

The frictional contact between the discs thus opposes the various axle shaft speeds (for example when a wheel spins) with a precisely defined force. This effect increases as the input torque increases. Since the locking forces are proportional to the torque that can be transmitted, the locking effect adapts to the changing engine torque and to the torque increase in the various gear steps, but the interlock value does not (see also Section 6.10.3 "The Interlock Value").

The cup springs *7* (shown in the bottom half of Figure 12.28) that can be fitted to preload the multi-disc clutch, create a constant interlocking effect that is independent of the torque that can be transmitted, but sometimes makes noticeable rattling noises. This makes the system capable of locking even on extremely unfavourable surfaces, for example black. There is nevertheless still the disadvantage that a differential of this type always has a slip-dependent basic locking torque. This can be undesirable when parking and when cornering without slip.

The torque-dependent contact pressure can also be used as the sole means of applying pressure through the gear spreading forces of the bevel gear differential. These contact pressures are less than those achievable with pressure rings by about a factor of 3. A further drawback that should be borne in mind is that during the self-locking or compensating process the tooth geometry of the bevel gears changes adversely, because the friction clutches that have to be applied must not be backlash-free.

Unsymmetrical self-locking differentials of this type are also used in transfer boxes with just one multi-disc clutch.

3/ Self-Locking Differential with Worm Gearwheels (TORSEN)

The design and function of the self-locking differential with worm gears, known as the TORSEN transfer differential, is described with reference to Figure 12.29. TORSEN stands for "torque sensing".

The driving crown gear *1* is linked to the differential housing *2*. The six worm gearwheels *3* run tangentially on pins in the housing, moving freely. They are braced against the housing at the face. Spur teeth *4* are torsionally fixed on each front face of the worm gearwheels. The spur teeth of neighbouring worm gearwheels are engaged, and three worm gearwheels of one half of the mechanism mesh with one of two worms *5*. The worms are radially centred by the symmetrical arrangement of the worm gearwheels relative to them. The worms have axial support available in the housing or at their common locating face. The worms are hollow, and carry gearing inside, into which the output shafts *6* are inserted.

When travelling straight ahead and with good grip on both wheels, the differential rotates as a block. The input torque reacts against the spur gear toothing of the worm gears. When cornering or when there is a high tendency for a wheel to spin, roll control can be provided through the spur gearing. But since all the components are rotating at different relative speeds, and are also under load, there are losses at all friction points. The losses in the worm gear pairs arise from the effect whereby the friction in the worm gearing increases the worm gearwheel circumferential forces on one worm, and decreases them on the other. This creates an uneven torque distribution to the two output sides, which is further amplified by the friction in the contact faces of the worms. The total of all torque losses is equal to the locking torque. Roll control is therefore not possible without load or without simultaneously building up the design-dependent basic interlock value.

The basic interlock value of this mechanism is 20%. In practice the interlock value is higher than 20% due to the friction between various components and the differential housing. The teeth of the worms and worm gearwheels are also deliberately mismatched to achieve interlock values greater than 60%.

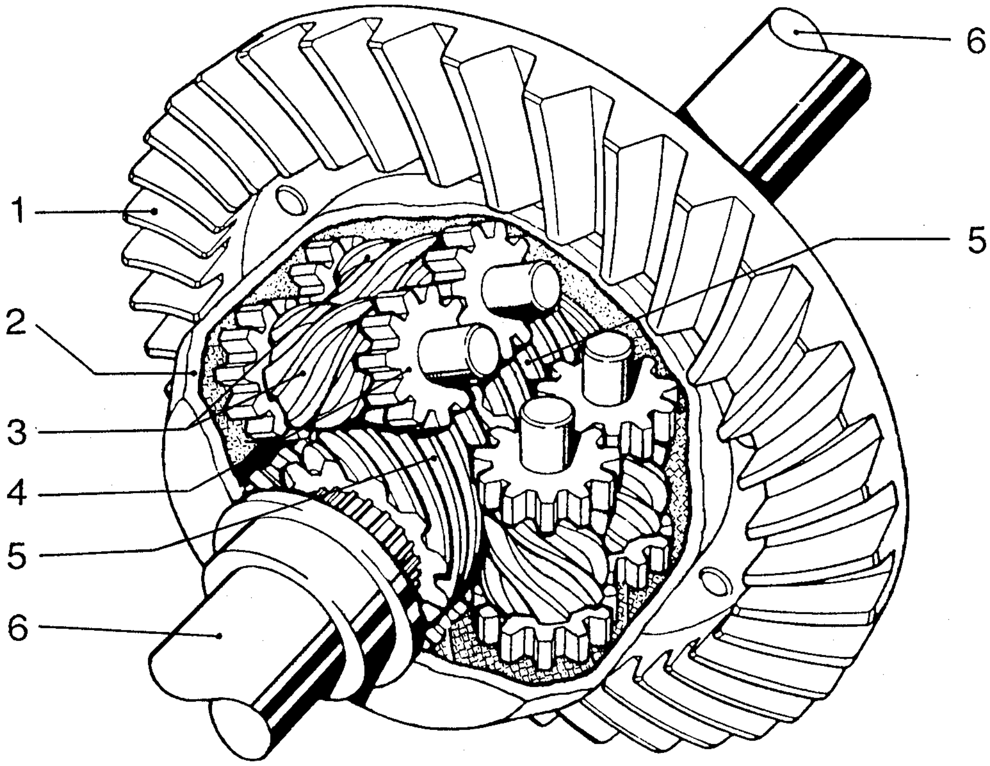


Figure 12.29. TORSEN, self-locking differential with worm gearwheels.
1 Input crown gear ; *2* Differential housing ; *3* Worm gearwheels; *4* Spur gearing;
5 Worms; *6* Output shafts

To build up the locking effect, this mechanism needs both a speed differential between the output shafts and also a positive input torque. The locking effect is momentarily suspended when the load is removed. Vehicles with this differential are therefore fully suitable for use with antilock braking systems.

4/ Self-Locking Differential with Fluid Clutch

A visco self-locking differential comprises of the components of a conventional differential, and also those of a fluid friction clutch. This is a completely sealed, annular cylindrical component, and has inner and outer discs and the fluid power medium, a high-viscosity silicone oil. In the example shown in Figure 12.30 with spur gear differential, the outer discs *5* are torsionally fixed in the outer disc carrier *6*, which is linked to the planet carrier *2* or the right-hand axle shaft inserted in it. The planetary gears *3* rotate on pins in the planet carrier. The inner discs *7* run torsionally fixed in the inner disc carrier *8*, which is connected to the sun gear *1* or to the left-hand axle shaft inserted in it. Drive is through the housing *9* with integral ring gear *4*. Other production versions also have the clutch between the input shaft and an output shaft.

With a slipping drive wheel or when cornering, relative movement arises between the two axle shafts, and so between the sun gear shaft and the planetary gear carrier. The planetary gears *3* seesaw on each other, in the ring gear and on the sun gear. Since this relative movement also arises between the inner and outer discs, the fluid power medium of the clutch is subject to shearing stress, giving rise to viscous resistance to the cause – the speed difference. The level of this moment of resistances depends on the speed differential. It acts to brake the wheel that is rotating faster, thus increasing the torque on the wheel with the better grip.

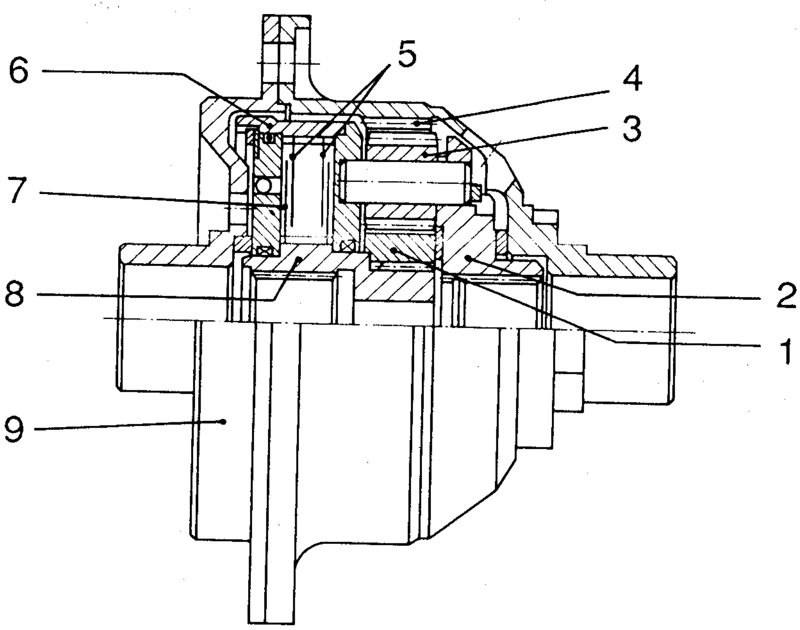


Figure 12.30. Self-locking spur gear differential with visco-clutch.
1 Sun gear; *2* Planet carrier; *3* Planetary gears; *4* Ring gear; *5* Outer discs; *6* Outer disc carrier; *7* Inner discs; *8* Inner disc carrier; *9* Housing

The viscous braking torque generated depends exclusively on the speed difference between the output shafts. The heat generated under protracted stress raises the pressure inside the clutch. This pressure presses contiguous inner and outer clutch discs together with greater force, displacing the fluid power medium from the space between them. The resultant friction contact transforms the clutch into a rigid component, and the increased interlock value of 100% persists even when the speed difference declines. This process is called the "hump effect" because it represents a hump in the speed/braking torque curve of a visco-clutch.

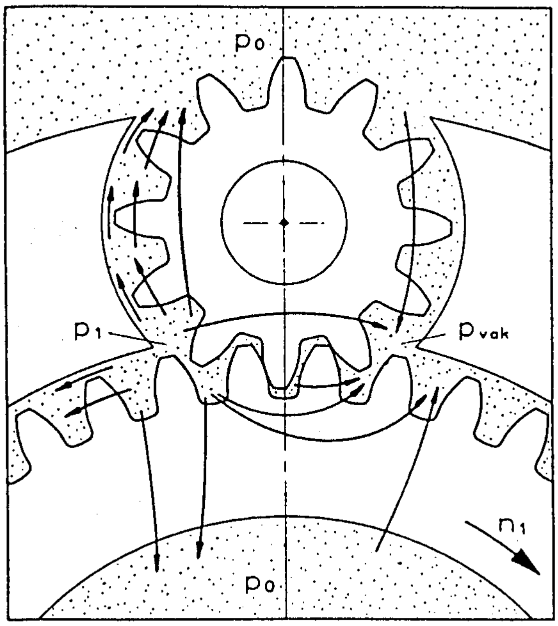
A differential of this type is controlled entirely by the differential speed. As a consequence, undesirable distortions can arise in the drive train on tight bends, even without load, depending on the clutch design. On the other hand, differential brakes of this type bite very gently, yet they reach very high interlock values of up to 100% when required.

5/ Slip-Dependent Self-Locking Differential with Hydrostatic Locking Forces

Hydrostatic self-locking planetary gears are used particularly for self-locking interaxle differentials in all-wheel drive vehicles, but they can also be used as an inter-wheel differential lock in front axle differentials (Figure12.31). This differential is based on a three-shaft planetary spur gear.

Contrary to usual practice, in this case the planetary gears are truly encased in the planetary gear carrier, so the planetary gear carrier is designed like a gear pump housing [12.9 to 12.11]. This housing encloses the sun and planet gearing both axially and radially. If a speed difference arises between the planetary gear carrier and the sun gear shaft, the planetary wheels rotate relative to the planet carrier, feeding oil from the sump or the shell into the gearbox.

Every meshing contact of the sun/planet gears effectively constitutes a gear pump. Compressive forces are generated which act against their cause, the speed difference, thus impeding free compensating movement. This relieves the faster-running output shaft of the locking torque arising from the compressive forces, which is applied to the slower-running output shaft.



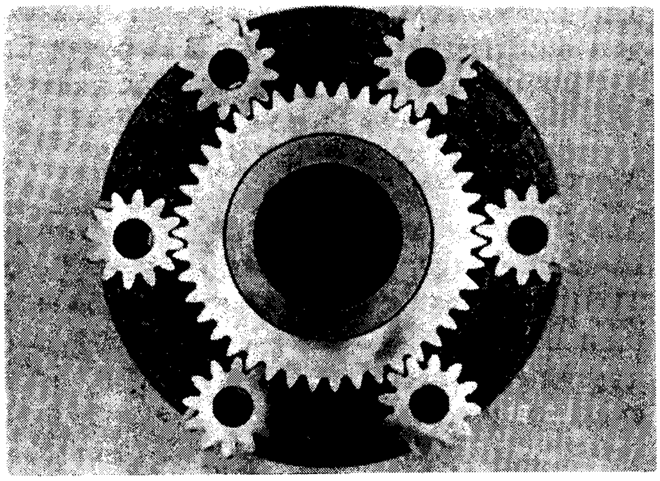


Figure 12.31. On the left: flow paths in the hydrostatic self-locking planetary gear. On the right: view of sun gear and planetary gears of a hydrostatic self-locking planetary gear with shell. Shown without ring gear

Gear units of this type have advantages over limited-slip differentials with visco-clutch in terms of bulk and of production cost and resources required.

6/ Cam Self-Locking Differential

Cam self-locking differentials are still sometimes found in older vehicles and racing cars. Cam plates with fulcrum pads are used instead of bevel gears or spur gears to compensate different wheel speeds when cornering. The internal friction of the fulcrum pads in their tracks is such that self-locking occurs under load above a particular speed difference. This relies on the fulcrum pads jamming in the tracks. Although this gives interlock values of up to 100%, the locking effect engages very harshly. This type of design cannot therefore be regarded as suitable for normal modern passenger cars.

7/ Automatic Locking Differential

This form of traction aid can no longer be classified as an automatic self-locking differential. The structural design is similar to the differential in Figure 12.28 – with two disc sets mounted symmetrically to the bevel gear differential (see also Figure 12.36 "Mercedes-Benz 4MATIC transfer box"). But the load-dependent contact pressure of these disc sets is generated not by pressure rings, but only by the spreading forces of the differential gearing arising under load when there is a speed difference between the output shafts. This results in a purely load-dependent maximum basic interlock value of 35%. Signals detected by wheel-speed and input-speed sensors are also processed by an electronic control unit. If the control unit detects excessive slip, oil pressure is applied to the disc sets to increase the degree of locking to 100%. This means that in extreme situations the traction potential of a rigid axle can be used, without restricting the smoothness of the compensating movement. Only when the input torque exceeds the maximum traction that can be transmitted by both wheels do both wheels spin together. This limit does however have to be "felt" with the accelerator pedal. To avoid difficulties when braking, locking is inhibited or engagement prevented when the service brake is operated.

12.7 Four-Wheel Drives, Transfer Gearboxes

This section considers passenger-car all-wheel drives as a particular case of passenger-car and commercial-vehicle all-wheel power trains. Numerous different four-wheel drives are available on the market. Various examples are used below to illustrate selected parts of four-wheel drive units.

Mercedes-Benz, G-Model

The first production unit considered is an off-road four-wheel drive unit. Figure 12.32 shows the transfer box of the Mercedes-Benz G model with permanent four-wheel drive. The manual or automatic gearbox has no specific four-wheel drive features. In this power train the engine torque is transmitted via the gearbox and a short input shaft to the transfer box. From there, the torque is distributed equally to the front axle and the rear axle by propeller shafts.

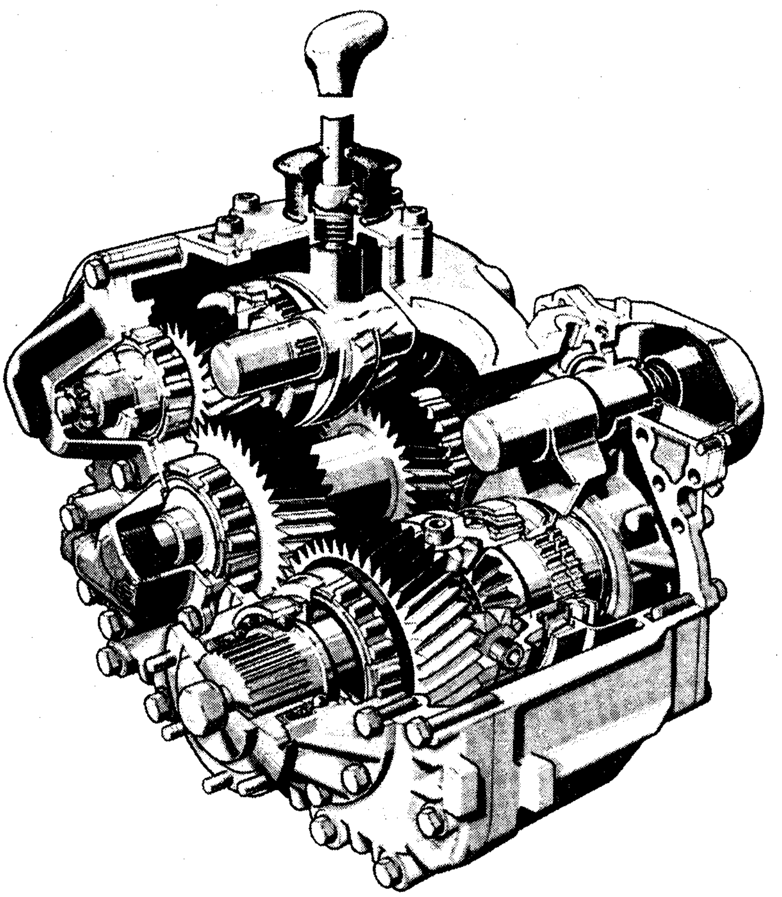


Figure 12.32. Mercedes-Benz G model VG 150 transfer box (permanent four-wheel drive)

The VG 150 transfer box shown in Figure 12.32 is a three-shaft box with a bevel gear differential for interaxle compensation. Torque distribution between front and rear axle is thus fixed at 50% to 50% in the unlocked state. With the positive differential lock, the interaxle differential can be locked 100%. The all-synchromesh transfer box houses both the differential and the switchover from the on-road ratio ($i = 1.05$) to the off-road ratio ($i = 2.16$). The front and rear axle interwheel differentials are likewise equipped with positive locks. When all three locks are engaged, off-road traction is optimised.

VW Syncro

The possibility of speed compensation between the front and rear axle when the locks are not engaged makes this four-wheel drive unit fully ABS-compatible. When a differential lock is activated, the antilock braking system control is automatically switched off. The antilock braking system control can also be switched off manually when driving off-road on loose surfaces (exploiting the braking wedge in front of the wheels on loose surfaces).

Figure 12.33 shows the VW Golf Syncro 5-speed front-wheel drive bevel gear selector gearbox with cardan shaft link, as an example of a *clutch-controlled four-wheel drive unit* [12.12]. This four-wheel drive unit is based on a vehicle with front-wheel drive and front mounted transverse engine with side flange-mounted gearbox. The associated rear axle gear unit with visco-clutch, free-wheel and differential can be seen in Figure 12.34.

The front bevel gear is flange-mounted on the right-hand side of the selector gearbox. The drive is direct from the differential housing of the selector gearbox via an additional slip-on gear mechanism. The output to the right-hand front-wheel as viewed in the direction of travel runs through the hollow shaft on which the bevel drive pinion sits. The bevel gear itself is helical and has a ratio $i_{front} = 0.952$ ($z_1 = 21$ to $z_2 = 20$).

The bevel drive of the rear axle gear unit (Figure 12.34) is of similar design to that of the front bevel gear. The ratio is $i_{rear} = 1.05$. The power train thus has a ratio of $i = 1.0$. A sprag unit is fitted between the differential and the bevel gear to disengage the wheels of the rear axle during braking, guaranteeing the vehicle's ABS-compatibility and directional control during braking. The sprag unit is locked up by a clutch when reversing. The positive lock-up clutch is activated when reverse gear is engaged.

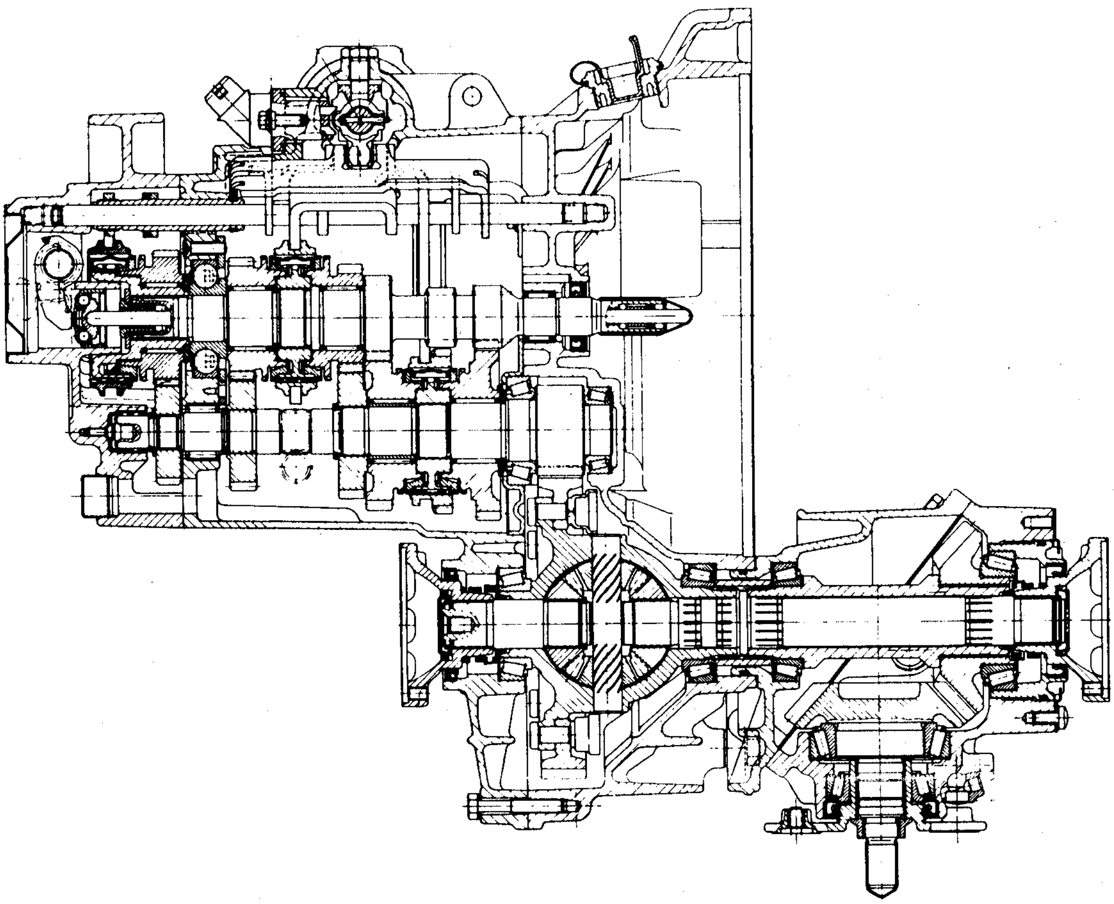


Figure 12.33. VW Golf Syncro 5-speed manual gearbox with bevel gear and cardan shaft link to the rear axle

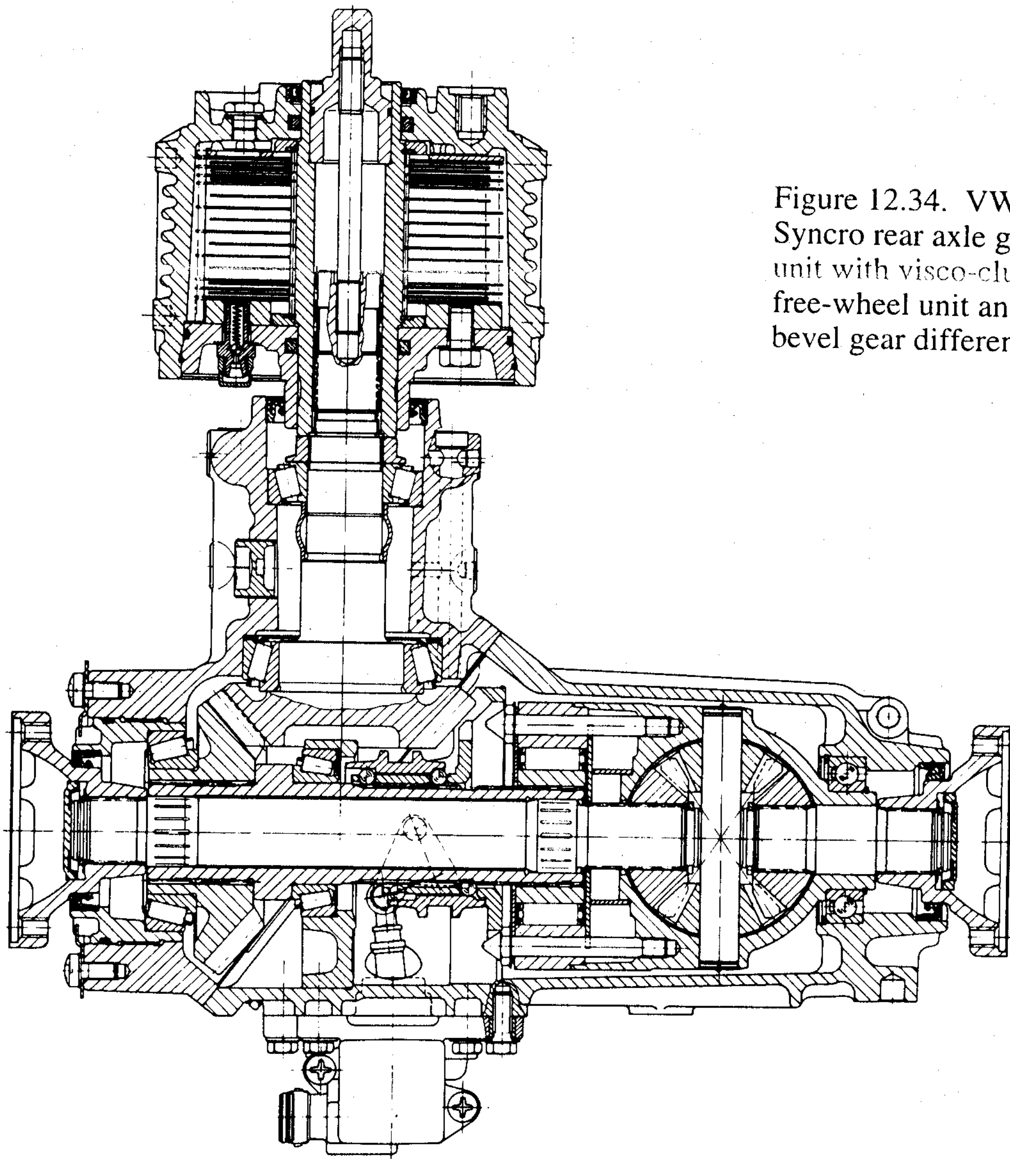


Figure 12.34. VW Golf Syncro rear axle gear unit with visco-clutch, free-wheel unit and bevel gear differential

Audi Quattro

Figure 12.35 shows the 5-speed B-80 transmission for the 1987 Audi 80, 90 and 100 models as adapted for permanent four-wheel drive with TORSEN differential [12.13]. It is developed from the Quattro transmission system first used by Audi in 1980 (flange-mounted rear-wheel drive, front-wheel drive with driving shaft running inside a hollow shaft, bevel gear differential with dog locking).

In the B80 5-speed transmission the output shaft is a hollow shaft, driving the planet carrier/spider of the TORSEN transfer differentials. The front axle is driven starting from the sun gear with a driving shaft running in the hollow shaft to the front axle drive, which is integral to the gearbox housing as in pure front-wheel drive. The rear axle is driven by the other sun gear of the TORSEN transfer differential via a two-part input shaft with constant-velocity universal joints. The rear axle gearbox contains a bevel gear differential with manually controlled 100% lock.

The locking function in the interaxle transfer differential is provided by the TORSEN transfer differential. Input torque is distributed between the front and rear axle under load by the locking effect depending on the torque applied. There is no locking effect without load, which ensures complete ABS-compatibility. The operation of the TORSEN transfer differentials is discussed in detail in Section 6.10 "Differential Gears, Differential Locks and Locking Differentials" and Section 12.6 "Differential Gears, Locking Differentials".

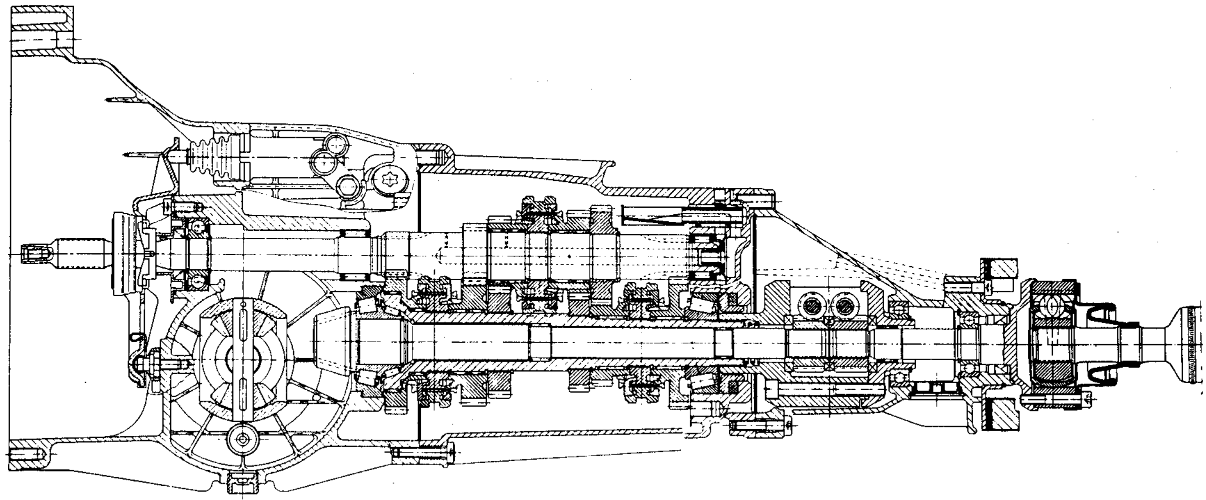


Figure 12.35. Audi 80/90/100 Quattro transmission with TORSEN transfer differential

Mercedes-Benz 4MATIC

The 4MATIC four-wheel drive unit was developed from the Mercedes Benz W 124 range standard [12.14 and 12.15]. The input torque is transmitted to the transfer box via a manual or automatic gearbox (Figure 12.36). The transfer box contains a spur gear interaxle differential, a multi-disc clutch for engaging the front axle drive, a multi-disc clutch to lock the spur gear interaxle differentials, and the output to the front axle. With the spur gear interaxle differential the torque is split 35% to the front axle and 65% to the rear axle when the multi-disc clutch is engaged.

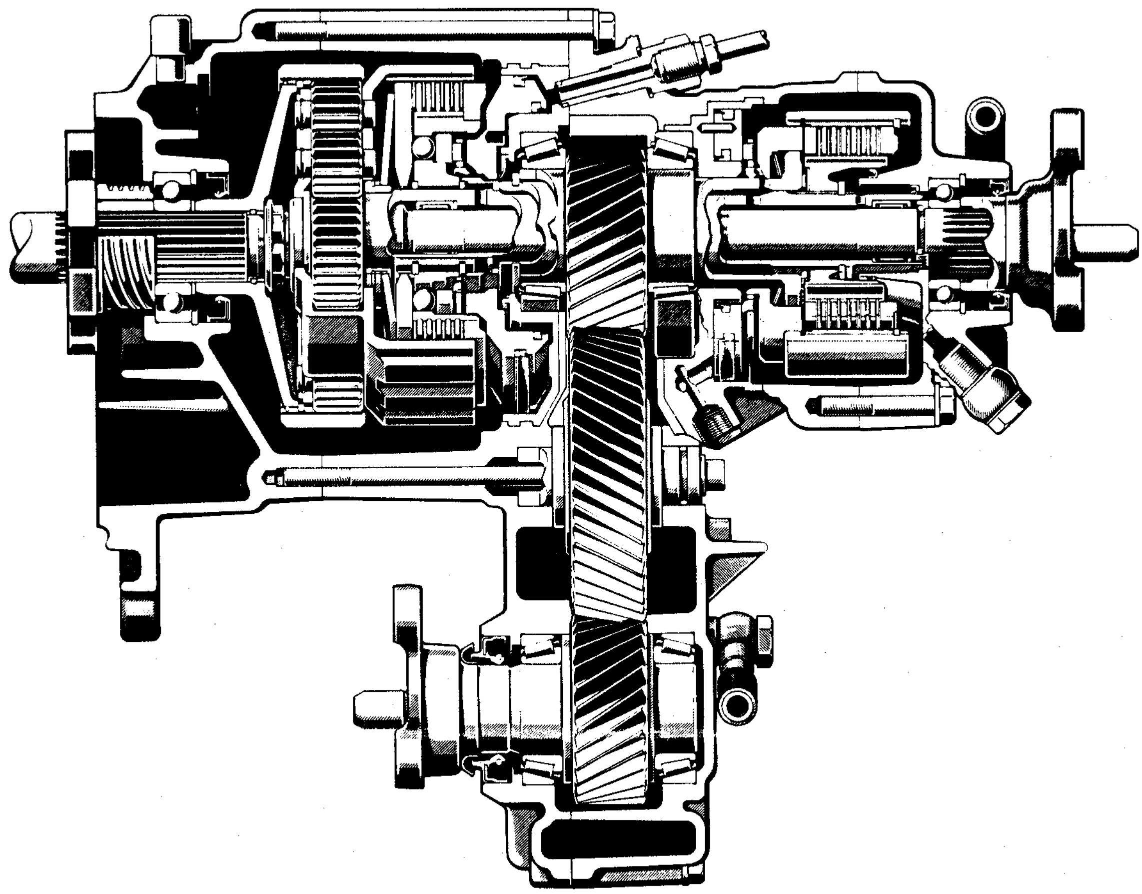


Figure 12.36. Mercedes Benz W 124 4MATIC VG 30 transfer box

The drive to the front axle is via two spur gears and one propeller shaft. The final drive and the differential are integral to the engine sump, but have a separate oil circuit. The input shaft to the right-hand front-wheel runs through the engine oil sump. The rear axle is driven directly from the transfer box via propeller shafts. The automatic limited-slip differential is fitted to the rear axle. The ASD is described in Section 12.6 "Differential Gears, Locking Differentials".

The following drive modes are possible, depending on the road surface and the driving conditions:

- pure rear-wheel drive,
- front-wheel drive engaged, "permanent" four-wheel drive with interaxle differential (torque split 35% front axle and 65% rear axle),
- interaxle differential locked,
- interwheel differential also locked.

The various drive modes are selected automatically in the sequence indicated, depending on the velocity, acceleration, slip, steer angle and brake application. Engaged front-wheel drive is used specifically to improve driveability, whereas the interaxle differential lock and the rear inter-wheel differential lock merely increase traction. Full ABS-compatibility is achieved by all locks being released when the brakes are applied.

Further features of the transfer box shown are the gear pump in the output to the front axle for supplying the gearbox with lubricating oil, and the cup spring loaded multi-disc clutch for front-wheel drive, which is opened by oil pressure. This preserves rear-wheel drive capability if the multi-disc clutch control fails.

BMW 325iX

The BMW 325iX is another example of a permanent (differential-controlled) four-wheel drive car developed from a vehicle with standard drive [12.16]. This is achieved by flange-mounting the ZF A 95 transfer box to the manual or automatic gearbox (see Figure 12.37). The transfer box contains the spur gear interaxle differential *1*, the visco-clutch *3* and a gear chain *2* for driving the front axle *4*.

The torque is split in a fixed ratio between the front and rear axle according to the ratio of the spur gear interaxle differentials, 37% to the front axle and 63% to the rear axle. When there are speed differences between the two axles, the compensating movement is retarded by the visco-clutch mounted between the sun gear (output to the front axle *4*) and the ring gear (output to the rear axle *5*). In order to prevent distortion, the visco-clutch has an interlock value of $S = 0.07$.

The front and rear axle gearboxes are driven by the transfer box via propeller shafts. The rear axle differential is retarded by a visco lock like the interaxle differential. The front axle gearbox is located to the left of the engine sump, to which it is bolted. The output shaft to the right-hand front-wheel runs through the engine sump. It runs in a block on the right-hand side of the engine sump. Both input shafts to the front-wheels are therefore of the same length.

Modifications to the ABS control unit and the low level of locking of the visco locks make this all-wheel drive unit ABS-compatible.

ZF HP 24A (All-Wheel) Automatic Transmission

The permanent four-wheel drive in the Audi V8 was combined with an automatic transmission for reasons of comfort and safety [12.17]. For this purpose a transfer box was rear-mounted on the modified ZF HP 24A automatic transmission (Figure 12.38). A planetary gear is used for speed compensation between front and rear axle. The ring gear is linked to the output of the automatic transmission. The rear axle is linked to the spider of the planetary gear by a spur gear stage. The front axle is driven by the ring gear via a spur gear stage with idler gear.

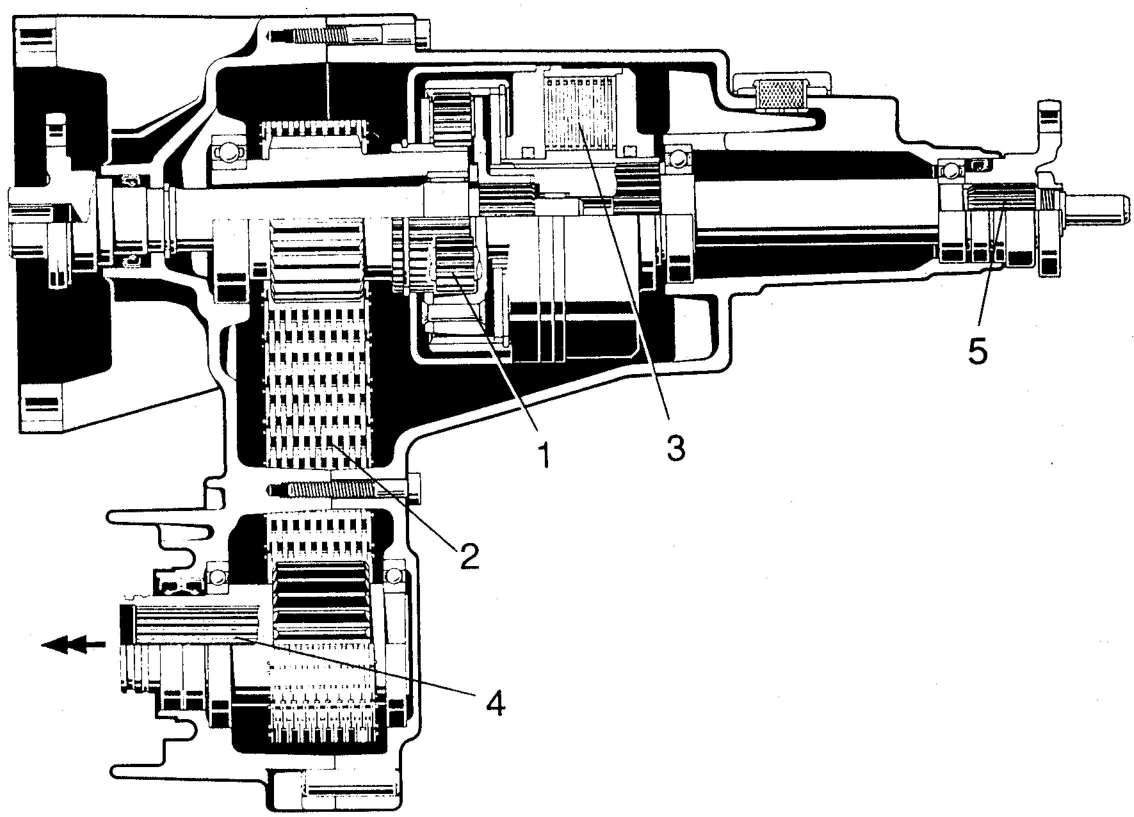


Figure 12.37. ZF A 95 transfer box (BMW 325iX). *1* Spur gear interaxle differential; *2* Gear chain; *3* Visco-clutch; *4* Output to front axle; *5* Output to rear axle

By selecting a fixed-axle gear ratio of $i_0 = 2.0$, the torque split between the axles is 50% to 50%. A hydraulically operated multi-disc lock is fitted between the spider and the sun gear. The transfer box is locked 100% when this lock is activated.

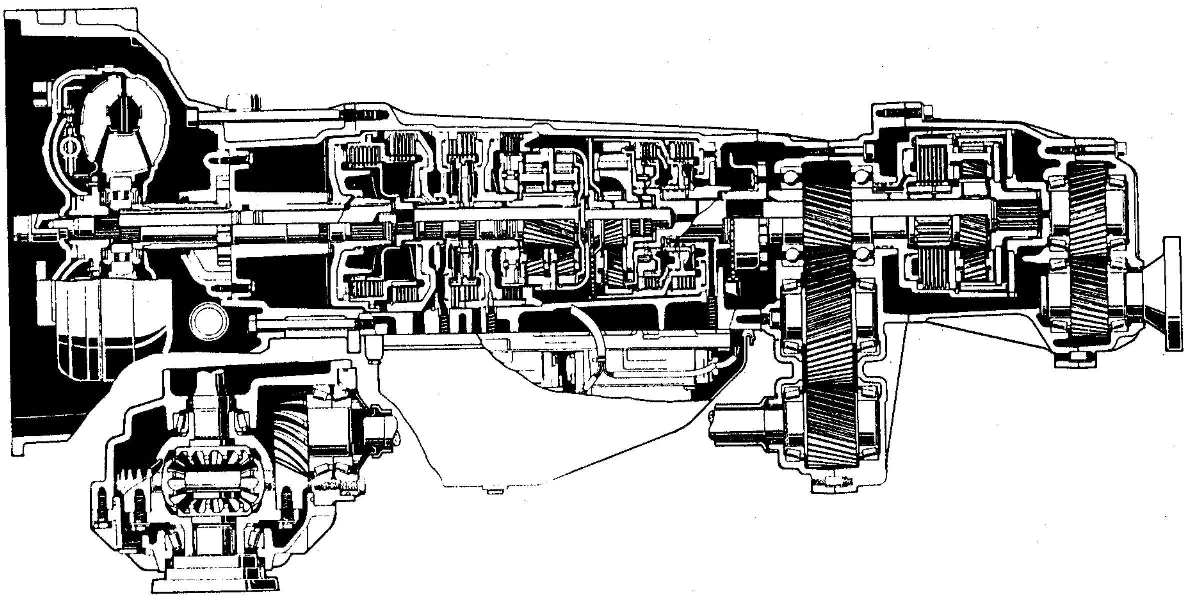


Figure 12.38. Automatic transmission (ZF HP 24A) with transfer box and front axle gearbox (Audi V8)

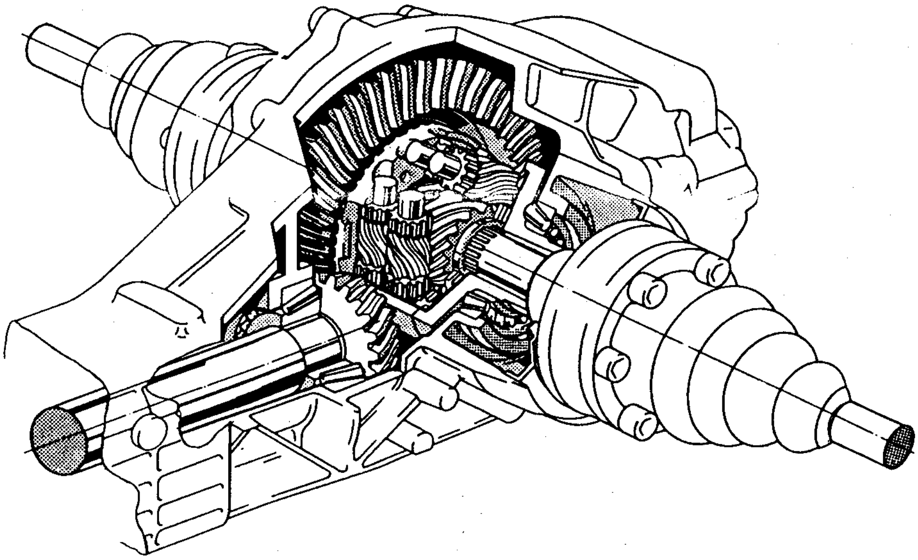


Figure 12.39. Rear axle gearbox with TORSEN differential (Audi V8)

In order not to impair driveability and safety, the lock is activated when the front and rear axles have different coefficients of friction. The lock is open when cornering and braking. A vehicle with a 100% lock can transmit approximately twice as much input torque to the road as a vehicle with a TORSEN transfer differential, such as in the Audi 80.

Using a TORSEN differential in the rear axle gear unit (Figure 12.39), which is also used as a central differential in the Audi 80 and Audi 100, in combination with the electronically controlled multi-disc lock in the transfer box, makes the four-wheel drive of the Audi V8 ABS-compatible.

Other 4-Wheel Drives

Other interesting examples of 4-wheel drive systems are the VW Bus Syncro [12.18], the Porsche Carrera 4 [12.19] and [12.20], and the Opel Vectra 4-wheel drive [12.21]. Other four-wheel drive components include the PDS (Porsche Dynamic Slip Control) [12.22] and [12.23], die PHA (Porsche Hydrodynamic Four-Wheel Clutch) [12.24] and the Viskomatic made by Steyr-Daimler-Puch-Fahrzeugtechnik [12.25].

13 Engine and Transmission Management, Electronics and Information Networking

Communication between various vehicle subsystems is essential

Electronics plays an increasingly important role in the control of the various drive components in modern vehicle power trains. Now that microprocessor-based mapping is used as standard to control ignition systems and fuel injection systems, increasing use is being made of "smart" clutch and transmission control units. The advantages of digital electronic controls over conventional mechanical, hydraulic or pneumatic controls are faster information processing and the ability to gather a larger number of relevant parameters. Different types of drive can be accommodated by simply changing the software, substantially reducing the development and testing effort required. Information networking enables shared use of sensors and interactive monitoring of control units.

Electronics is also used to further reduce the demands on the driver using servo units to reduce physical effort. Electronic systems are used for engine management, traction control, antilock braking systems, automatic master/gearshifting clutches, transmissions with automatic gearshift action, and fully automatic geared or continuously variable transmissions with adaptive shifting and control strategies. Please refer also to the sections on semi-automatic and fully automatic passenger car transmissions (Sections 6.6 and 12.2.1) and commercial vehicle transmissions (Sections 6.7 and 12.2.2).

13.1 Overview of Electronic Systems in Current Use

Electronic systems always seek to improve one or more of the following aspects [13.1]:

- economy,
- safety,
- environmental impact, and
- ease of use.

Figure 13.1 shows some existing examples of power-train electronic systems. The heart of electronic control systems is a microcomputer with memory for programs, vehicle and diagnostic data, with inputs for sensors, outputs for actuators, and communication interfaces for exchanging data with other control units or diagnostic equipment. Sensors serve to detect commands given by the driver, and to gather information on the current state of the engine, transmission and vehicle.

Both digital and analogue sensors are used. Digital sensors are used to detect switching movements (gearshift lever, kick-down switch, retarder switch, etc.) and monitor limit values (oil pressure, etc.). Analogue sensors are used to detect pedal positions, rotational speed (engine, transmission, road wheel), temperature and acceleration. Other information can be derived indirectly by calculation. For example the vehicle's lateral acceleration can be calculated from the velocity profile.

Electronically controlled function		To increase: Efficiency	Safety	Environmental compability	Comfort
Engine	Performance map injection/ignition	X		X	
	Catalytic converter, particulate filter			X	
	Road speed governor		X	X	X
Clutch, gearbox	Automatic clutch	X	X	X	X
	Automatic shifting	X	X	X	X
Axle, brakes	Retarder	X	X		
	Antilock braking system	X	X		
	Traction control	X	X		
	Differential lock	X	X		
Diagnosis		X	X	X	

Figure 13.1. Electronic systems in the power train, and their purpose

The resultant information can be used to make decisions about actions to be carried out by actuators, which act to convert electrical signals from the control unit into mechanical adjustments with the aid of electrical, hydraulic or pneumatic energy. Pneumatic power is usually available anyway in commercial vehicles. These actuators can also be either digital devices with two or more positions (e.g. valves), or actuators with a controlled adjustment range such as those used in automatic master and gearshifting clutches.

Using electronic control modules for control functions eliminates unreliable hydraulic, pneumatic and mechanical linkages, but also increases the amount of wiring. Since electrical connections are one of the main sources of faults in motor vehicles the challenge is to reduce the amount of wiring. One suitable way of achieving this aim is the use of *bus systems* [13.2] (Figure 13.2).

A bi-directional bus enables control modules to intercommunicate as a means of information networking. One example is adapting the engine speed to improve the shift action of automatic transmissions. Another example is the influence of the anti-lock braking and traction control system on the clutch, transmission and engine. Communication between control modules enables them to share sensors, reducing the number required.

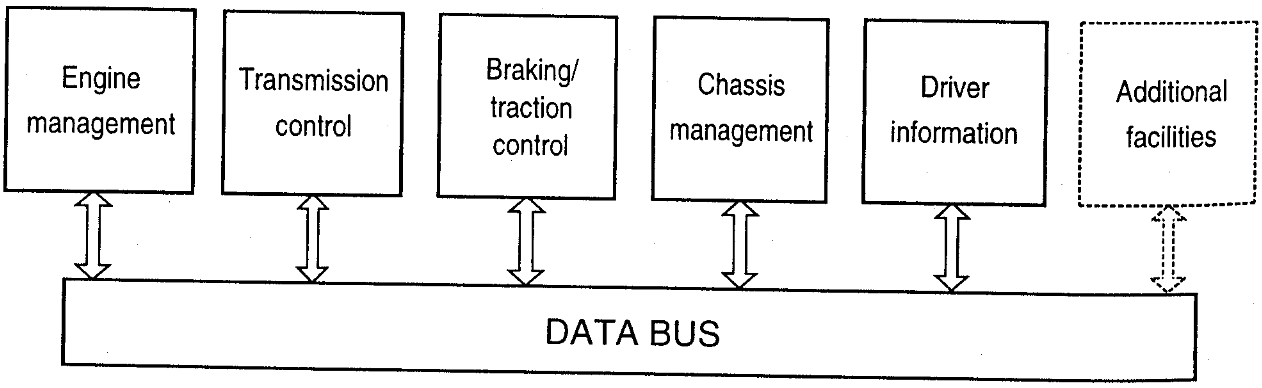


Figure 13.2. Interconnecting control units via a data bus

Information calculated from other modules can be substituted in the event of a sensor defect. Maintenance is also facilitated by having just one diagnostic port for all modules, which may be integrated into the bus or one of the control units.

13.2 Engine Management

Microprocessor controls are now widely used to optimise engine management. Such controls can handle a far larger number of sensor input parameters than mechanical devices can. The electronic engine management system fulfils several functions including:

- fuel injection timing and quantity,
- ignition management in spark ignition engines,
- engine monitoring (temperature, oil pressure, knock limit).

13.3 Transmission Control

The main function of semi-automatic and fully automatic transmission control units is to reduce the demands made on the driver; see also Table 6.12 "Degrees of automation of manual passenger car and commercial vehicle gearboxes".

13.3.1 Automatic Master/Gearshifting Clutch

The electronic control of the master/gearshifting clutch is one step on the way to automating manual transmissions. The function of the clutches is to improve ease of use and to extend the service life of the power train by preventing incorrect operation.

The system diagram of the EKS electronic clutch system (Mannesmann Sachs) in Figure 13.3 is an example of an automated clutch. The production design of a similar system (EKM by Luk) is shown in Figure 12.16a.

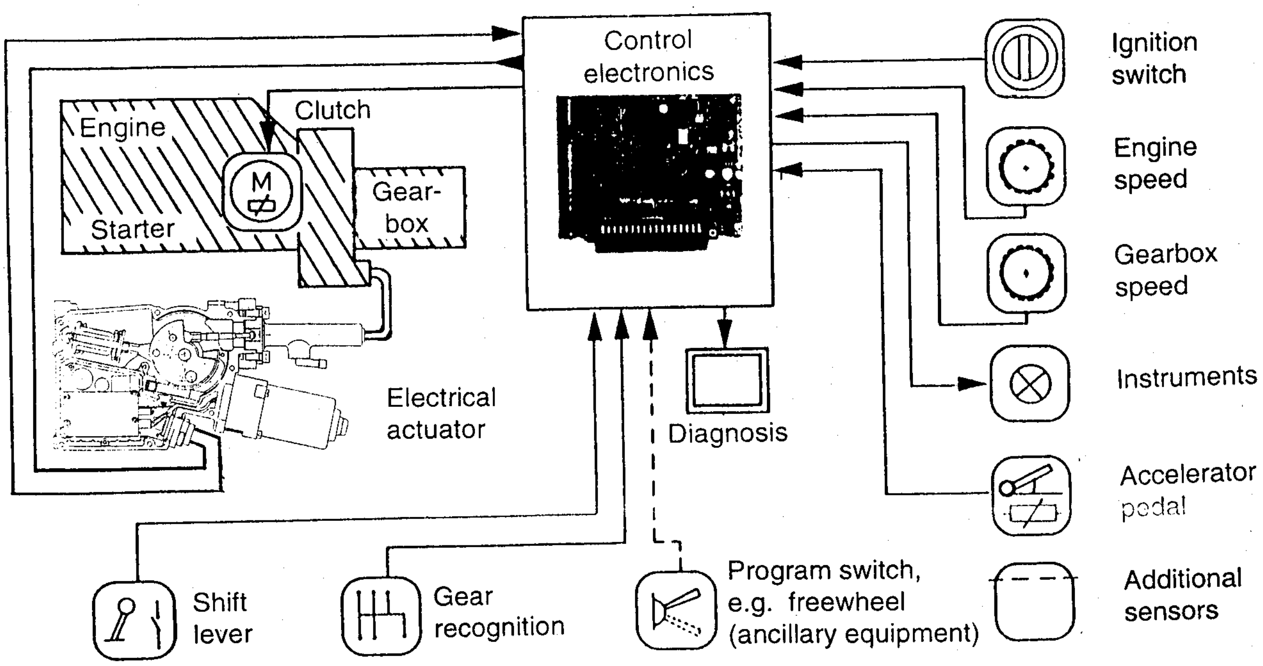


Figure 13.3. Electronic clutch system (EKS) diagram (Mannesmann Sachs) [13.3]

The EKS clutch system comprises the following components:

- dry clutch,
- electrically powered actuator,
- electronic control unit,
- sensors for:
  - engine and transmission speed,
  - accelerator pedal position,
  - gear lever movement,
  - the gear currently engaged,
- function and warning display.

The actuator motor moves a rod by means of a non-self-locking worm gear. This activates the clutch sensor cylinder. The electric motor is disengaged by an over-centre spring mechanism.

Adjusting the clutch with an electronically controlled actuator enables further functions, such as eliminating overrevving when shifting, and filtering out torsional vibrations by controlling clutch slip and anti-lock braking and traction control system support.

13.3.2 Semi-Automatic Manual Transmissions, Automatic Gear Selection

Table 6.12 shows the different degrees of automation of manual transmissions. There is a continuum extending from manual gearboxes to fully automatic selector gearboxes. In Section 9.1.1 "Shifting Elements" a distinction is made between:

- electronic transmission control with mechanical clutch activation, and
- electronic transmission control with automatic clutch activation and engine management.

In the latter variant the whole gearshift operation is electronically controlled.

13.3.3 Fully Automatic Transmissions, Adaptive Gearshift Strategy

The function of automatic transmission control units is to reduce demands on the driver, and to select the optimal ratio in terms of consumption, performance and comfort for the particular driving situation. There also has to be a manual override facility for the driver at all times to handle specific situations.

The shift controls that provide the interface between the control unit and the driver can be designed to meet the particular requirements.

Adaptive Gearshift Strategies – The Porsche Tiptronic-Transmission

Let us consider the Porsche "Tiptronic" transmission control [13.4] to illustrate the potential of an adaptive transmission control – one which adapts to the driving situation. This is a conventional powershift automatic transmission with an additional manual shift facility. The transmission is based on an electronically controlled 5-speed planetary gear with torque converter, lock-up clutch and electronic control unit. The control unit is in the form of a slotted link gearshift lever moving in an automatic gate and a manual gate. The automatic gate provides the familiar lever positions P, R, N, D, 4, 3, 2, 1 of a 5-speed automatic transmission; the Bowden cable link to the gearbox enables it to be operated manually if the by-wire unit fails. The control can be switched to the manual gate from the selector lever position D. Moving the selector lever forwards or backwards shifts one gear up or down. The message is passed on to the transmission control unit electrically. Figure 13.4 shows a simplified system diagram of a passenger car with a Tiptronic transmission.

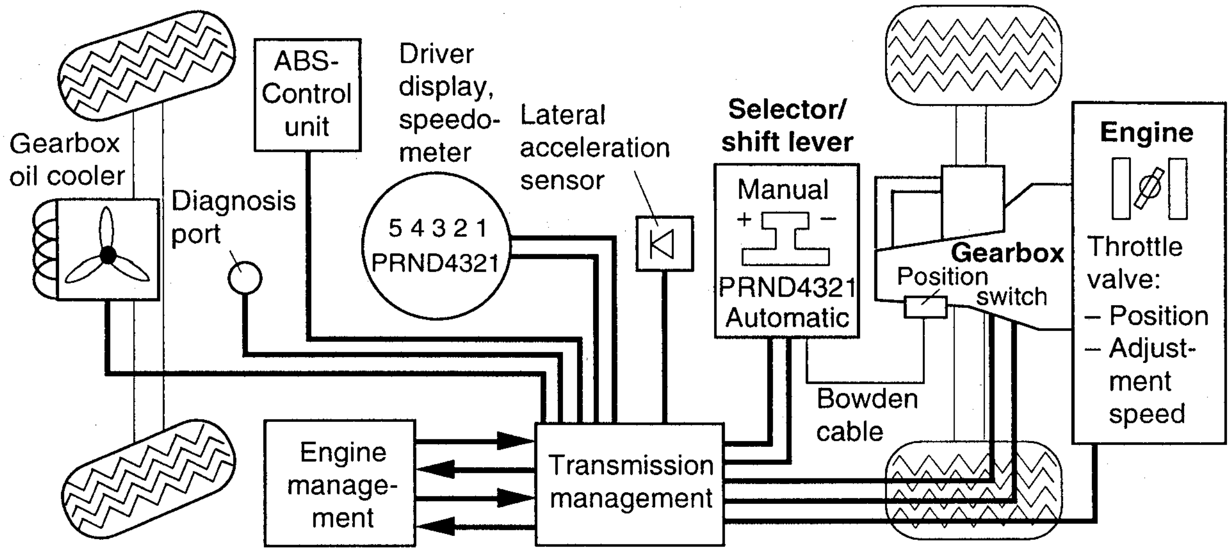


Figure 13.4. Tiptronic system diagram. Fully automatic geared transmission with manual override (Porsche/ZF)

Automatic Mode

In automatic mode the gearshift program is adapted to driving style. For this purpose the control unit works with different shifting profiles, ranging from economy to high-performance.

The control unit "learns" from the driving style. For example, it detects a performance-oriented driver from the high longitudinal and lateral acceleration forces within a given period, and selects the appropriate shift pattern.

The program allows for a degree of hysteresis to prevent the profiles changing too frequently, constantly altering the gearshift response, which may irritate the driver.

The lock-up clutch is controlled by its own torque converter lock-up maps, which are consumption or performance oriented as required. In automatic mode, comfort has a higher priority than in manual mode. In manual mode the engaging and disengaging speeds of the lock-up clutch are displaced towards low speeds, to simulate the character of manual shifting.

The undesirable shifting up that can occur with automatic transmissions when the driver decelerates approaching a bend is prevented in the event of sharp deceleration, harnessing the braking effect of the engine. With slow deceleration the Tiptronic behaves like a conventional automatic transmission.

There is a function for holding the same gear when cornering. A lateral acceleration sensor acts in combination with the road speed to suppress gear changing if it would impair stability in the bend.

Wheel slip can be actively monitored during overrun in automatic and in manual mode by using existing antilock braking system components, causing the lock-up clutch to disengage and shift up if slip increases.

Manual Mode

Manual mode enables the driver to shift one gear up or down. The control unit prevents the engine by automatically shifting up when there is a danger of overrevving, and preventing shifting down where excessive engine speeds would result. The engine is prevented from stalling by automatically shifting down, and by disengaging the lock-up clutch when the idling limit is approached.

Further functions of the transmission control are monitoring the gear oil temperature and the diagnostic facility.

13.3.4 Continuously Variable Transmissions

The control system is particularly important in the case of continuously variable transmissions. Since continuously variable transmissions are intrinsically less efficient than geared transmissions, they require control strategies that take into account the efficiency of both the engine and the transmission, having regard also for comfort and ease of use, especially driveability. Adaptive strategies are appropriate here too; see also Sections 4.5 and 5.3.4.

Chain converter transmissions require a regulated or multistage oil pump to produce adequate taper disc contact pressure for a satisfactory level of efficiency; see also Figure 15.9.

13.4 Electronically Controlled Braking and Traction Systems

Traction control systems, both for propulsion and for braking, contribute to active safety. The slip of the driven or braked wheels is monitored by a control circuit and limited to a maximum of 15%, which represents the optimum for power transmission. Whereas the antilock braking system relies solely on regulating the braking pressure at each wheel, there are several options for the traction control system. One popular method is to reduce the drive force by reducing the engine torque by means of retarding the ignition and adjusting the throttle valve or the fuel injection pump. Since adjusting the throttle valve produces a relatively sluggish response, it is also possible to apply the service brakes to the wheel that is spinning, to slip the clutch in a controlled manner, or to activate the differential lock. There are benefits in sharing components (wheel speed sensors and braking pressure regulators) between the antilock braking system and the traction control system to brake individual wheels, in the form of an integrated antilock braking/traction control system; see also Section 6.10.4 "Alternatives to Self-Locking Differentials". In the case of commercial vehicles there is also the option of controlling the engine brake and retarder electronically, keeping the road speed constant even on downhill stretches by linking to an automatic road speed governor.

13.5 Safety Concepts

Safety is an important consideration in the use of electronics in motor vehicles. The general requirement is that electronic systems should be at least as reliable as comparable mechanical systems. Individual modules must be self-monitoring, for example using built-in watchdog circuits, and sensor range monitoring. Failure of one module must not seriously impair the functioning of the remaining modules. Each actuator must have a fail-safe function; if the associated control unit fails, the system may for example default to a defined state depending on current parameters. Emergency running with restricted functionality is a further requirement. The design must provide for redundancy and interactive mutual monitoring of electronic components, especially in the case of safety-critical functions such as electronic brake activation.

Faults are displayed to the driver and stored in the control unit memory, even if they are only transient. This fault memory bank can then be accessed by a diagnostic unit for servicing purposes.

14 Overview of the Development Process, Product Planning and Systematic Engineering Design

Engineering design is creativity with discipline

The purpose of this book is to present the development process for vehicle transmissions in its totality. A product is only successful if people buy it! The next three chapters are devoted to tools for designing and developing vehicle transmissions.

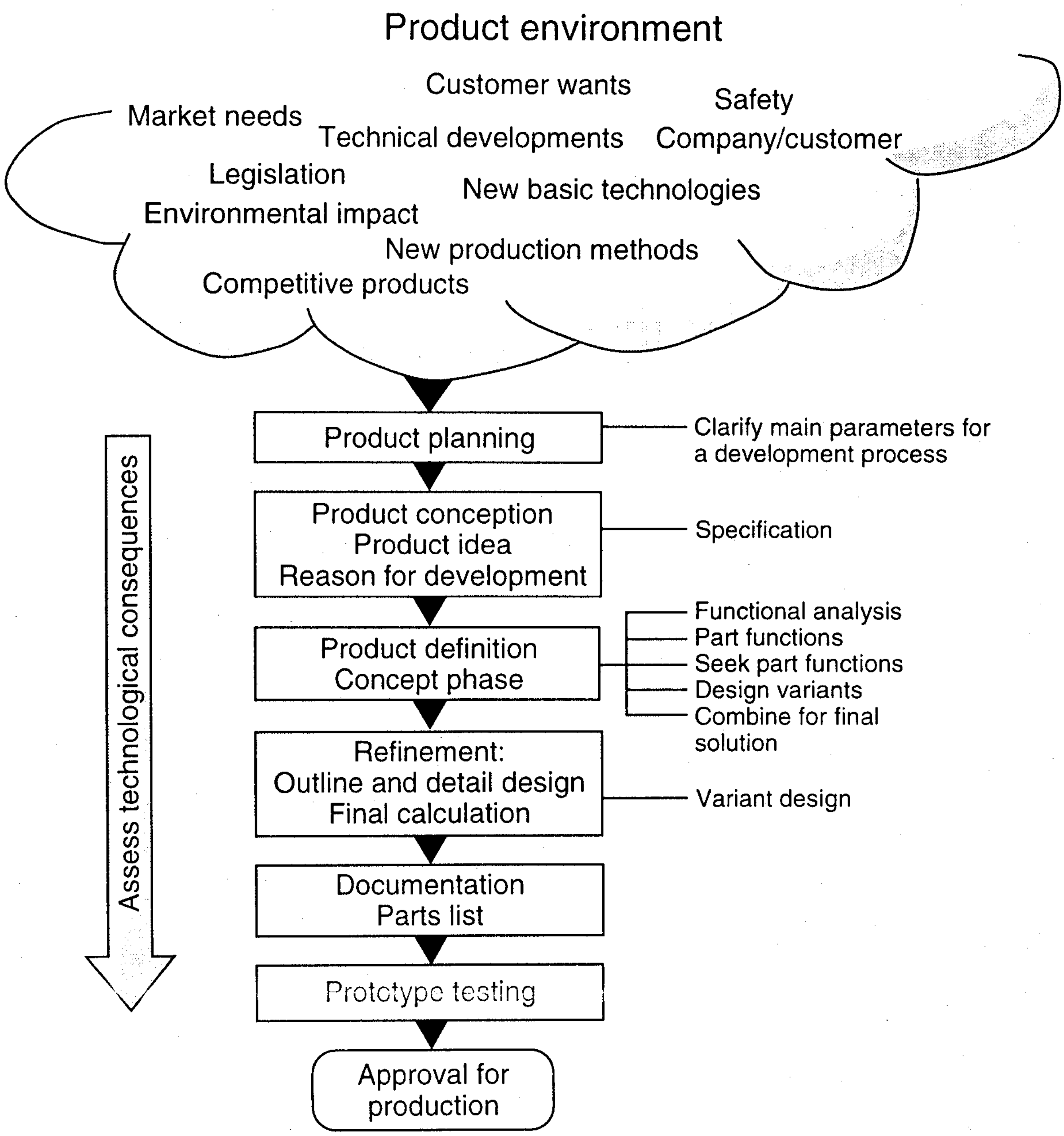


Figure 14.1. Market factors and the development process

The product development process (Figure 14.1) starts with *product planning,* taking into account the main market factors impinging on the product. Next follows the *concept phase,* which is based on formulating the list of requirements (specification); in this phase, variants are proposed and evaluated and the most suitable solution selected. This concept is then refined in the *design phase* by progressive refinement of all the constituent elements. The new product finally goes into production when all the documentation is completed and prototype testing has been successfully concluded. See also "Simultaneous Engineering" and "Rapid Prototyping", Section 14.5.

14.1 Product Life Cycles

All products have a limited life. Every product is eventually replaced by a new one. There are several reasons for this:

- new technical developments offering improved functionality,
- more efficient methods of producing new products,
- fluctuating demand and fashion trends,
- consumer attitudes,
- legal and economic requirements,
- inadequate or inappropriate market policy.

All products pass through various life cycle phases, which can have different profiles (Figure 14.2). Companies have to be aware of this life cycle – i.e. they have to know where each product is in its life cycle, so that action is taken in good time to develop new products.

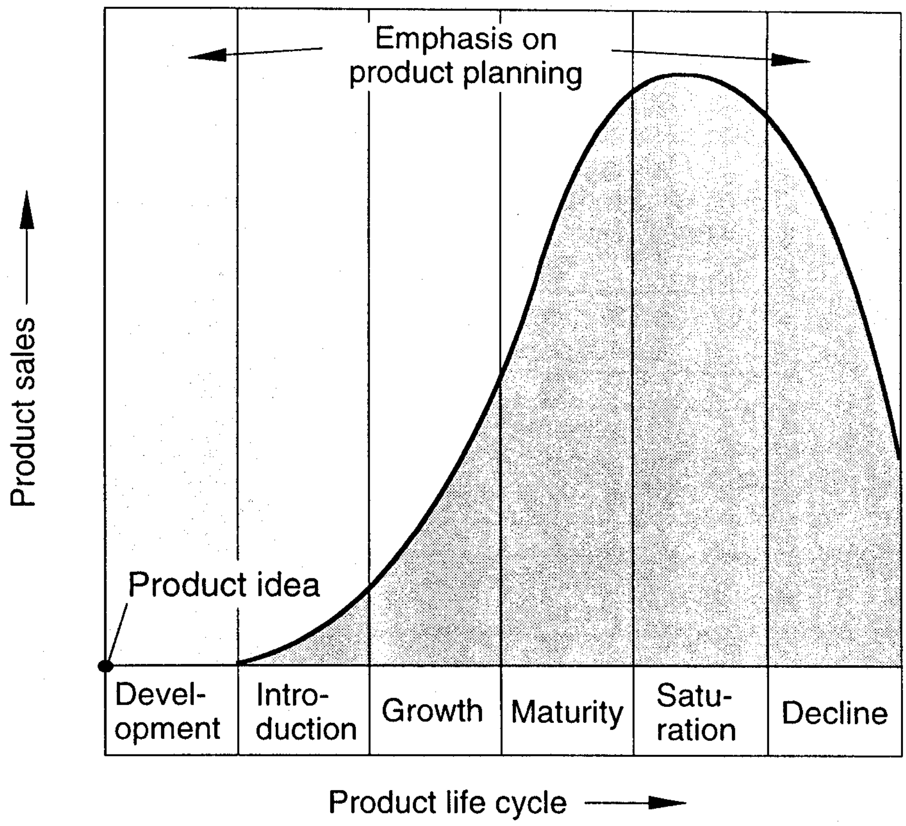


Figure 14.2. Life cycle of a product

The decline in sales of a product as it approaches the end of its life cycle has to be offset by timely development of new products (Figure 14.3).

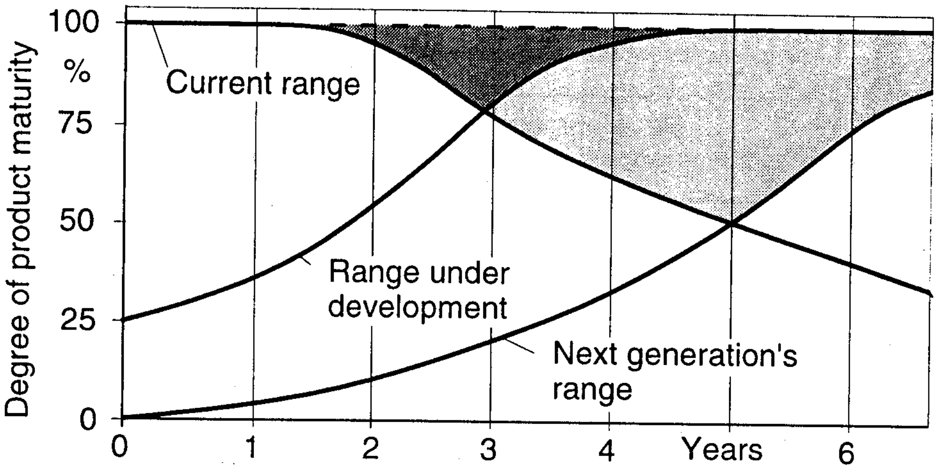


Figure 14.3. Maintaining sales volume by continuously developing new products

Most companies produce a variety of products to achieve a balanced mix of sales volume and profit levels. The age structure of this product range has to be balanced, and it must not be allowed to become outdated. A company is only healthy if products no older than three to five years account for 50% of its turnover. Figure 14.4 shows the age structure of various product ranges, illustrating a healthy product range and an ageing product range.

The life span of various products can differ greatly, and depends very much on how well attuned the product is to consumer needs. The pace of product innovation is continuing to quicken, with rapid advances in technological development. Companies need to provide a balanced product range to satisfy market demands. They have to be in a position to quickly replace an ageing product with a new one. Product flexibility is essential for a company to remain competitive.

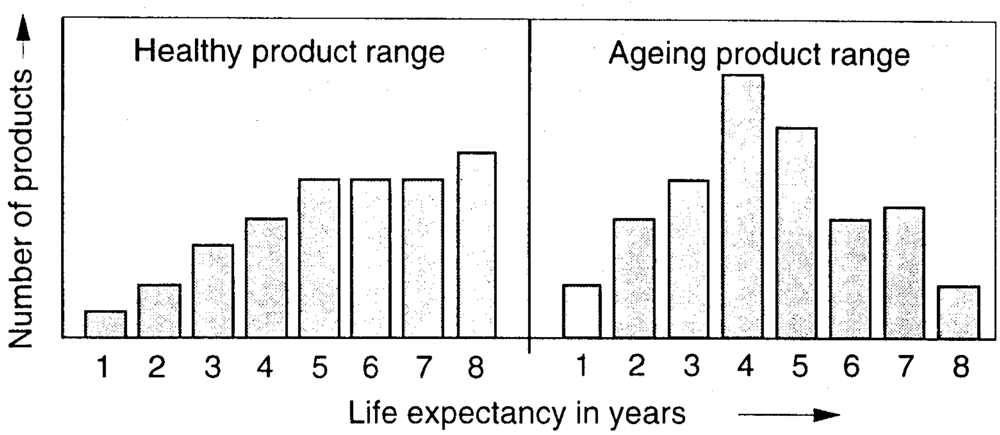


Figure 14.4. Age structure of product ranges

The product life cycle can be defined with the aid of systems engineering. Figure 14.5 shows the stages of the product life cycle proceeding from market needs, through implementation to processing of the waste product.

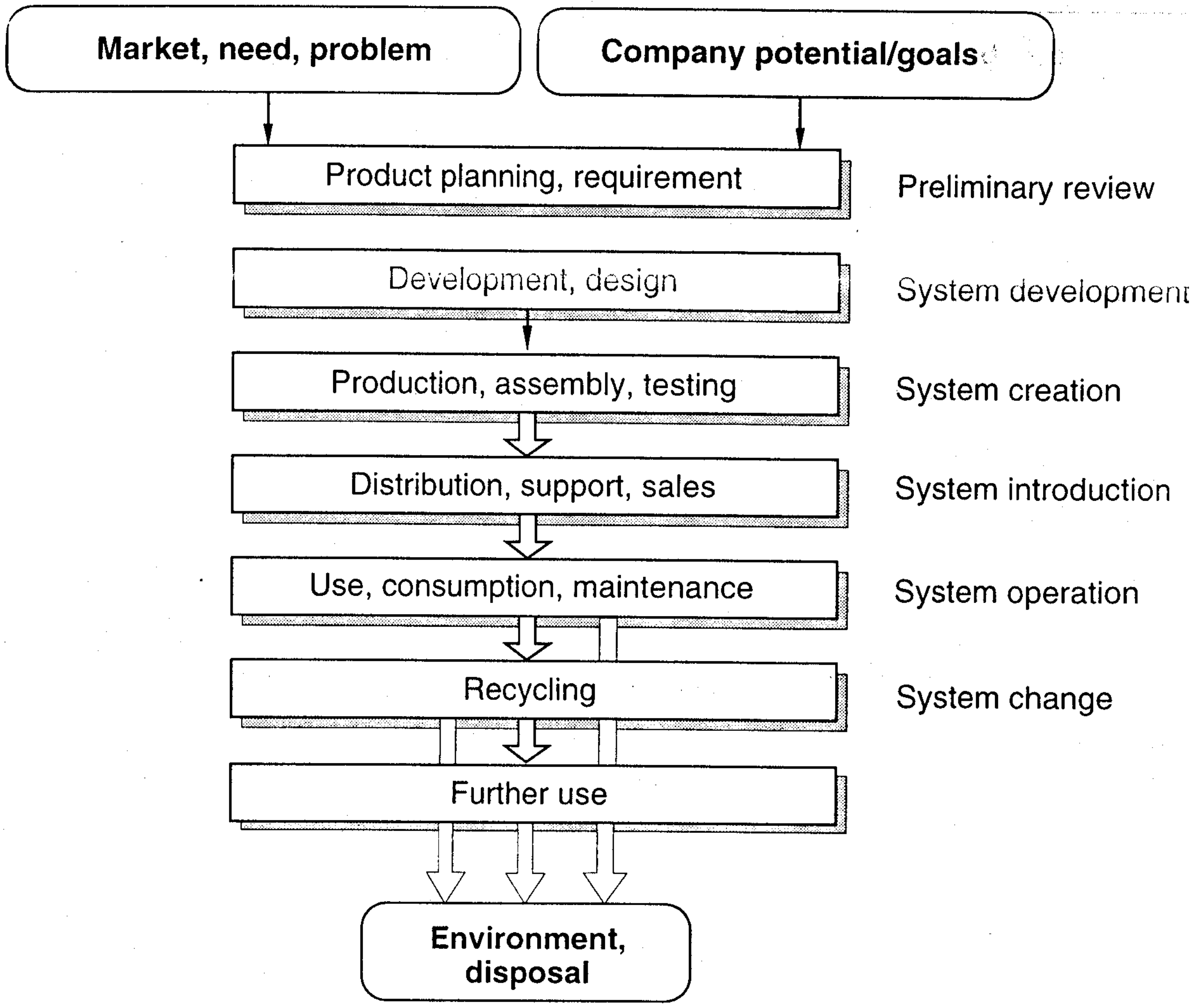


Figure 14.5. Life cycle phases of a product (VDI guideline 2221) [14.1]

14.2 Product Planning

The established principles of new product development must be observed if mistakes are to be avoided. Mistakes arise typically from inadequate research into what is required of the product, and inadequate formulation of the task. The causes and consequences of these shortcomings are illustrated in a light-hearted way in Figure 14.6.

Product planning comprises "systematic integration, co-ordination and evaluation of all the relevant factors from the market, science, technology and industry in order to optimise product development" [14.2].

Product Planning

- seeks and promotes new ideas,
- scans for new products (market, legislation),
- assesses the feasibility of development projects, and directs them,
- plans and monitors the development process.

Securing the future of a company and ensuring long-term job security for its workforce requires development in the short, medium and long term, depending on the urgency. Short-term development might for example involve developing a quiet gearbox; a medium-term development project might involve an electronically controlled selector gearbox linked to an engine management system; a long-term development might be a heavy-duty continuously variable transmission.

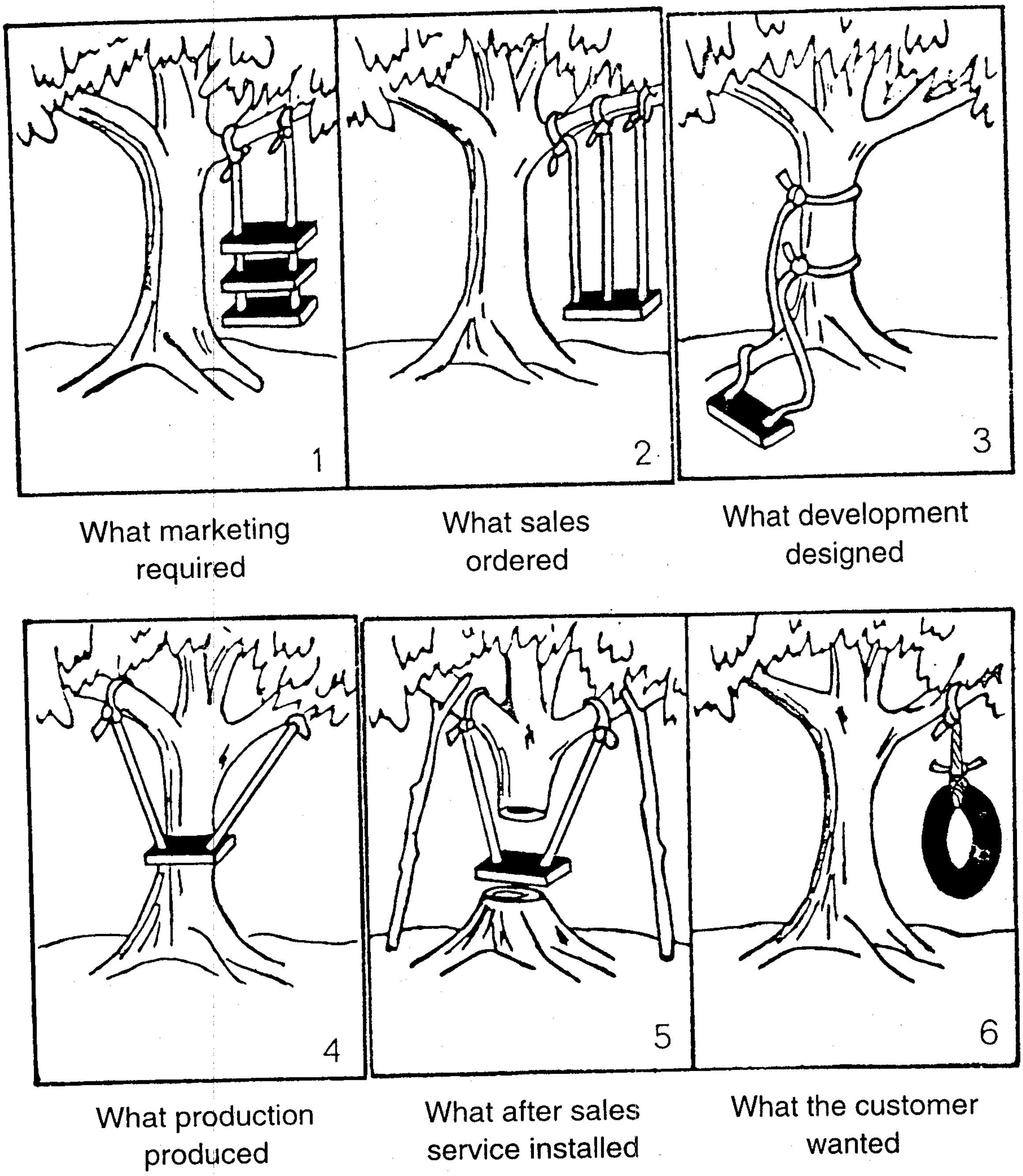


Figure 14.6. Consequences of inadequate product planning

Products can be developed for traditional markets or for new ones, using traditional technologies or new ones (Figure 14.7). The most speculative are new technology products (new products) destined for new markets. For example, a gear systems manufacturer developing robots would be high risk; a refrigeration company developing air conditioning would be low risk.

It is important to bear in mind in product planning that not all ideas are feasible in practice. Out of 100 product concepts, only one or two will succeed in the marketplace. Rigorous review of product concepts can serve to reduce unnecessary expenditure of development funds.

Realistic scheduling is essential. Working out the idea and the concept of a new product takes only a fraction of the time subsequently needed to develop it. This is referred to as the "iceberg effect" (Figure 14.8), whereby the original design accounts for only 6% of the total effort.

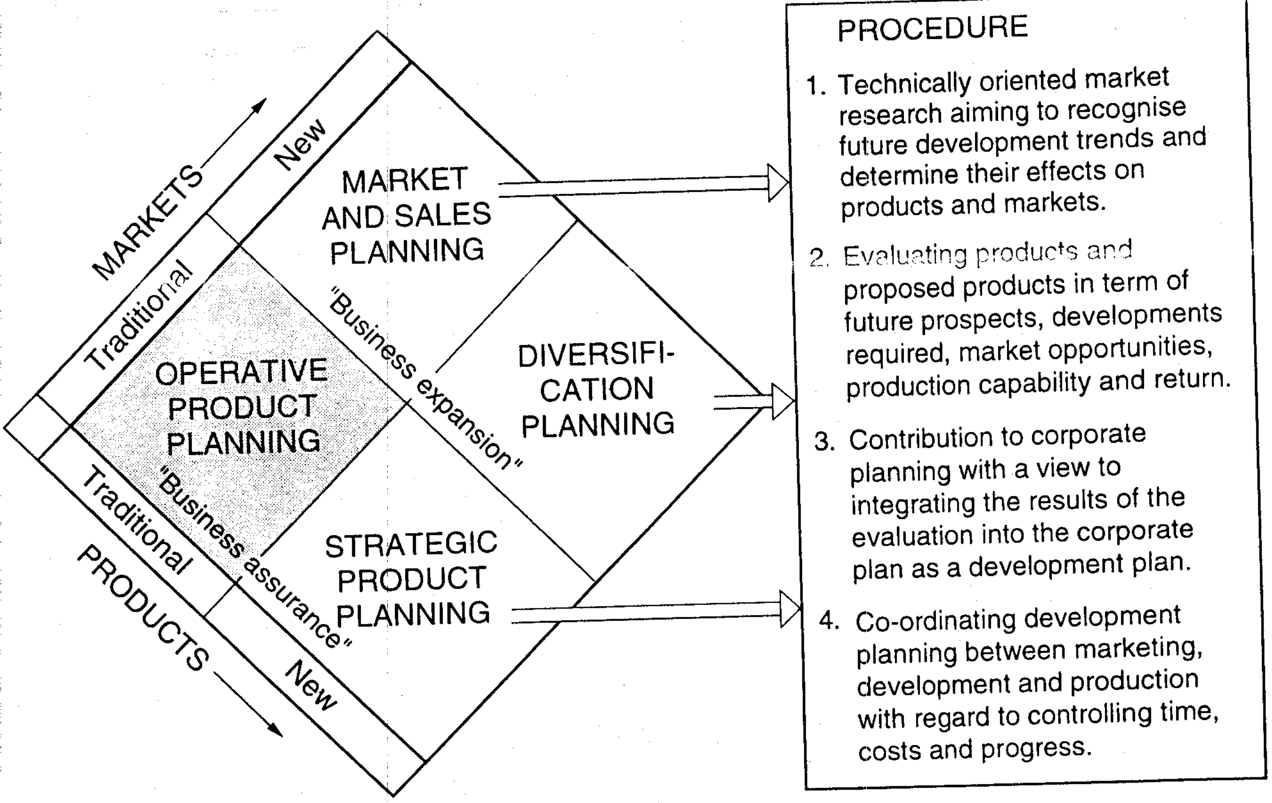


Figure 14.7. Product planning expenditure

The planning and monitoring of progress, costs, loading and capacity utilisation is an important function of product planning in the case of development projects. One tool available to product and project planners is the *decision network diagram,* which defines all the activities in terms of time and cost.

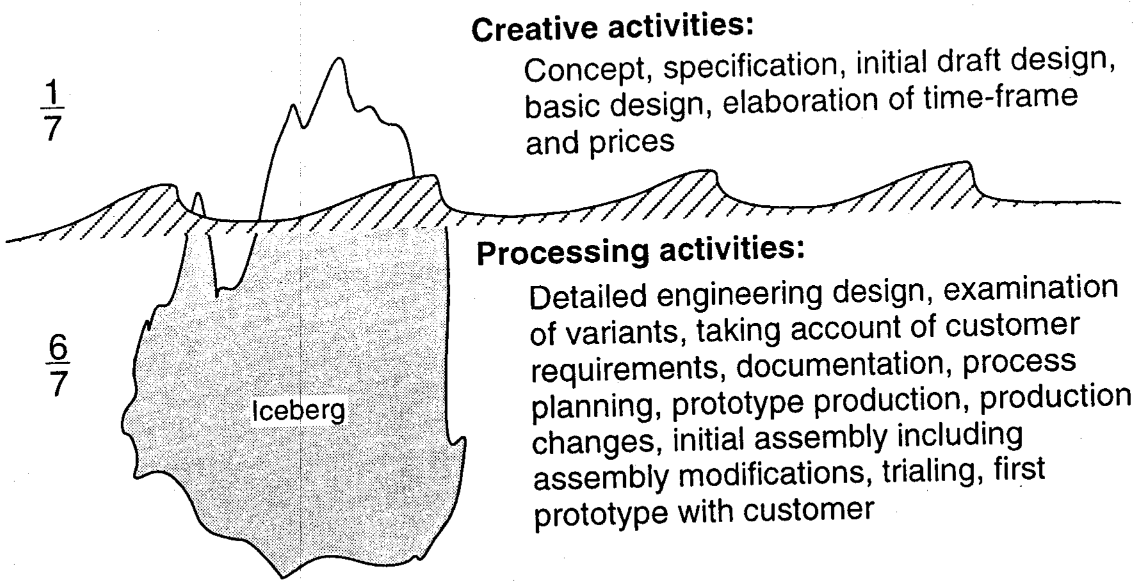


Figure 14.8. Iceberg effect with development projects

Ongoing comparison of actual progress against targets serves both to progress the development project and to point up planning adjustments which can be made on the basis of better information. *"Planning is only an effective guide for implementation if all concerned know that deviations are logged and analysed"* [14.3].

14.3 The Development Process

The terms of the development process: "research", "development", "engineering design", "product design" and "testing" are defined in Figure 14.9.

RESEARCH is activity with a view to acquiring *new* knowledge.

DEVELOPMENT (general term) is functional discharge of a new technical/scientific task on the basis of *existing* knowledge using theoretical, experimental and/or engineering investigation.

ENGINEERING is designing a product or process from the development stage in a cost-effective and practicable form, including appropriate documentation, or

ENGINEERING is providing all information to produce an optimal machine.

DESIGN is the creative process giving the elements of a machine their functional form.

TESTING prototypes

Figure 14.9. Definition and hierarchy: Research, development, engineering design, functional design, and testing

Engineering design is generally divided into 4 activities of varying degrees of complexity (Table 14.1). The creative/intuitive elements of the activity recede as the engineering design process advances, whilst the deterministic activities increase (Figure 14.10).

Table 14.1. Main activities involved in the design process

	Innovation	Development	Adaptation	Alteration
Frequency	15%	20%	30%	35%
Difficulty	very great	great	moderate	low

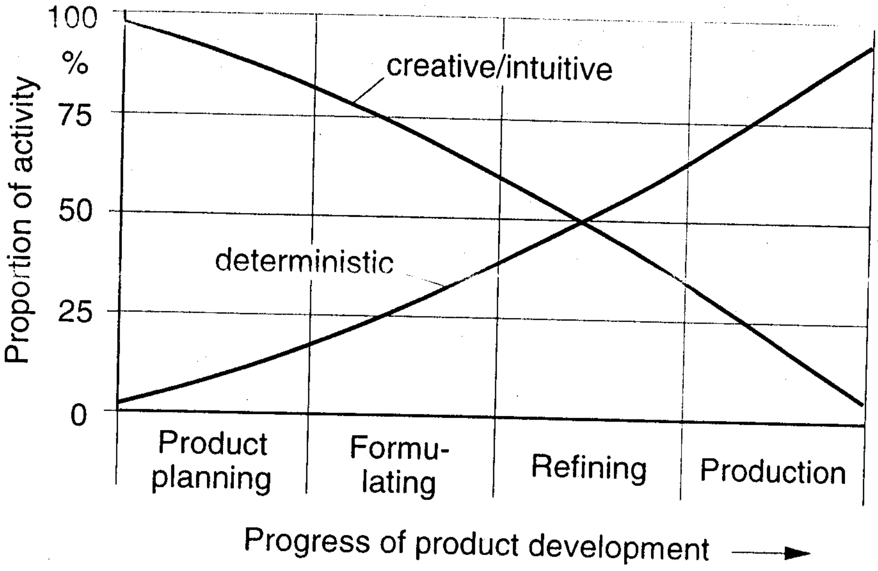


Figure 14.10. Changing activity profile as product development proceeds

But the engineering design process as a component of the development process includes not only developing of new products but also maintaining their value. Changes have to be made to the products throughout their life to optimise their function, adapt their performance, prevent failures and adapt to market trends (styling, design). Figure 14.11 shows the various activities involved as a proportion of the overall operation.

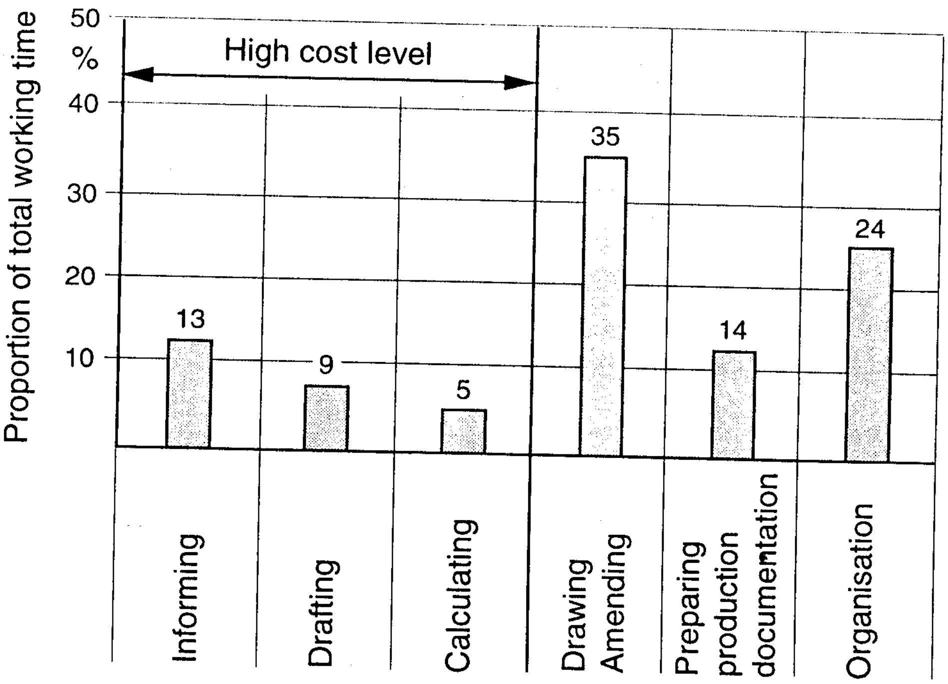


Figure 14.11. Time required for typical development activities

The design engineer has the main responsibility for the product. In developing the product he has to take into account its functional design, its serviceability and durability, reliability and maintenance aspects, technological questions and cost considerations. He is also responsible for modifications to the product as a result of feedback from users and the market. He is responsible for the environmental impact and recycling of the product.

Figure 14.12 shows the product life cycle from the initial idea, through the birth of the product to the end of its life, from the point of view of Computer Integrated Manufacturing (CIM).

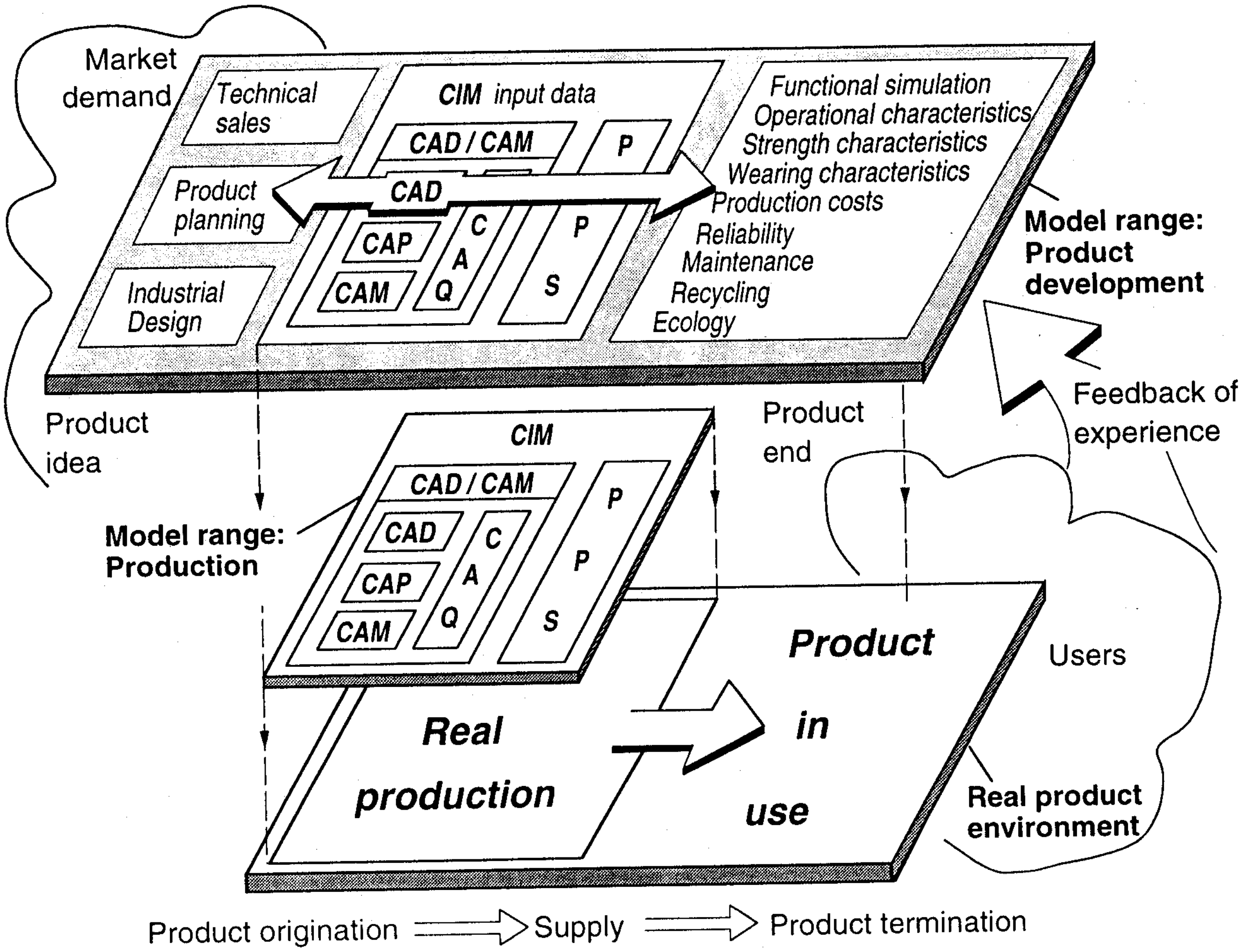


Figure 14.12. Product life cycle in terms of computer integrated manufacture (CIM). Source: EHRLENSPIEL

The main aim of CIM is integration. It enables the data generated in development to be linked, especially with product planning and product control. CIM includes computer-aided design (CAD), computer-aided planning (CAP), computer-aided manufacturing (CAM) and computer-aided quality assurance (CAQ), which are co-ordinated by production planning and control.

The essential function of creative engineering design is supported by "systematic engineering" (Figure 14.13).

14.4 Systematic Engineering

The purpose of systematic, methodical engineering design is to support the creative design engineer. *Systematic engineering cannot replace creativity!* Systematic design has been promoted especially in Germany since 1965 [14.4 to 14.15]. In the Anglo-Saxon countries the main emphasis is placed on creative engineering design [14.16 to 14.17].

Systematic engineering is of particular relevance for innovation and refinement. For product adaptations and modifications, an abbreviated process is appropriate. The main phases in systematic engineering as defined in VDI guideline 2222 (Figure 14.14) are: *planning, formulation, draft, refining*.

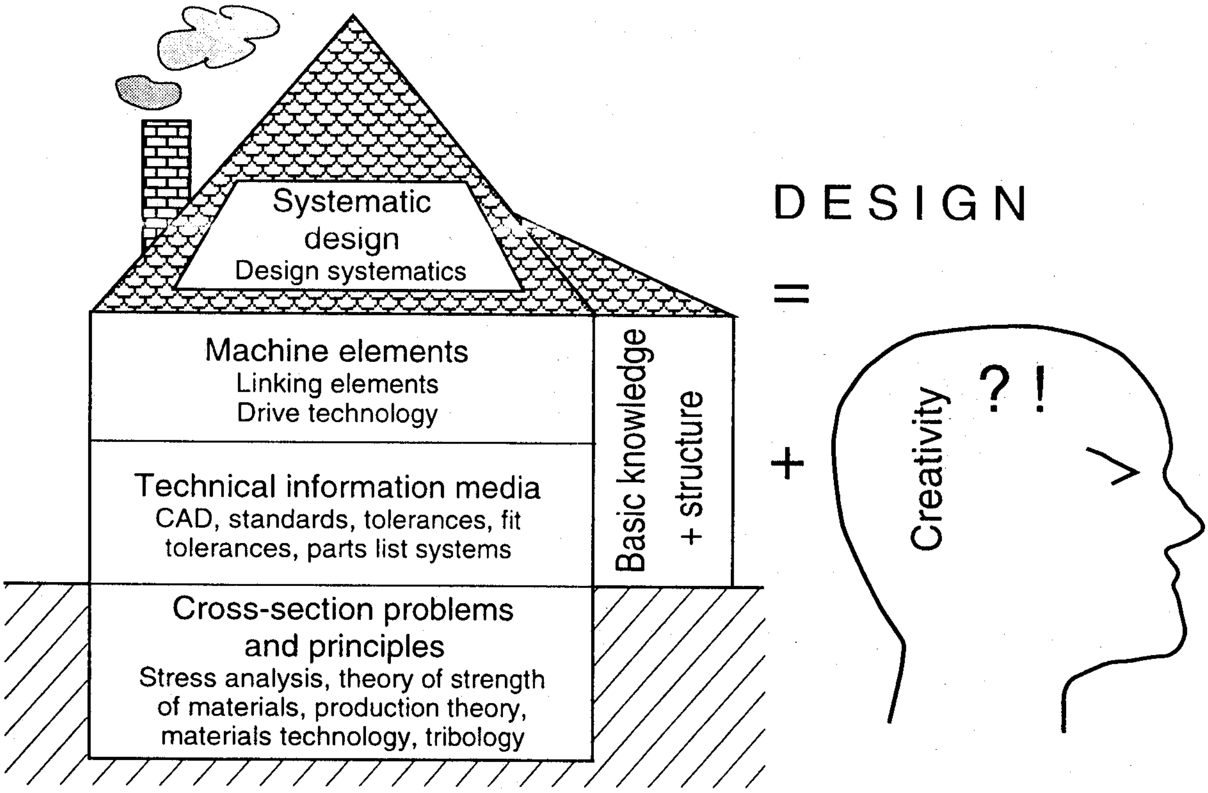


Figure 14.13. The "systematic engineering design" structure (fundamentals and presentation) complements creative design

These phases of the task are punctuated by product evaluations, and decisions on the future of the project. In practice this systematic development process is often adapted to the circumstances of the particular company.

BEITZ [14.18] offers some new insights into engineering design methodology. He suggests breaking down the self-contained, strictly methodical sequence into a number of individual "design modules". These individual modules can be combined individually for the specific development project. The individual tasks identified by BEITZ (Figure 14.15 (1–8)) are as follows:

1/ definition/specification of the terms of reference,
2/ setting up functional structures,
3/ searching for principles to solve the problem,
4/ combining partial solutions to achieve an overall solution,
5/ selecting solutions,
6/ draft application design,
7/ final application design,
8/ specification and itemising.

As the design process advances, the requirements are gradually firmed up, which requires review and optimisation at each stage. The decision-making processes involved are a key factor affecting the efficiency of the design process. The solution selected at each stage defines the parameters for the stage that follows.

The golden rules are:

- If a solution seems too good to be true, it probably is!
- A successful new product that performs well comprises around 70% proven parts and only 30% newly developed parts.

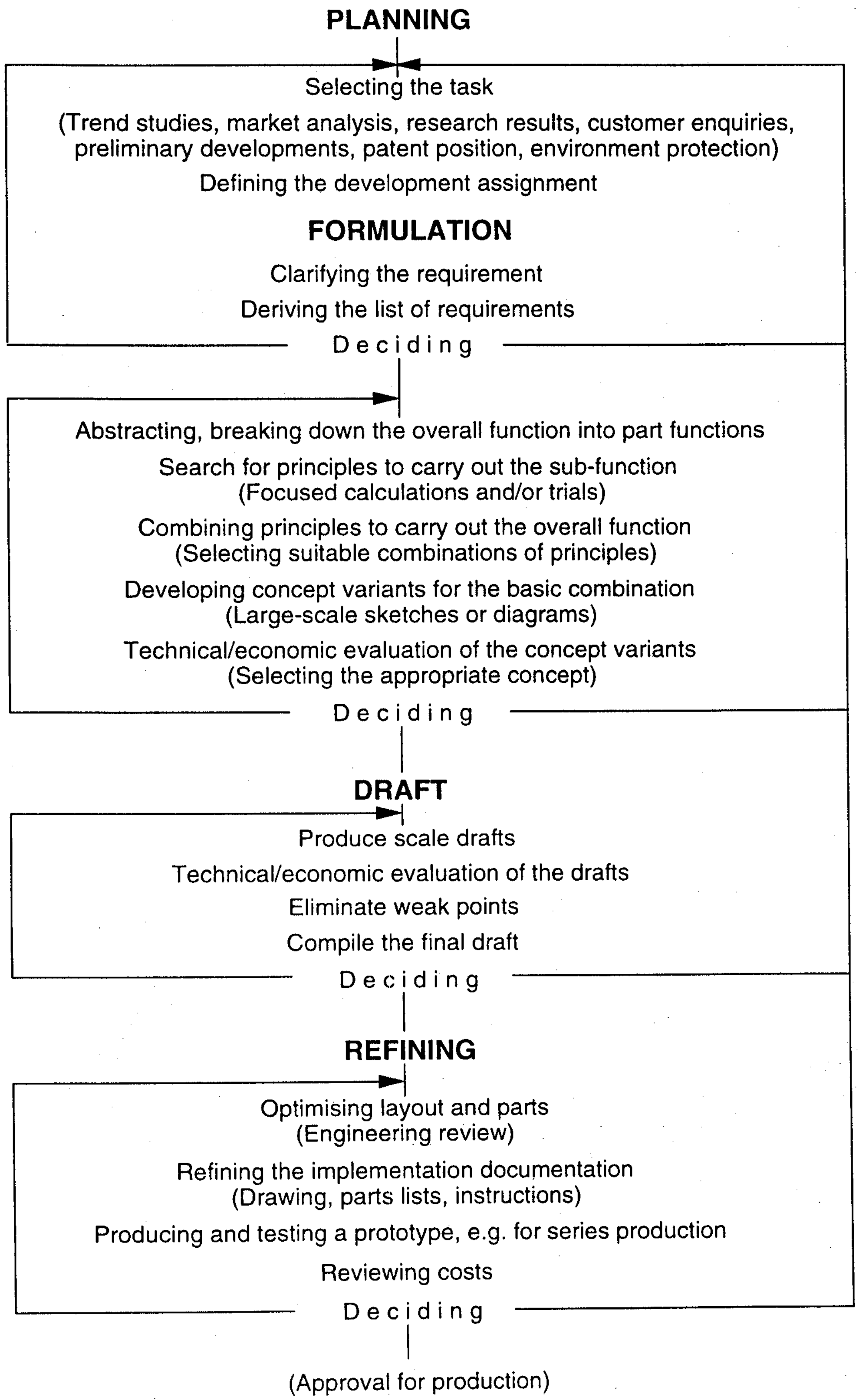


Figure 14.14. Practical approach to product development [14.19]

1/ Formulating the Terms of Reference

Clarifying the task helps to gather information on the requirements the solutions has to satisfy, and generates a schedule of requirements. This specification includes both *functional* and *operational* requirements.

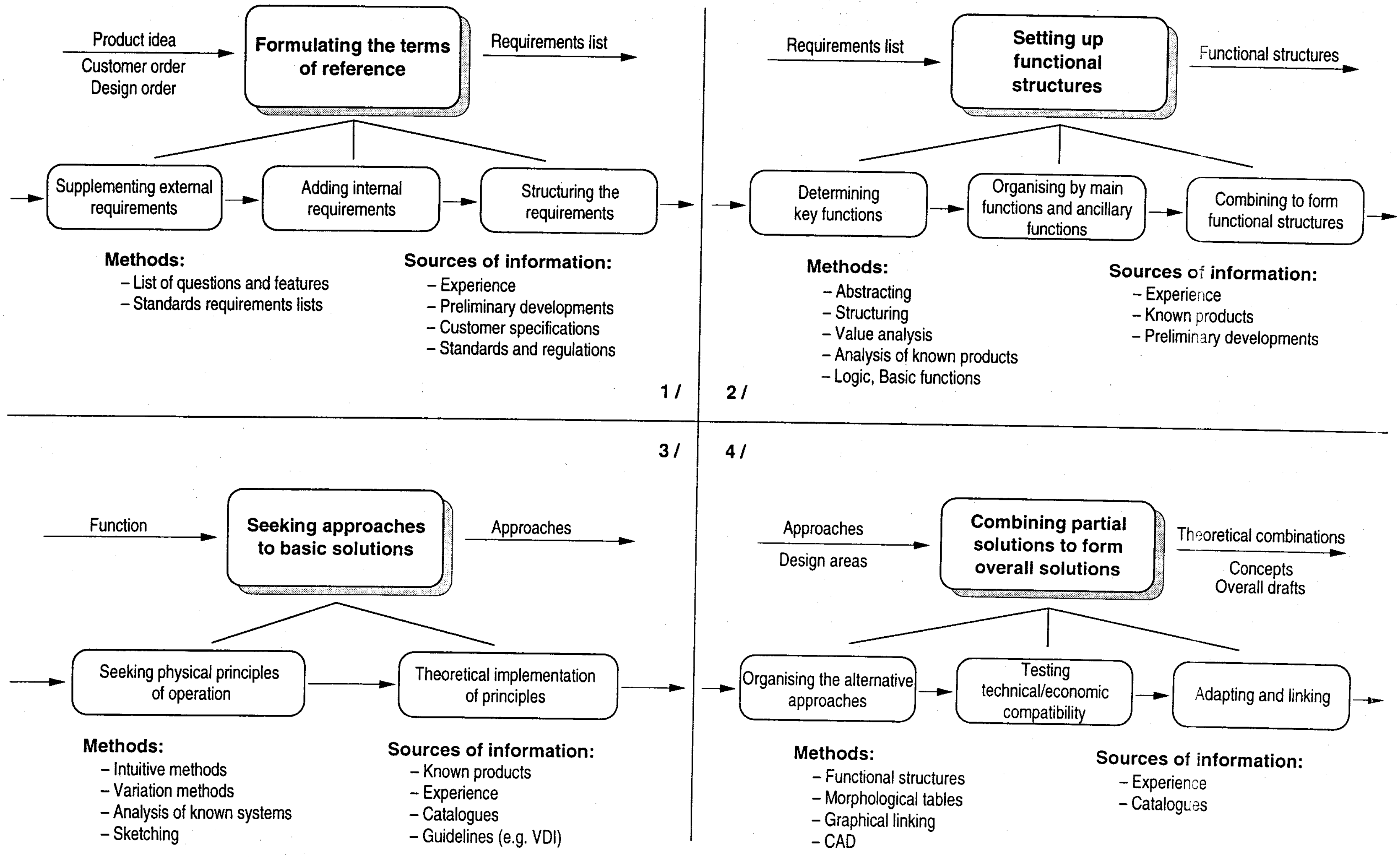


Figure 14.15a. The modules of the systematic design process [14.18]

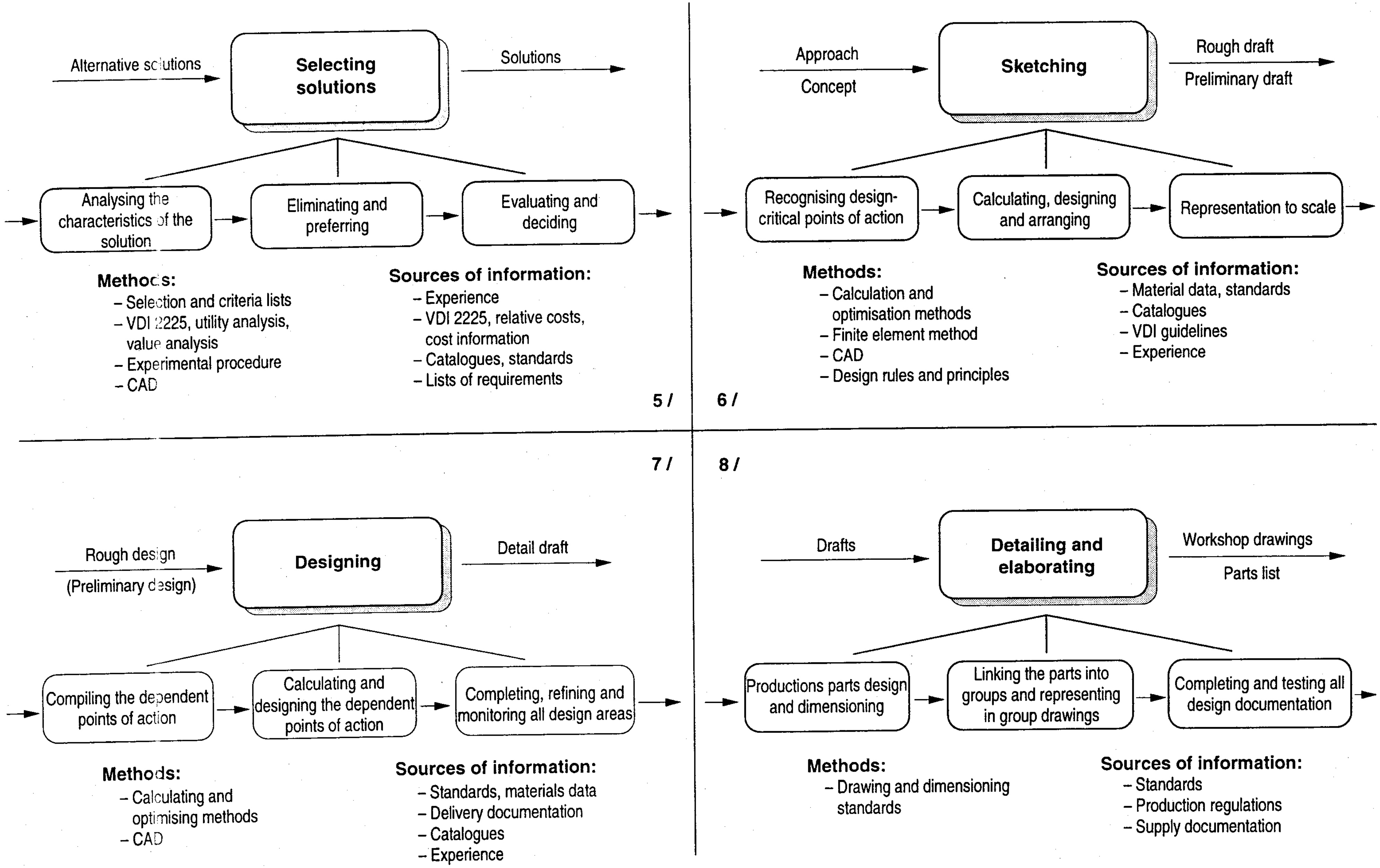


Figure 14.15b. The modules of the systematic design process [14.18]

Functional requirements are broken down into main functions and ancillary functions; operational requirements include reliability, cost-effectiveness and human aspects.

The external form of a schedule of requirements depends on the operational circumstances, but should always include the various requirements the product has to meet (indicating whether they are essential or merely desirable), the department responsible and a description of the changes.

It is advisable to follow the *"Must – Want – Nice"* rule, so as not to overburden the specification with too many requirements. "Nice" features and even "Wants" may have to be sacrificed if the project is to remain feasible.

2/ Setting up Functional Structures

The design structure can be abstracted from the engineering drawing; the functional and operational structure of a technical system can then be derived by further abstraction. Figure 16.14 shows the functional structure of a single taper synchroniser, represented as a functional block diagram.

The *design structure* is a schematic representation of the production version of the design. Unlike the functional structure, it is subject to dimensional constraints. The *functional structure* is the simplification of a design to a function that can be described in mathematical terms, and can be regarded as a kind of "circuit diagram" of an assembly. PAHL and BEITZ [14.11] propose that the *function* is the general relation between the input and the output of a system, for the purpose of performing a task (Figure 14.16). The main function of the vehicle transmission as a system is to convert torque and speed.

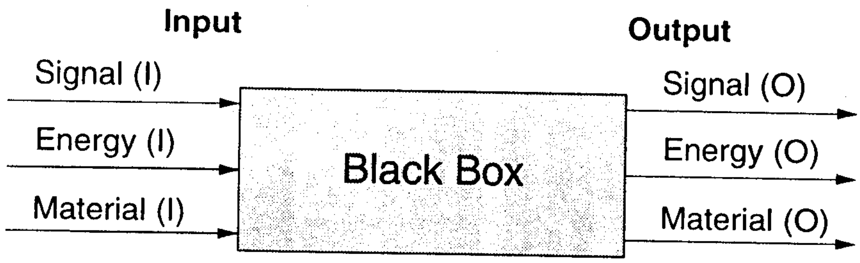


Figure 14.16. Representation of a technical system as a Black Box

An *overall function* can usually be broken down into distinct *sub-functions*. (The sub-functions of a transmission are to enable moving off, transmit power, and control torque/speed conversion.) Certain functional constraints apply to combining the sub-functions to perform the overall function; often certain sub-functions must be concluded before other sub-functions can be performed. A functional analysis has to be carried out to find solutions for the sub-functions, and is embedded in the concept phase of the development process.

The *active structure* represents the interlinking of active principles from the various sub-functions. At the *locus of action* the physical *principle of action* is applied by arranging *action interfaces* and selecting *action movements* to carry out the function [14.18].

3/–8/ Further Design Engineering Tasks

Solutions have to be found for the sub-functions (Figure 14.15a3). Since there are normally several part solutions available, they have to be narrowed down by testing and evaluation. The alternatives are then evaluated by carrying out comparisons of different variants, taking into account both technical and economic criteria. The process demands systematic variation of solutions, and critical, formal selection of the preferred solution [14.19].

The *basic solutions* can be derived systematically, using one of several design engineering tools:

- *Engineering design catalogues* can be used to select solutions. The information on the individual stages in the process is taken from catalogues by applying selection criteria. "Algorithmic engineering design selection technique using catalogues".
- The systematic search for solutions can make use of analytical tools that represent catalogues of solutions organised by type and complexity. They provide a means of combining solutions to build a viable system. Such tools are called *morphological tables* [14.20]. The critical factor is selecting the classification factors.
- A more intuitive approach that relies on the dynamics of group interaction is *brainstorming*. This involves sharing ideas in a group drawn from different disciplines, in a non-judgemental context, to tease out possible solutions. This method is primarily intended for non-technical problems, but can be applied in design engineering.

Having established approaches for the various sub-functions, a solution matrix can be compiled. Various approaches can be derived for the overall solution by alternative *combinations* (Figure 14.15a4). The main problem with the combination method is deciding which combinations do not clash. The solution matrix is constrained by considerations of avoiding clashes.

There then follows an *evaluation, selection and decision-making process* (Figure 14.15b5). The appropriate solutions are selected from the range of solutions available, using systematic, verifiable selection procedures. This process is assisted by compiling selection lists. The main characteristics of the combinations of principles proposed must be considered qualitatively and quantitatively. Selecting the right evaluation criteria is particularly important.

The subsequent design tasks can be broken down into sketching, designing, detailing, and elaborating.

Sketching is the first step in the draft phase, and is characterised by large-scale sketches (Figure 14.15b6). The main functional elements determining the overall design must be roughly determined, taking into account the parameters set.

The next step is *designing* (Figure 14.15b7). The design of the main and subsidiary functional elements is defined by applying design rules, legal requirements, standards, calculations and test results.

When checking and evaluation have been completed, the draft is finalised and the parts list drawn up to form the basis for *detailing* (Figure 14.15b8). Here the individual parts are drawn in detail taking into account criteria for optimising shape, material, surfaces, fits and tolerances. The objectives at this stage are good material utilisation and detail design that is cost-effective and efficient to manufacture, taking into account applicable standards, and using as many bought-in parts or existing in-house parts as possible.

The final stage of *elaborating* involves structuring or organising the production documentation in the form of parts lists and drawings, by arranging products for function or for production/assembly. The production documentation is completed by finalising operating instructions, installation, assembly or transport instructions.

14.5 Linking Development and Production

Product development and production scheduling must be co-ordinated. Production planning and control must be involved from the start of the development process. This applies in particular to mass-produced products such as vehicle transmissions, which are constantly being refined in terms of function and cost.

Computerisation throughout the manufacturing process (including development, design engineering, scheduling, production, assembly, quality control and sales) provides an effective means of achieving increased efficiency, greater flexibility in respect of market requirements, and optimum utilisation of existing resources (see also Figure 14.12 "CIM").

The current market is such that products are required to be innovative, and increasingly to be complex. This implies longer development times and thus higher development costs. Product life spans are also becoming ever shorter. The technique of "simultaneous engineering" has been developed to meet this challenge, enabling product development to be carried out in parallel (Figure 14.17).

Integrated information processing (shared databases, compatible software) enables parallel working between the various departments involved in a product development project. This can shorten the development time required for a product.

The technique of "rapid prototyping" can in addition bring to bear at an early stage of development findings and decisions that would only have emerged at a later point in time in a conventional serial development process.

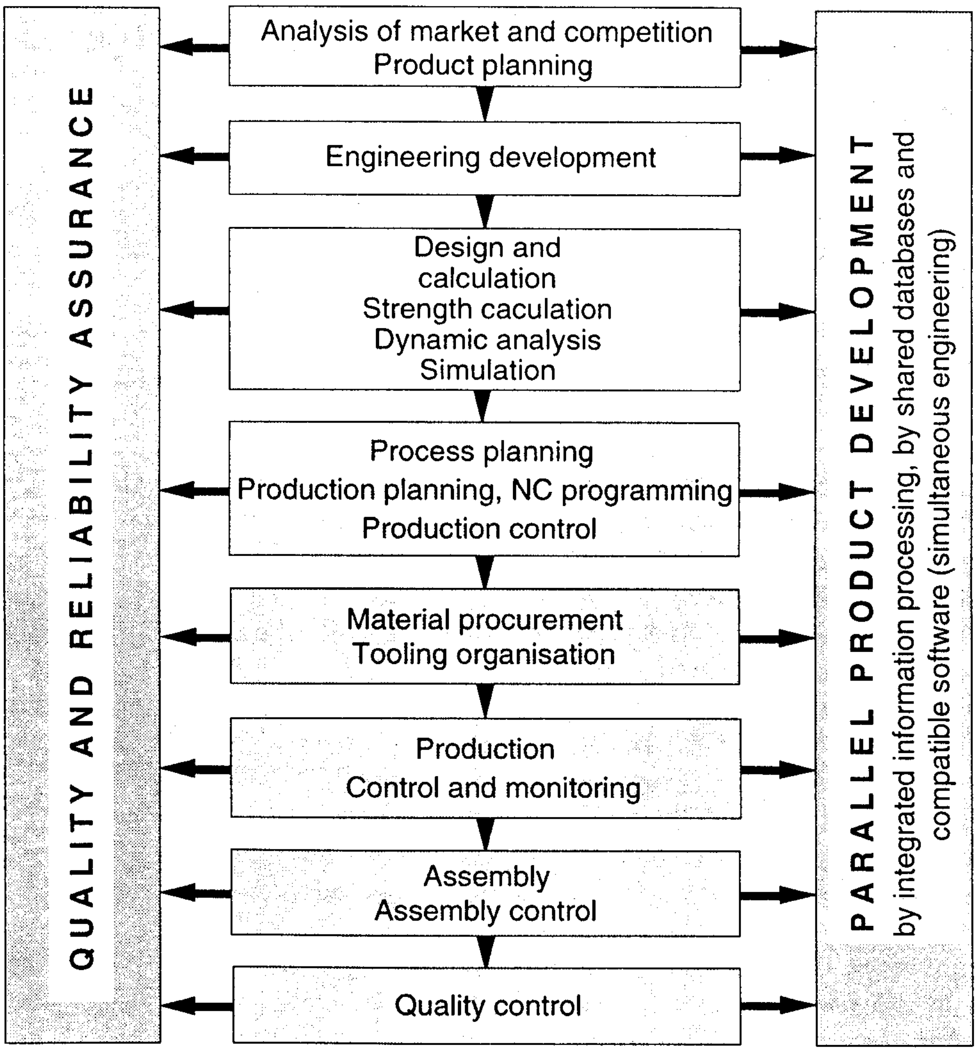


Figure 14.17. Development and production process with parallel product development

15 Computer-Aided Transmission Development, Driving Simulation

Simulation means representing
real-world phenomena on a computer

Modern development processes such as "Simultaneous Engineering" or "Rapid Prototyping" (Section 14.5) are coming to rely increasingly on simulation of technical and technological processes. Such computer-based simulation can reduce development time and thus the cost of development [15.1].

Computer-aided design and development of vehicle transmissions are now very widespread. At every stage of development, the transmission engineer has a wide range of CAE tools available (CAE = Computer-Aided Engineering) (Figure 15.1).

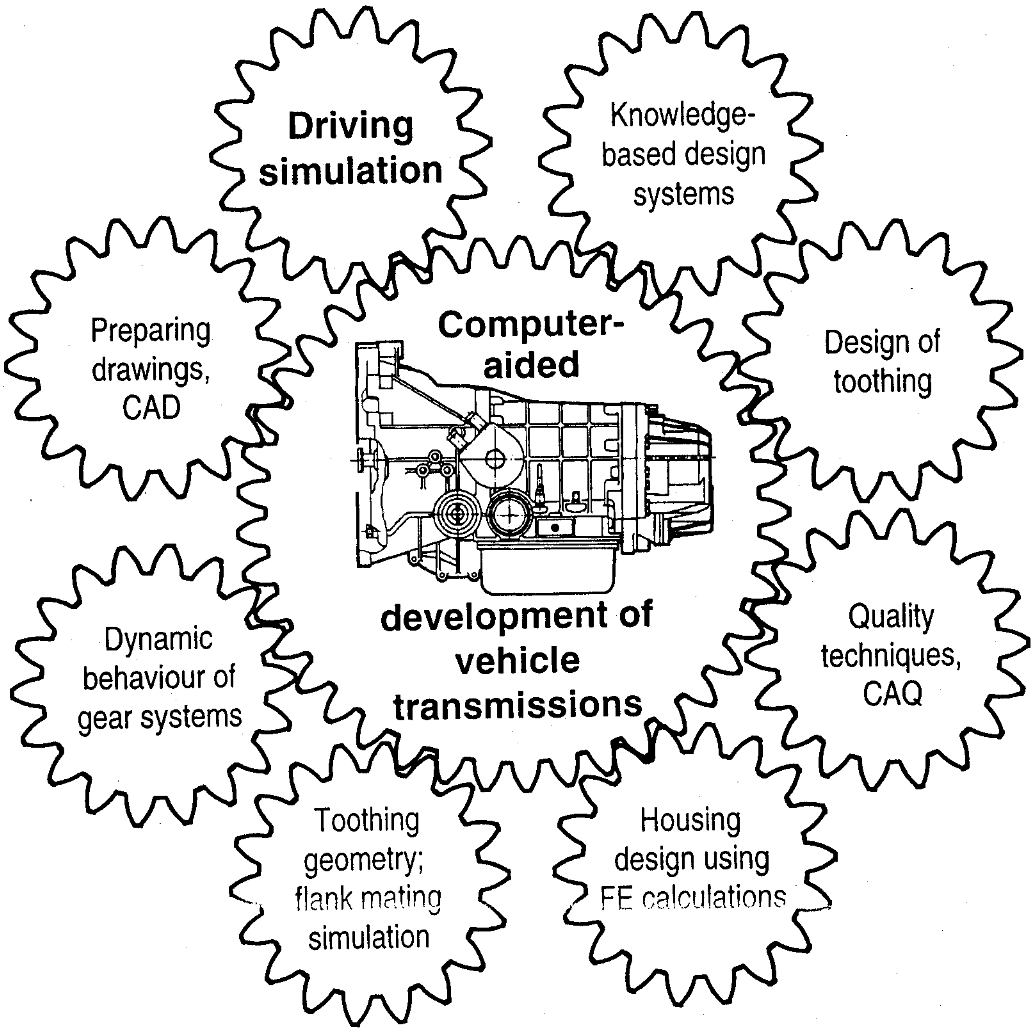


Figure 15.1. Computer-aided development of vehicle transmissions

Computer-aided driving simulation is discussed in greater detail below, as an important application of computers in the development of vehicle transmissions. Driving simulation processes for automotive and transport engineering questions can be divided into three categories, as shown in Figure 15.2 [15.2].

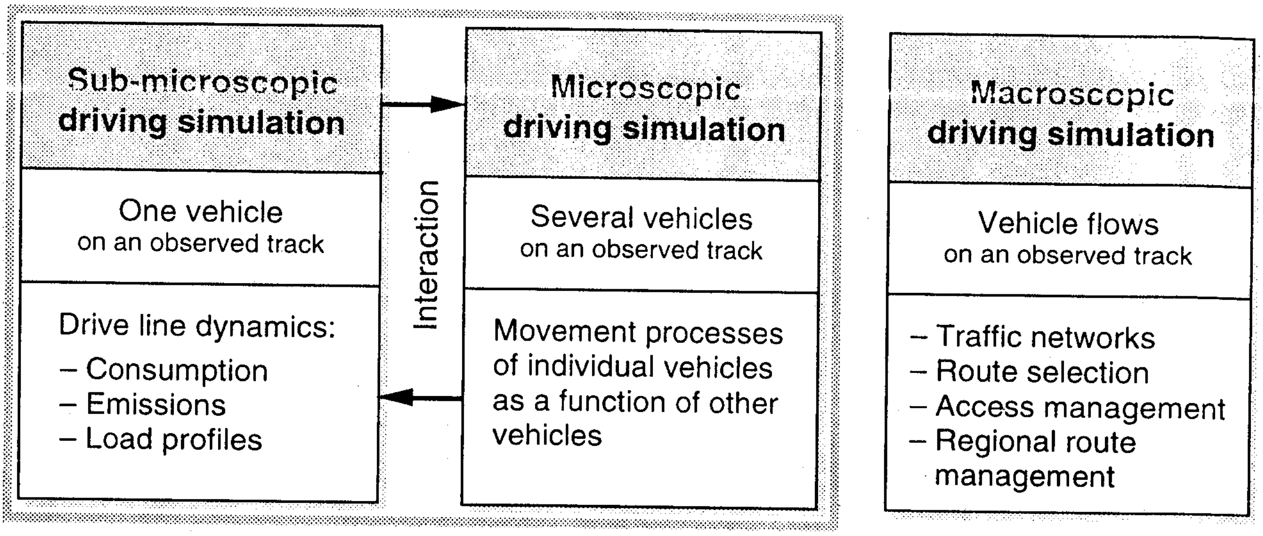


Figure 15.2. Rough classification of driving simulation methods. Sub-microscopic and microscopic driving simulation are increasingly used in combination

Sub-Microscopic Driving Simulation (Driving Simulation)

Sub-microscopic driving simulation is the classic application of driving simulation with regard to optimising power-train components. The first applications arose back in the 1970's. The vehicle is represented in exploded form in the computer, i.e. all the vehicle's or the power train's significant components are modelled. The depth of modelling depends upon the particular requirements. Sub-microscopic driving simulation is generally referred to as "driving simulation".

Microscopic Driving Simulation (Traffic Simulation)

The driver rarely encounters *unconstrained* driving conditions leaving him free to choose his speed and style of driving. Drivers increasingly encounter *partly constrained* or even *constrained* traffic situations. Nose-to-tail traffic with no means of overtaking is an example of constrained driving conditions.

In microscopic driving simulation the vehicle is considered in its traffic context [15.3]. Attention is focussed on the movements of the vehicle under investigation related to the prevailing traffic situation. These movements in turn act on the components of the individual vehicle. Sub-microscopic and microscopic driving simulation are therefore increasingly combined (Figure 15.2).

Macroscopic Driving Simulation (Traffic Flow Simulation)

In the case of macroscopic driving simulation, whole vehicle flows are considered using continuum mechanics. The technique is generally used to study traffic management questions [15.4].

The term "driving simulation" discussed in the next section refers to sub-microscopic driving simulation at the component level, viewing the power train as a set of components. Simulation procedures for investigating directional control (so-called twin-track models) are not considered here.

15.1 Driving Simulation

The significance of simulation is constantly increasing as development cycles in automobile engineering become ever shorter. The use of simulation significantly reduces development time, particularly where alternative solutions can only be evaluated by cost-intensive test runs or bench tests. This applies particularly to the development and optimisation of power-train components. Computer-based tests can not completely replace road tests, but can reduce and complement them (Figure 15.3).

Computer simulation of *vehicle longitudinal dynamics* gives an indication of certain variables at very early stages of development when there are still no prototypes available. These variables include:

- consumption (fuel, electrical energy),
- emissions,
- performance,
- load profiles for predicting service life, and
- driveability.

The complex system comprising driver, vehicle and road can be represented in abstract form so that the effect of individual changes on the system as a whole can be investigated.

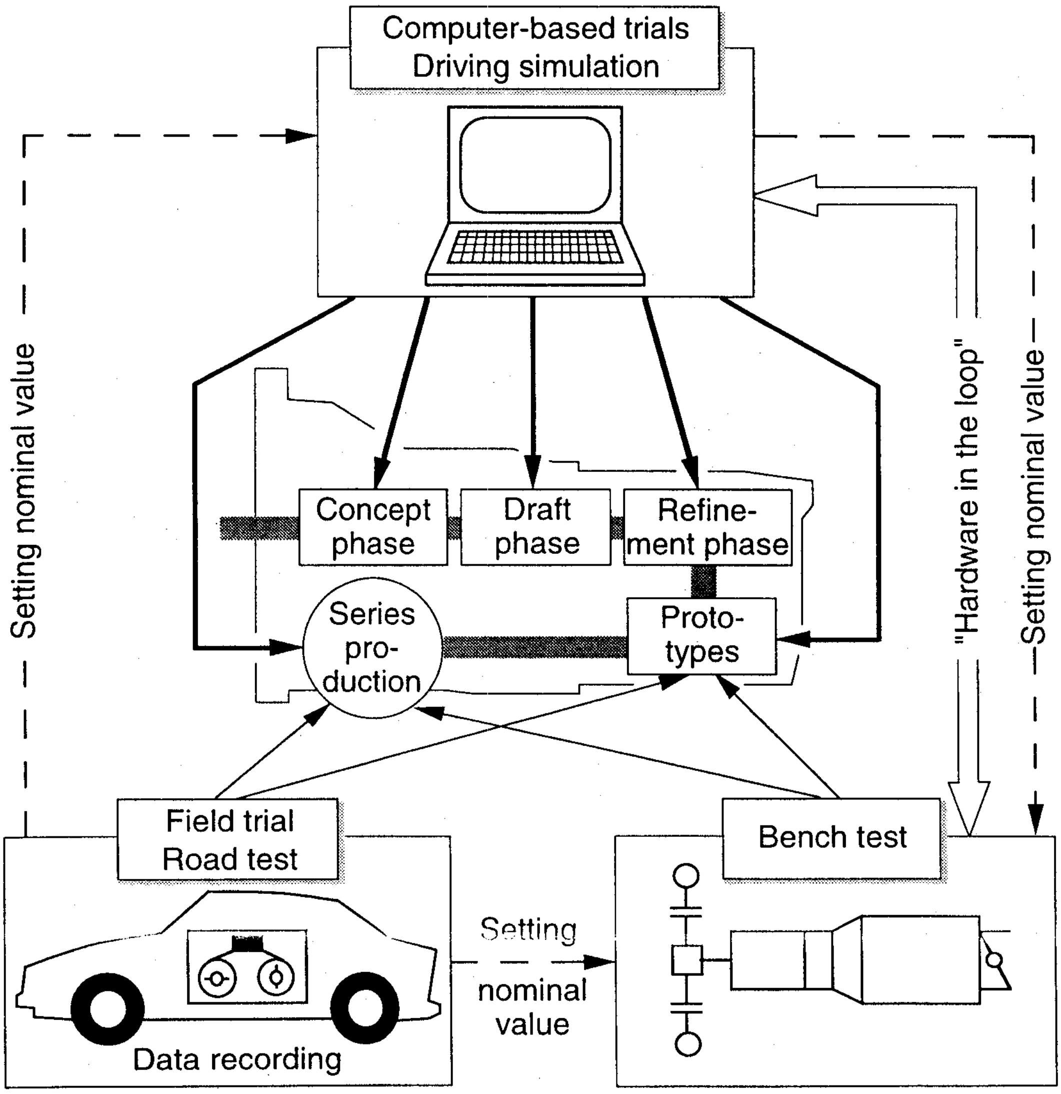


Figure 15.3. Using computerised driving simulation to develop and optimise motor vehicle power trains

Driving condition data derived from simulation can be used to specify nominal values for power-train test beds. Another increasingly popular option for developing and testing complex units at an early stage is "hardware in the loop". This technique involves including the part under investigation (for example the transmission control) in hardware form, whilst the remaining components of the driver/vehicle/road system are simulated as a numerical model on the computer [15.5].

The key advantages of driving simulation compared to road tests are [15.6]:

- reproducible conditions,
- good time/cost ratio,
- calculations can be carried out throughout the development process, and
- reduced development time with parallel product development, "simultaneous engineering".

Simulation means representing real-world phenomena on the computer. This means that not all the variables influencing the system under investigation can be fully represented in the computer model [15.6] (Figure 15.4a). In the case of driving simulation, this applies in particular to modelling the driver and the traffic. Simplifications and ideal assumptions are necessary. This "loss of reality" is necessary to achieve a viable relationship between effort and results (Figure 15.4b).

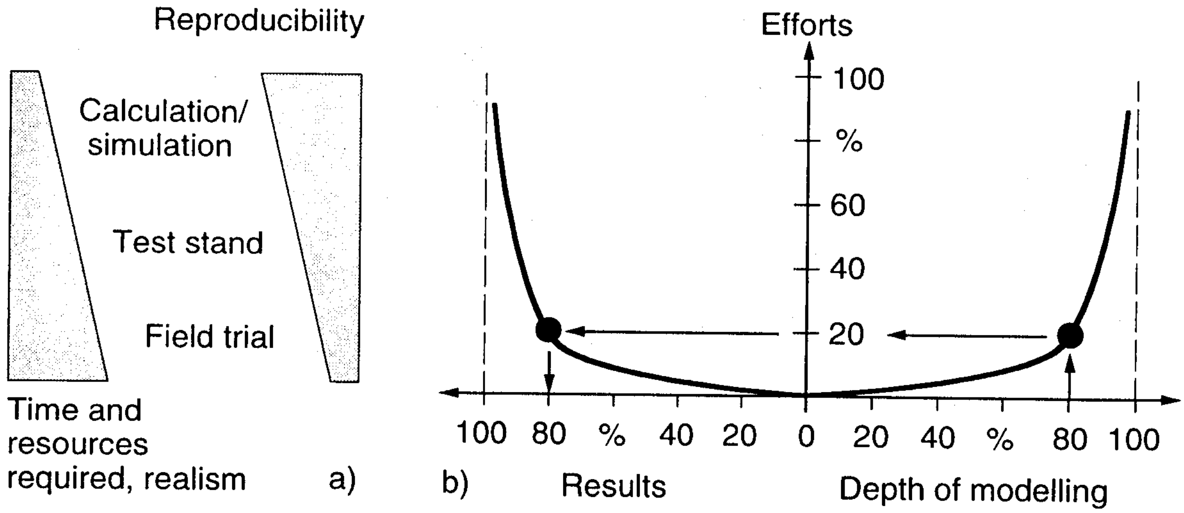


Figure 15.4. **a)** System idealisation; **b)** PARETO Principal of the simulation of technical systems [15.7]

15.1.1 Extraneous Factors

The core of driving simulation is the driver/vehicle/road control loop (Figure 15.5). The driver controls the vehicle as a control system with a view to adapting the actual speed of the vehicle as closely as possible to the desired-speed control variable. The speed desired is the speed at which the vehicle would move if there were no impinging extraneous variables. Extraneous disturbance variables influence both the driver as controller and the vehicle as control system. This includes other road users, the rules of the road, weather conditions, and the three-dimensional route profile.

The conditions in which the vehicle is used have a major impact on the vehicle as a system. The route and driving style depend on the driver, and therefore vary considerably. The design of the transmission has to take account of typical operating conditions and also extreme operating conditions.

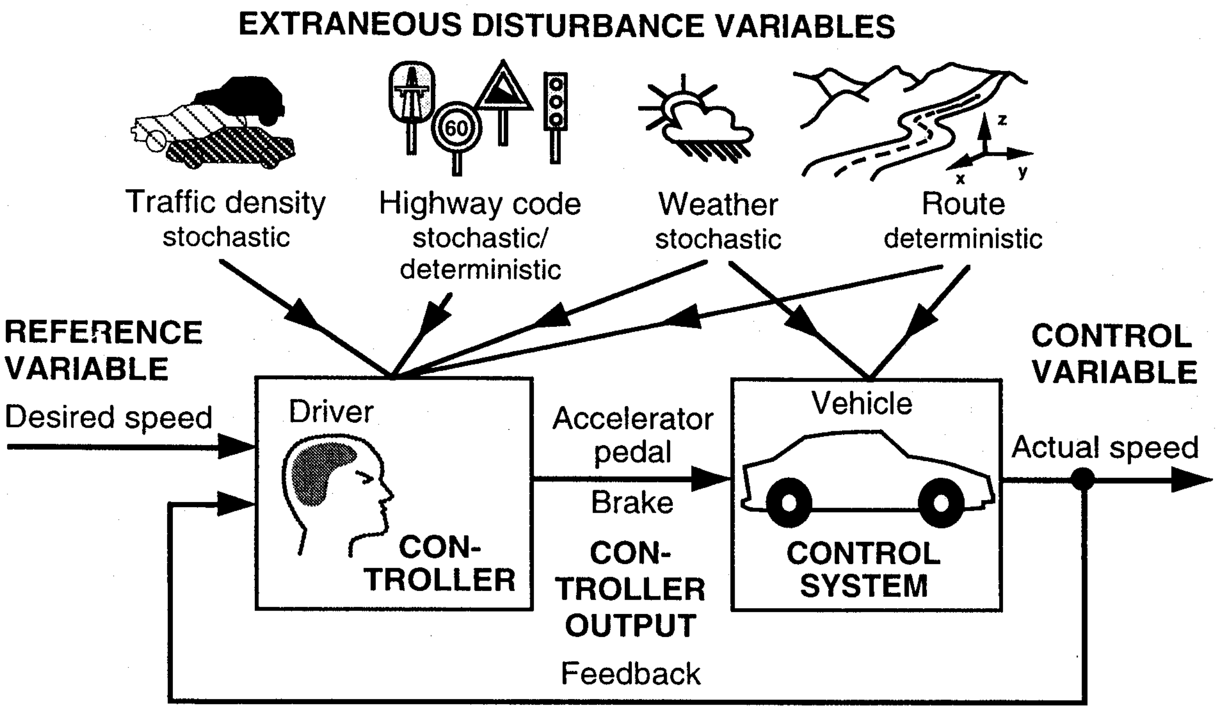


Figure 15.5. Driver-vehicle-route control loop. Impinging variables

It is therefore not sufficient for the driving simulation to take account only of standard cycles (Table 5.4), but real routes have to be stipulated. The more realistic the route specification data are, the better different operating conditions can be simulated on the computer [15.8].

The operating conditions can be recorded in the course of a *pilot run*. During the pilot run, road space information and speed are recorded as well as the three-dimensional road profile. This speed profile is called the pilot speed profile. The pilot speed takes account of the extraneous disturbance variables in effect at the time of the pilot run on the road.

Using the pilot speed as the specified speed for the driving simulation calculation is a proven procedure. Alternatively the specified speed can be arrived at using traffic simulation (microscopic driving simulation). In traffic simulation the speed of the vehicle under observation is calculated from the desired speed and the extraneous disturbance variables.

It is advisable to link sub-microscopic driving simulation, which considers the individual vehicle and its components, to microscopic driving simulation. But too many specified or set parameters and excessively complex traffic models can also be a hindrance to achieving clear results from the driving simulation calculations.

15.1.2 Route Data Set, Route Data Acquisition

It is necessary to acquire the vertical route profile for the driving simulation in order to simulate the gradient resistance. The height or

- gradient profile

of a route must therefore be known in the greatest possible detail.

- Horizontal route profile and
- road space information such as built-up areas, speed limits, carriageway widths, etc.

can be used as additional decision criteria for the driver controller, or as parameters for traffic models and traffic simulation calculations derived from them. If the recorded pilot speed profile is also used as the

- specified speed for the simulation,

the type and engine of the pilot vehicle must be similar to that of the vehicle to be simulated. The route data recording device must therefore be mobile, and usable in any vehicle without major expenditure of resources.

Some methods of recording route data are listed below.

- *Barometric height measurement*
  With height profile measuring equipment based on the measurement of barometric air pressure, the vertical route profile is recorded over the route. The analysis software enables the air pressure to be converted into altitude in metres, applying various corrections [15.9].
- *Gyroscope systems*
  Gyroscope systems are also used for three-dimensional mapping of route data [15.10]. Such inertial navigation systems provide relatively precise results.
- *Satellite-based data recording using GPS*
  The satellite based **G**lobal **P**ositioning **S**ystem (GPS) has been available for some time, enabling the three-dimensional position and the inherent speed of a vehicle in normal traffic conditions to be determined and recorded [15.8].
- *Joint positioning systems*
  Combination of two or more methods.

15.2 Driving Simulation Programs

15.2.1 Classification

The study of the sub-microscopic simulation procedures reported in the literature leads to classification into regression procedures and analytical procedures [15.6].

Regression procedures involve an attempt to provide a functional description of the complex relations of the driving process. After evaluating numerous test drives, the functional interrelations between driving parameters and route parameters are determined by regression analysis. With the functions thus determined, frequency distributions for speed and traction are compiled as a function of data relating to the driver, vehicle, route and other environment data. The overall distribution of all vehicle operating conditions is derived in simulation calculations from summing the frequency of the traction and speed classes arising.

The engine operating conditions can be determined from the vehicle operating conditions by means of gear allocation, and thus also fuel consumption. Gear allocation can be a function of driver mentality. Regression processes are to be regarded as historic, and not well suited to simulating new and alternative power-train structures.

The second type of simulation process is analytical methods. They are based on the fundamental equations of the dynamics of vehicle movement, Equations 3.18–3.22. They are also referred to in the literature as geodetic or dynamic driving simulation. They have a direct time/distance relation. The movement equations are integrated along individual section intervals in accordance with the nominal speed. The real operating conditions are determined by pilot runs.

The main advantages of the dynamic driving simulation method are its versatility in terms of applications and model extension.

Modern driving simulation programs are based on the "cause and effect principle" [15.11]. The driver controller reacts to deviations of the actual speed from the desired speed by altering the position of the accelerator. The "software driver" calculates from the accelerator pedal position and the engine performance map to the drive wheels. Simulation programs operating on the reverse principle are easier in terms of driver regulation, much faster in terms of calculation speed, but also less versatile. Calculation proceeds in this case from required power at the wheels resulting from the driving resistance, to the operating point in the engine performance map.

15.2.2 Modular Construction

To enable modular construction for a driving simulation program on the cause/effect principle, a suitable approach is to link the components using spring/damper elements, which may be regarded as the state of the art [15.11–15.15].

The spring/damper elements also enable vibration investigations to be carried out. Whilst rough estimates of spring stiffness and damping values are sufficient for longitudinal dynamic applications, these values must then be specified exactly.

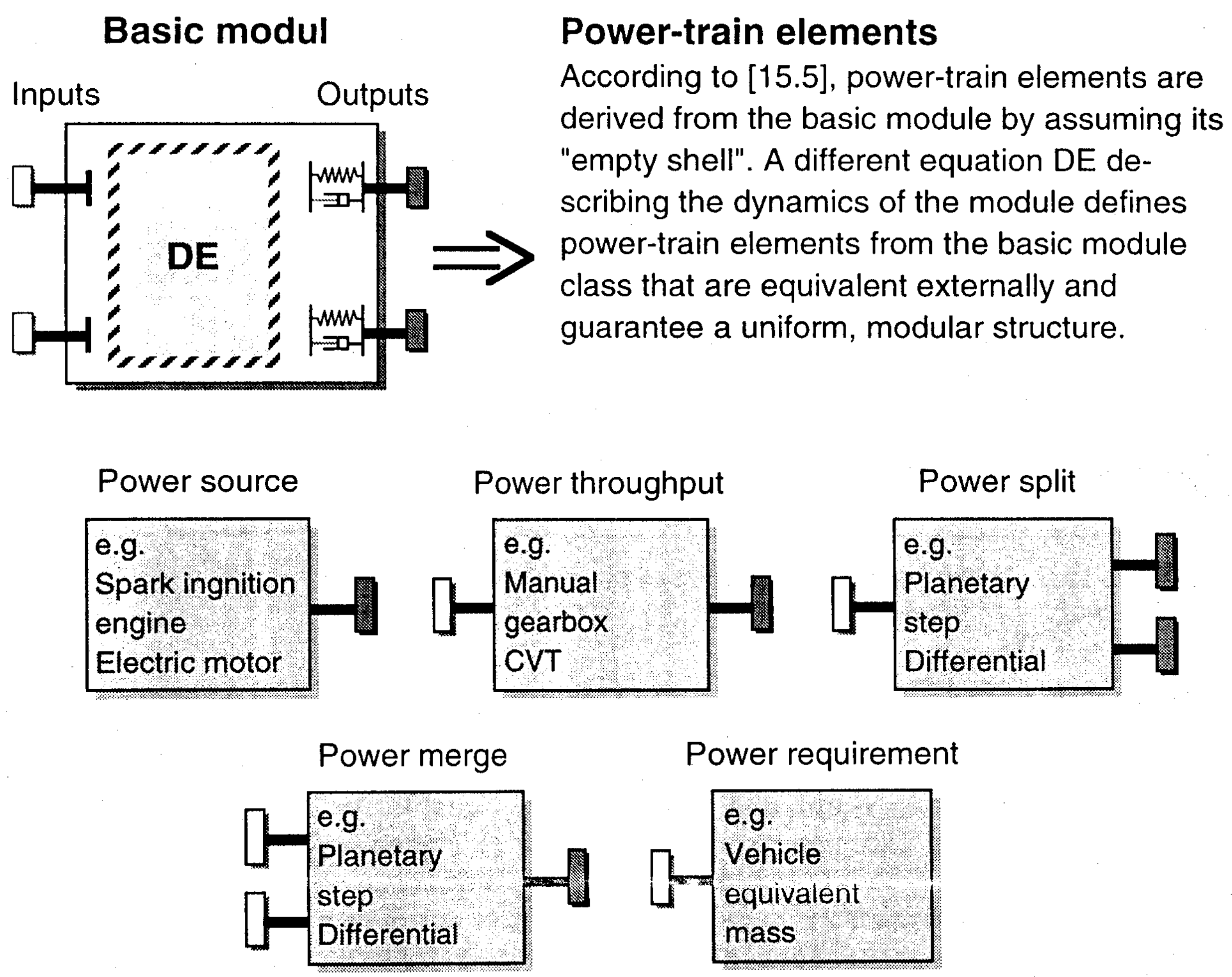


Figure 15.6. Basic module and resultant power-train elements. The two inputs and outputs normally available are not used in each module

Figure 15.6 shows power-train elements derived from an empty base module shell. The dynamics of the power-train modules defined by differential equations is integrated by the explicit EULER method proven in vehicle dynamics [15.16], which states as follows:

$$z^{k+1} = z^k + h \cdot f\left(z^k, t^k\right),$$
$$t^{k+1} = t^k + h. \tag{15.1}$$

Where k is the simulation interval; z is the state variable; h is the interval increment; t is time. The advantages are simplicity of programming and the fact that only one function call is necessary per interval.

Power-train simulation as thus presented represents a modular problem. The individual components of the power train can be regarded as modules linked by precisely defined interfaces. Object-oriented programming languages are therefore a suitable tool for resolving the engineering problems arising [15.17].

Figure 15.7 gives the example of a parallel hybrid power train comprising the power-train modules described. Losses arising from pumps, accessories, etc., and the efficiency of the various power-train elements must be taken into account.

All information on engine speed, torque and losses can be read at the module interfaces. Depending on the object of the investigation, the power-train modules can be shown in finer resolution, or summarised in larger units.

The driver model requires great care, since the driver has a major impact on energy consumption, emissions, performance and load profiles.

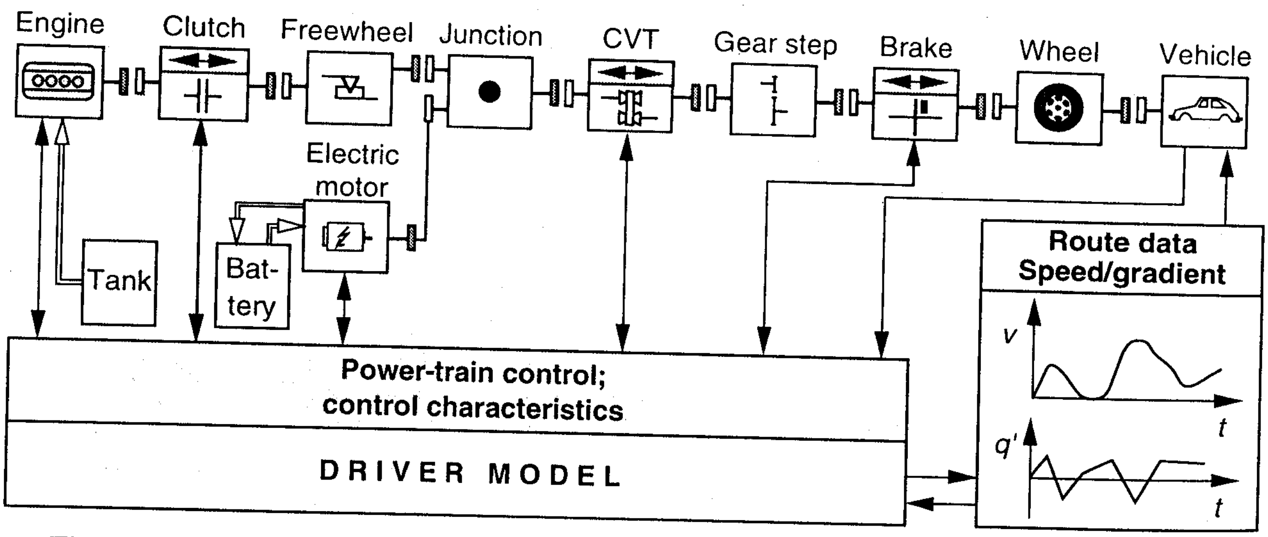


Figure 15.7. Hybrid drive line made up of the power-train modules

15.3 Applications of Driving Simulation

Computer-based driving simulation is particularly suitable for comparative investigations, which is to say parameter studies. Thus for example the broad field of possible hybrid drive structures under consideration can be reduced to a few promising alternatives by simulation calculations in the concept phase of development [15.15].

A typical application of simulation calculations for products which are already in production is the selection of the rear axle ratio of buses [15.18]. Driving simulation makes it possible to determine the optimum final drive ratio depending on typical operating conditions (route topography, number of passengers, etc.).

Example of Parameter Study

The effect of variator pressure control on fuel consumption in the case of chain converter CVT's is given as an example of parameter study using driving simulation. As already discussed in Sections 5.3.4, 6.6.4 and 12.3.1, the efficiency of chain variators has a decisive influence on fuel consumption. The potential for optimising the chains themselves is largely exhausted. The losses arising from the variable displacement/contact pressure pump under part load can still be significantly reduced by judicious control of the variator contact pressure.

The contact pressure of the variator discs must be sufficient to transmit the power reliably, i.e. in all operating situations, without the chain slipping. But contact pressure force in excess of what is physically necessary reduces the efficiency of the power train.

Figure 15.8 shows the effect of the minimum contact pressure force on the overall efficiency of the power train. The minimum contact pressure is the minimum pressure that needs to be exerted on the variator discs, even if a lower contact pressure would be adequate to transmit the power. Figure 15.9 shows the effect of the minimum contact pressure on the fuel consumption of the vehicle concerned.

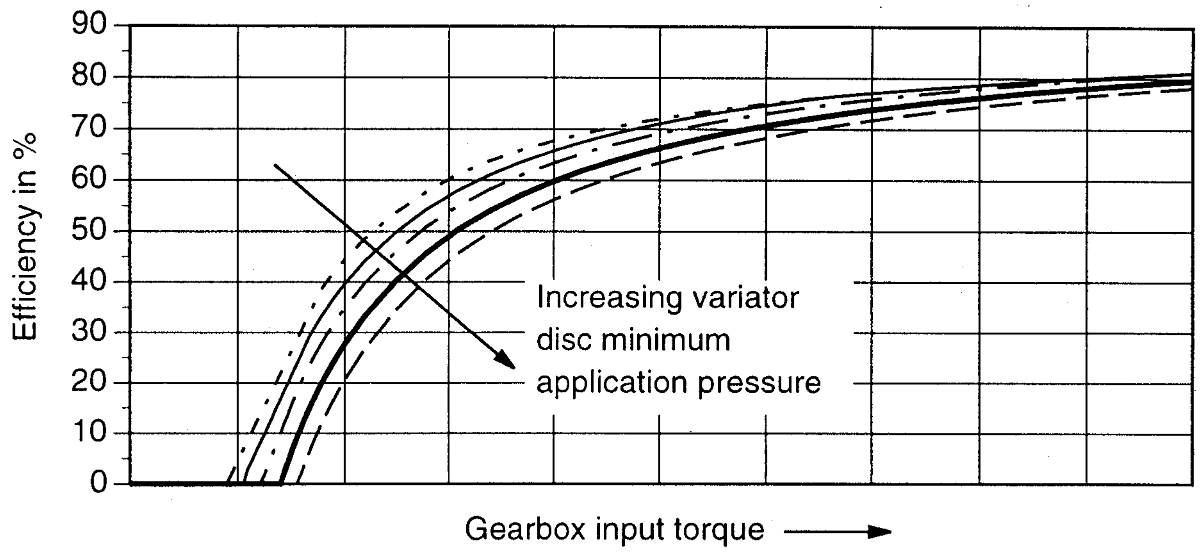


Figure 15.8. Overall power-train efficiency as a function of the minimum variator disc pressure

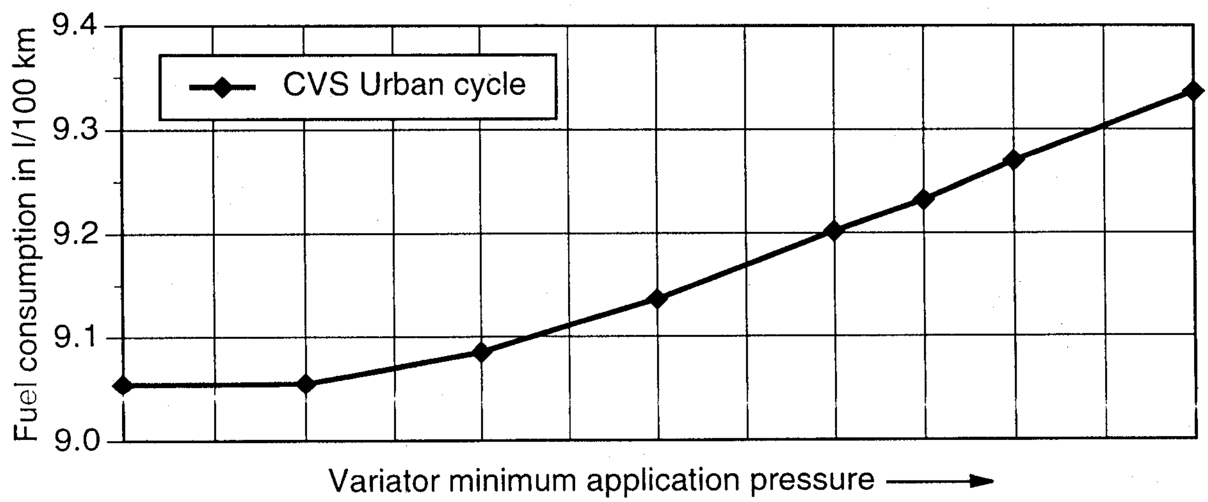


Figure 15.9. Fuel consumption of an engine/vehicle/transmission combination for the CVS urban cycle as a function of minimum variator disc contact pressure

These simulation calculations illustrate the potential of optimising the pressure control. A major advantage of "test drives on the computer" is the fact that all relevant variables can be detected, stored and analysed. The test drive is transparent.

16 Reliability and Testing of Vehicle Transmissions

The automobile and its components are an outstanding example of complex technologies combined with a high degree of reliability

New legislation (e.g. that relating to product liability and environment protection), shorter innovation cycles, and increased customer expectations require ever greater efforts to produce reliable, safe products. To achieve this, certain ground rules need to be observed, even during the development of the product. The most important ground rules to be observed during the development of vehicle transmissions to ensure reliability are set out below:

- precise specification,
- as few component parts as possible,
- elimination of risk parts,
- interchangeability of wearing parts,
- computer simulation of practical use,
- investigating the dynamic behaviour of the power train,
- early component tests,
- comprehensive test bed and road testing,
- most rigorous quality assurance in-house and with subcontractors,
- random inspection of production.

An automotive gearbox cannot be regarded as a single component. It must rather be regarded as a complex system comprising many different components. The individual components are accordingly subjected to the most varied influences and stresses. The reliability of the vehicle transmission system is therefore determined by numerous influencing variables. These influencing variables can best be divided into two categories, "internal" and "external" (Figure16.1).

Just as there are numerous influences, so there are also several criteria for defining a reliable vehicle transmission, the most important being:

- *The transmission must have a high average service life expectancy.*
- *The transmission must have practically no premature failures.*

Based on these requirements, two basic measures to improve the reliability of automotive transmissions can be derived:

- *The permissible stressing of the weak elements must be increased and the range of these permissible stresses narrowed down.*
- *Strict quality control measures must be implemented to minimise production and assembly errors.*

In order to take account of the above requirements in development, it is necessary for the design engineer to have a number of methods available to him for calculating the reliability of components and complete component systems, or at least estimating them. This chapter discusses the necessary mathematical and statistical principles of reliability theory necessary for this purpose.

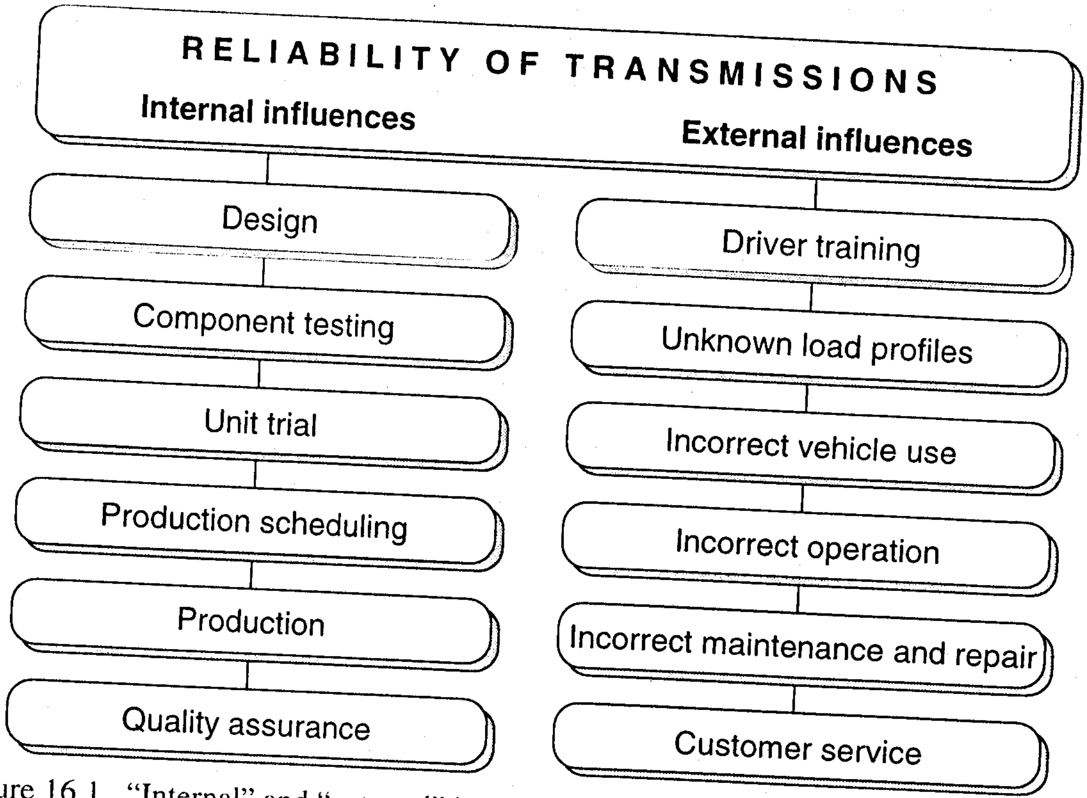


Figure 16.1. "Internal" and "external" influences on the reliability of a transmission

16.1 Principles of Reliability Theory

The principles of reliability theory are set out below. The reader is referred to the relevant literature for more detailed treatment of this topic [16.1].

16.1.1 Definition of Reliability

It must be possible to define the variable "reliability" qualitatively and quantitatively for the purpose of objective assessment and calculation. The definition of technical reliability in the VDI guideline 4001 [16.2] is:

RELIABILITY
is the probability that a product will not fail during a defined period under given functional and environmental conditions.

Reliability is thus a time-related probability of not failing. It should be noted that in addition to the period considered or the load cycles endured, the precise functional and environmental conditions are essential to determine reliability. Reliability indications are always given for particular, precisely defined operating conditions.

16.1.2 Statistical Description and Representation of the Failure Behaviour of Components

The service life t, describing the failure behaviour of a component is not to be interpreted as a variable to be determined discretely. It is rather a random variable that is subject to a particular distribution [16.3].

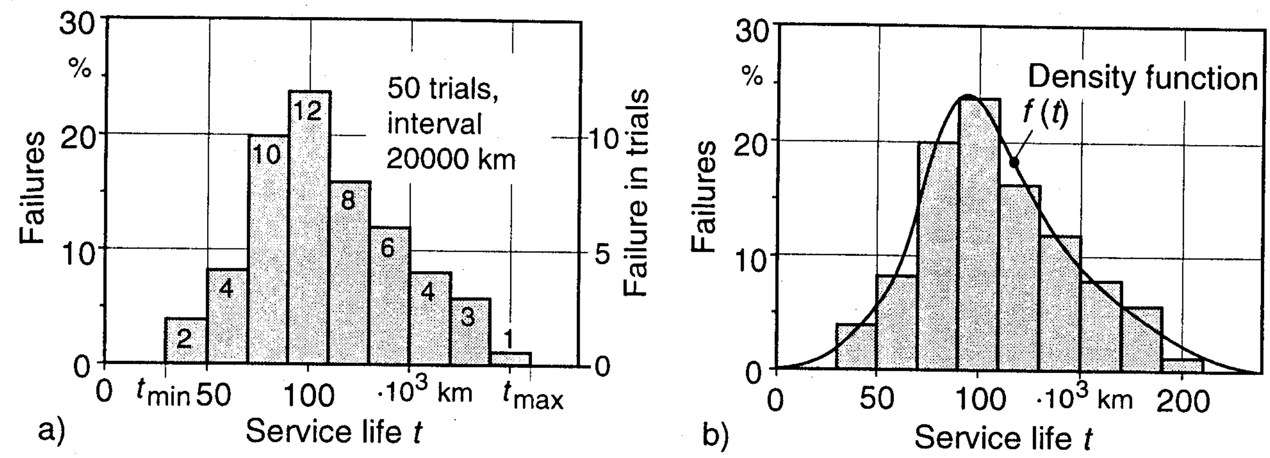


Figure 16.2. **a)** Histogram of failure frequency of a durability test; **b)** Failure frequency and density function $f(t)$

Figure 16.2a shows a histogram of failure times for a service fatigue life test. This show not only the dispersion, but also the different frequency of occurrence of the failure times. Figure 16.2a shows how component failure times can be displayed within a service fatigue life test. In the histogram the range of dispersion $t_{max} - t_{min}$ is divided into an appropriate number of intervals, and the defects observed allocated to the intervals. The height of the bars then represents the total number of defects occurring in that interval. As the interval width reduces, the profile of the histogram can be approximated by the curve of the density function $f(t)$ (Figure 16.2b).

The causes of the distribution are shown qualitatively in Figure 16.3. A defined value cannot be allocated either to the material (the resistance) or to the stress. Both values are distributed over a particular range. Different material characteristic values result from different samples. Then there are also the differences arising from processes the material undergoes (e.g. cold forming, resulting from permissible tolerances). Different stresses can for example be caused by different operating conditions.

If the failures noted are added with consecutive interval number, this results in the histogram of cumulative frequency shown in Figure 16.4a. The outline of this histogram can in turn be approximated by a smooth curve with reduction of the interval width. This curve is referred to in statistics as the distribution function $F(t)$ (Failure), and in reliability theory as the probability of failure $F(t)$.

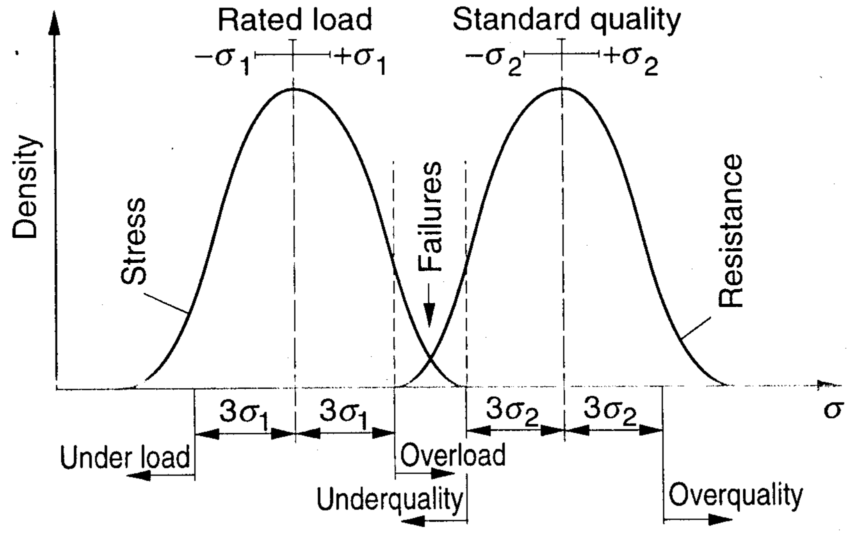


Figure 16.3. Dispersion of stress and resistance

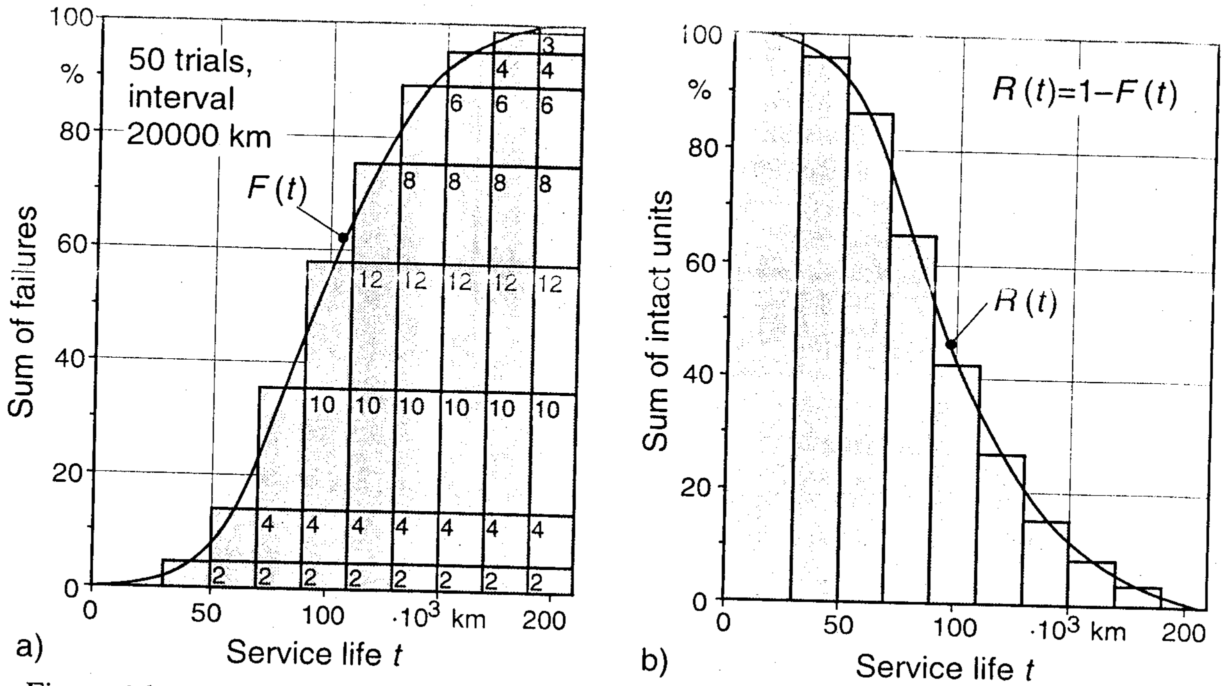


Figure 16.4. **a)** Histogram of the cumulative and distribution function or probability of failure $F(t)$; **b)** Histogram of the probability of survival or reliability $R(t)$

Between the density function $f(t)$ and the probability of failure $F(t)$ the following relations apply

$$F(t) = \int f(t)\,\mathrm{d}t \qquad \text{or} \qquad f(t) = \frac{\mathrm{d}F(t)}{\mathrm{d}t} \; . \tag{16.1}$$

To represent the units that are still intact, the probability of survival $R(t)$ (**R**eliability) is used (Figure 16.4b). Since the probability of failure $F(t)$ describes the total of defective parts, the probability of survival $R(t)$ is derived as a complement of $F(t)$ as 1,

$$R(t) = 1 - F(t) \; . \tag{16.2}$$

The probability of survival $R(t)$ is sometimes also referred to as reliability $R(t)$ in reliability theory.

A further statistical variable often used to characterise failure behaviour is the failure rate $\lambda(t)$. To determine this failure rate $\lambda(t)$, failures at a given point in time t or in a time interval dt is related to the number of units that are still intact

$$\lambda(t) = \frac{\mathit{Failures}}{\mathit{Intact\ units}} \; . \tag{16.3}$$

Since the density function $f(t)$ describes the failure density and the probability of survival $R(t)$ describes the intact units, the failure rate (λ rate) $\lambda(t)$ can be derived as the quotient of these two functions

$$\lambda(t) = \frac{f(t)}{R(t)} \; . \tag{16.4}$$

The failure rate $\lambda(t)$ can be determined as a measure of the failure risk of a part if it has already survived up to this point in time t.

If the failure behaviour of a product from its production to the end of its life is considered, a typical curve profile emerges (Figure 16.5). Because of its shape, this is called a *bathtub curve*. There are three distinct sections: section 1 relates to early failures, section 2 to random failures, and section 3 to failures due to wear and fatigue.

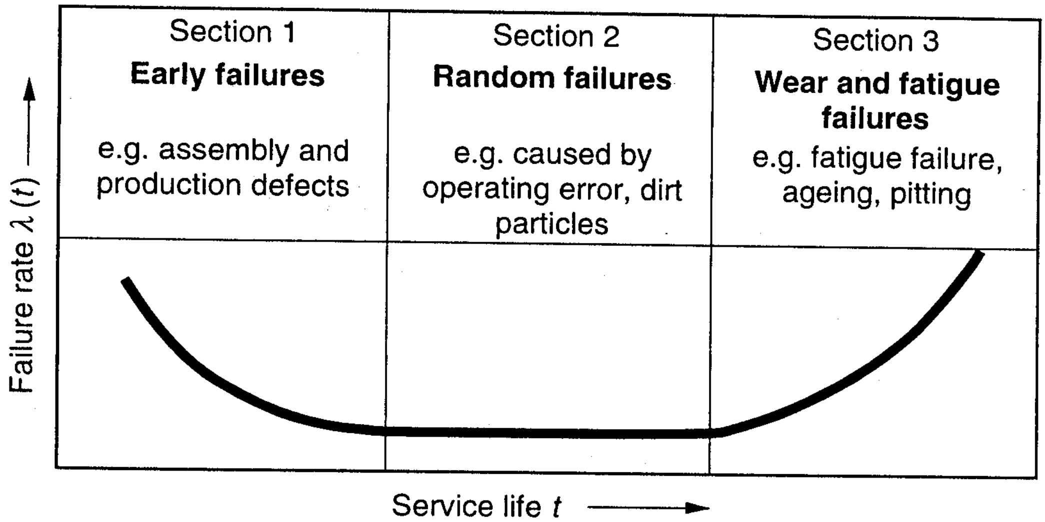


Figure 16.5. Bathtub curve

Section 1: Early Failures

Section 1 is characterised by a declining failure rate. The risk of a part failing decreases with time. These early failures are caused mainly by production and assembly errors. Early failures can only be countered by extremely rigorous production and quality assurance.

Section 2: Random Failures

Random failures in section 2 have a constant failure rate. The risk failure of a part is thus always the same. The risk is usually relatively low. These random failures are caused e.g. by operating errors or dirt particles.

Section 3: Failures Due to Wear and Fatigue

In section 3, failures due to wear and fatigue, the failure rate increases sharply. The risk of failure increases for a part as its service life increases. The defects occurring here are for example caused by fatigue fracture, ageing, pitting or wear. This section is the most interesting for the design engineer, since the service life of a part is largely determined by this section. It can therefore be substantially improved by taking special account of the possible causes of failure and designing the parts accordingly – lifetime calculation.

The bathtub curve applies not only to individual components but also arises with complete systems.

Since the four functions $f(t)$, $F(t)$, $R(t)$ and $\lambda(t)$ are interdependent, the other three unknown functions can always be determined for a known function. It is therefore sufficient to determine only one of these functions.

16.1.3 Mathematical Description of Failure Behaviour using the Weibull Distribution

In the previous section we saw how failure behaviour can be represented by various statistical functions. Of particular interest however is the precise profile of these functions for a specific case, and how the curve can be described analytically. For this purpose the failure functions derived empirically are to be replaced or approximated by curves that can be described analytically. The service fatigue life distributions used for this purpose are considered in this section.

1/ Normal Distribution

The best-known of these service fatigue life distributions is the *normal distribution*. It has as density function $f(t)$ the well known "bell curve" which is completely symmetrical around a central value. The failure density is highest at the central value. Thus basically only *one* type of failure behaviour can be described. The normal distribution is however frequently used in reliability theory.

2/ Exponential Distribution

The density function $f(t)$ of the *exponential distribution* decreases monotonically as an inverse e-function of an initial value. Thus only one failure behaviour can be described, in which initially a high number of failures is observed which then continuously decreases. In addition to the continuously decreasing density function, the constant failure rate λ is a key characteristic of this distribution. That means the risk of failure is unrelated to time. Here too it can be noted that the exponential distribution is mainly only suitable for describing a specific type of failure behaviour.

The exponential distribution is frequently used in electrical engineering and aeronautics, whilst in machine building the service fatigue life distributions most frequently used are the *log normal distribution* and the *Weibull distribution*.

3/ Log Normal Distribution

The logarithmic normal distribution is derived from the normal distribution. The random variable t is used in the logarithmic form $\ln t$. This means that the logs of the failure times follow a normal distribution.

With the log normal distribution similar curves can be achieved in similar variety as with the Weibull distribution discussed below. The mathematical use of the log normal distribution is more difficult than the Weibull distribution, since it cannot be solved analytically, but only numerically.

4/ Weibull Distribution: Basic Concept and Equations

The Weibull distribution is capable of effectively describing very different failure behaviour. This is shown most clearly by the density functions represented with the Weibull distribution (Figure 16.6). The formulae and relations of the Weibull distribution are given in Table 16.1.

4.1/ Two-Parameter Weibull Distribution

The density function changes distinctly as a function of a parameter of the distribution – form parameter b. For small values of b ($b < 1$) the failures are described similarly to the exponential distribution, i.e. there is a very high number of failures at the beginning, which then gradually decreases (Figure 16.7). The form parameter $b = 1$ gives exactly an exponential distribution. For form parameter $b > 1$ the density function always starts with $f(t) = 0$, then with increasing service life reaches a maximum and finally levels off. The maximum of the density function is displaced for increasing values of b more and more towards larger service life values. With the form parameter $b \approx 3.2$ to 3.5, the distribution is approximately normal.

Table 16.1. Formulae and designations of the Weibull distribution [16.1]

Two-parameter Weibull distribution

Probability of survival	$R(t) = \mathrm{e}^{-\left(\frac{t}{T}\right)^{b}}$	(16.5)
Probability of failure	$F(t) = 1 - \mathrm{e}^{-\left(\frac{t}{T}\right)^{b}}$	(16.6)
Density function	$f(t) = \frac{\mathrm{d}F}{\mathrm{d}t} = \frac{b}{T}\left(\frac{t}{T}\right)^{b-1} \mathrm{e}^{-\left(\frac{t}{T}\right)^{b}}$	(16.7)
Failure rate	$\lambda(t) = \frac{f(t)}{R(t)} = \frac{b}{T}\left(\frac{t}{T}\right)^{b-1}$	(16.8)

Three-parameter Weibull distribution

Probability of survival	$R(t) = \mathrm{e}^{-\left(\frac{t-t_0}{T-t_0}\right)^{b}}$	(16.9)
Probability of failure	$F(t) = 1 - \mathrm{e}^{-\left(\frac{t-t_0}{T-t_0}\right)^{b}}$	(16.10)
Density function	$f(t) = \frac{\mathrm{d}F}{\mathrm{d}t} = \frac{b}{T-t_0}\left(\frac{t-t_0}{T-t_0}\right)^{b-1} \mathrm{e}^{-\left(\frac{t-t_0}{T-t_0}\right)^{b}}$	(16.11)
Failure rate	$\lambda(t) = \frac{f(t)}{R(t)} = \frac{b}{T-t_0}\left(\frac{t-t_0}{T-t_0}\right)^{b-1}$	(16.12)

Parameters

- t: Statistical variable (stress time, load cycle, operations, ...)
- T: Characteristic service life, "position parameter". Where $t = T$, $F(t) = 63.2\%$ or $R(t) = 36.8\%$.
- b: Form parameter or failure gradient. This determines the shape of the curve.
- t_0: Time without failure. The parameter t_0 determines the time when the failures start. This is a time shift along the t axis.

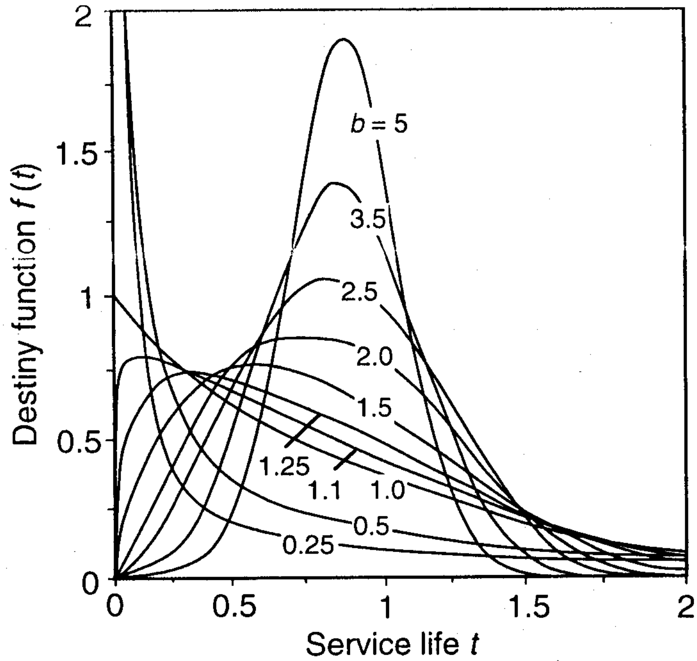


Figure 16.6. Density functions $f(t)$ of the two-parameter Weibull distribution for various form parameters b (Characteristic service life $T = 1$)

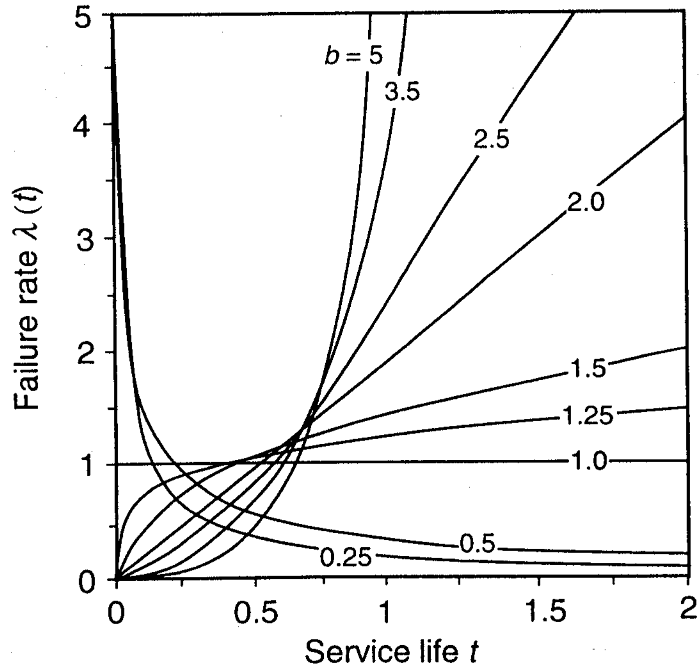


Figure 16.7. Failure rates $\lambda(t)$ of the two-parameter Weibull distribution for various form parameters b (Characteristic service life $T = 1$)

The different failure rates of the two-parameter Weibull distribution in Figure 16.7 can be divided into three sections identical with the sections of the bathtub curve in Section 16.1.2:

- $b < 1$: The failure rates decrease as service life increases. This relates to early failures.
- $b = 1$: The failure rate is constant. The form parameter $b = 1$ is thus suitable for describing random failures in section 2 of the bathtub curve.
- $b > 1$: The failure rates increase distinctly as service life increases. Values of b greater than 1 thus relate to failures attributable to wear and fatigue.

The Weibull distribution can be sub-divided into a two-parameter and a three-parameter distribution (Table 16.1). The two-parameter Weibull distribution has as its parameters the characteristic service life T (position parameter) and the form parameter b. The characteristic service life T is assigned the failure probability $F(T) = 63.2\%$ ($R(T) = 36.8\%$).

The form parameter b is a measure of the dispersion of failure or for the form of the failure density. Failures are always described starting from the point in time $t = 0$ in the two-parameter Weibull distribution.

4.2/ Three-Parameter Weibull Distribution

The three-parameter Weibull distribution has in addition to the parameters T and b an additional parameter, the failure-free time t_0. In the case of failures due to wear and fatigue, the failure-free time t_0 is based on a certain time being needed for failures to arise and spread. Without this assumption, failures due to wear, fatigue, ageing, etc. could arise even after a very brief period of operation. This is however contradicted by general experience and expectations. This third parameter can thus serve to describe failures which only start after a point in time t_0 [16.4].

The two-parameter Weibull distribution can thus be represented as a simplification of the three-parameter Weibull distribution in which $t_0 = 0$ for the failure-free time. Similarly the three-parameter Weibull distribution can however also be derived from the two-parameter Weibull distribution by means of a time transformation. For this purpose only the failure time t and the characteristic service life T has to be replaced by $t - t_0$ and $T - t_0$. The full formulae are also shown in Table 16.1.

The equations of the Weibull distribution contain the statistical variable t in related form t / T or $(t - t_0) / (T - t_0)$. For the point in time $t = T$ the quotient always becomes equal to 1 and the probability of failure is calculated as:

$$F(t) = 1 - e^{-1} = 0.632 \ . \tag{16.13}$$

This assigns a failure probability $F(T) = 63.2\%$ or a probably of survival $R(T) = 36.8\%$ to the characteristic service life T. The characteristic service life T can thus be treated as a characteristic value, somewhat as with the median value, for which $F(t) = 50\%$. A further important characteristic is the B_x service life. This is the service life for which the probability of failure of the element under investigation is x%.

The average value t_m of the Weibull distribution, which is very rarely used, can be derived using the so-called "gamma function":

$$t_m = T \cdot \Gamma\left(1 + \frac{1}{b}\right) \ . \tag{16.14}$$

The function values of the gamma function are for example listed in [16.5].

Graphic Representation of the Weibull Distribution

The failure probabilities $F(t)$ follow an S-shaped curve. Using special Weibull probability paper it is possible to draw the functions $F(t)$ of the *two-parameter Weibull distribution as straight lines* (Figure 16.8). This enables failure behaviour to be shown in a simple graphic form. There are also advantages in evaluating tests, since this enables the experimental values to be entered along the straight line. The abscissa is logarithmically divided, whilst the ordinate has a double logarithmic scale:

$$x = \ln t \ , \tag{16.15}$$

$$y = \ln\left[-\ln\left(1 - F(t)\right)\right] \quad \text{or} \quad y = \ln\left[-\ln R(t)\right] \ . \tag{16.16}$$

Every two-parameter Weibull distribution can thus be represented as a straight line in the Weibull probability network (see Figure 16.8).

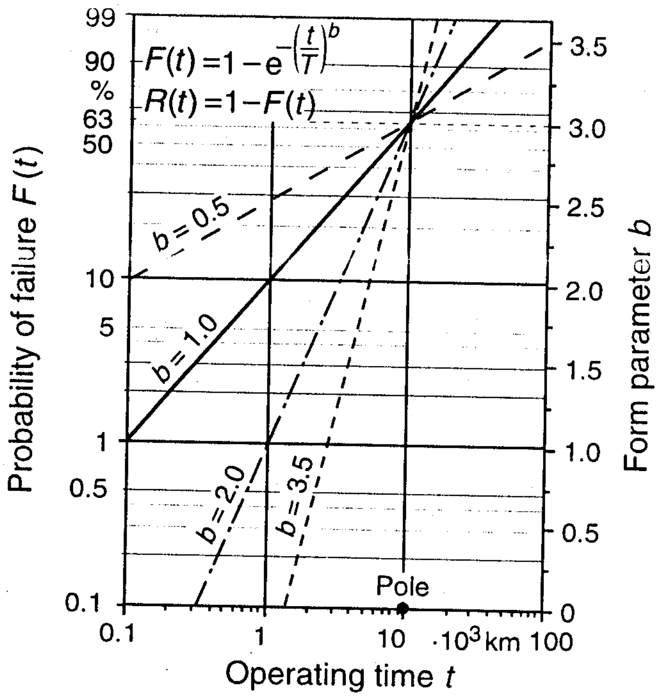


Figure 16.8. Weibull probability paper. Failure curves of the two-parameter Weibull distribution (Weibull lines) with different form parameters *b*

The gradient of the straight lines in the probability network is a direct measure of the form parameter *b*. The form parameter *b* can be read off on the right ordinate in Figure 16.8, if the straight line is shifted parallel through the pole *P*. The position of the pole *P* and the division of the linear ordinate for the form parameter *b* can be determined with Equations 16.15 and 16.16:

$$b = \frac{\Delta y}{\Delta x} = \frac{\ln\left[-\ln\left(1 - F_2(t_2)\right)\right] - \ln\left[-\ln\left(1 - F_1(t_1)\right)\right]}{\ln t_2 - \ln t_1} \, . \tag{16.17}$$

A *three-parameter Weibull distribution does not produce a straight line* on Weibull probability paper, but a *curve* (Figure 16.9a), although a three-parameter Weibull distribution can also be drawn as a straight line if the failure times $(t - t_0)$ corrected by t_0 are removed on the abscissa. This time transformation serves to return the three-parameter Weibull distribution to a two-parameter Weibull distribution (Figure 16.9b).

16.1.4 Reliability with Systems

Engineering products have to be regarded as complex systems consisting of several components. The failure behaviour of the individual components can, as described in the previous chapter, be represented by graphic means such as a Weibull distribution with the parameters *b*, *T* and t_0. The failure behaviour of the total system is derived using a system theory that links the reliabilities of the elements in a suitable manner. One of these system theories is the BOOLEAN theory, the chief premises of which are:

- a system is "non-repairable" (first system failure terminates the system's service life),
- components can only accept the two states failed/not failed,
- components are "independent" (the failure performance of a component is not affected by the failure behaviour of the other components).

Under these conditions many engineering products can be treated using the Boolean theory. This theory is used exclusively below.

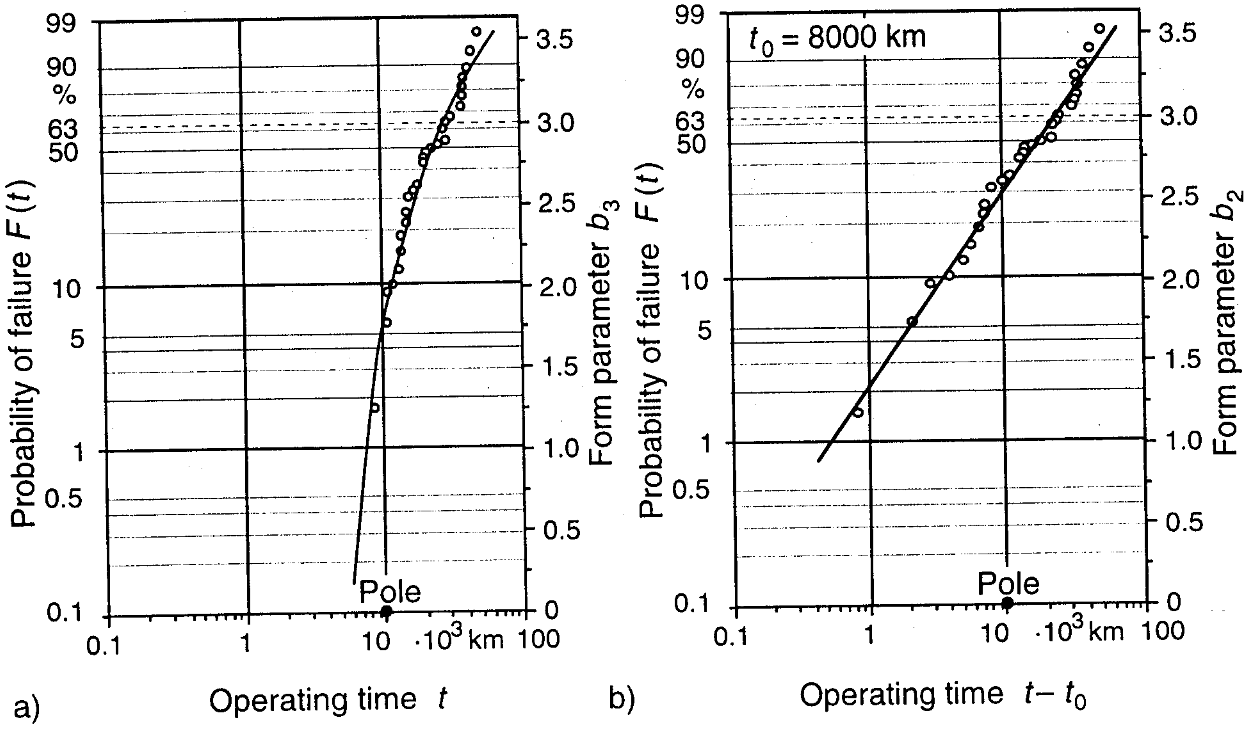


Figure 16.9. Three-parameter Weibull distribution on Weibull probability paper. **a)** Original values and failure curve (Weibull curve) of the three-parameter Weibull distribution; **b)** Three-parameter Weibull distribution with the failure times $(t - t_0)$ reduced by t_0

Reliability diagrams can be constructed with the components, indicating how the failure of one element affects the system as a whole. The links between the schematic input (I) and output (O) (Figure 16.10) represent the possibilities for the system's functionality.

The system is thus functional precisely when there is at least one input – output link in the reliability diagram between input and output, on which all the components shown are intact. In a serial structure (Figure 16.10a) the failure of any component leads to failure of the whole system. In a parallel structure (Figure 16.10b) the system only fails when all the elements have failed.

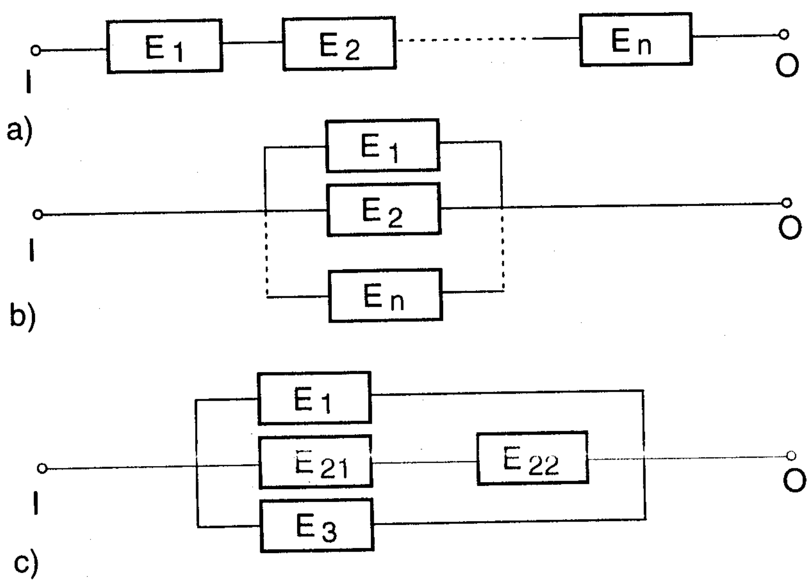


Figure 16.10. Basic structures of reliability diagrams. **a)** Serial structure; **b)** Parallel structure; **c)** Mixed structure; *I* Input; *O* Output; *E* Element

It should be noted that the structure of the reliability diagram does not relate to the mechanical structure of a design. For example a component can quite possibly occur at several points in the reliability diagram. Almost all the systems used in machine building have serial structures, since the construction of parallel redundancies is complex and expensive.

The reliability of a serial system is calculated according to the product law of survival probabilities

$$R_S(t) = R_1(t) \cdot R_2(t) \cdot R_3(t) \ldots \quad \text{or} \quad R_S(t) = \prod_{i=1}^{n} R_i \; . \tag{16.18}$$

Since the survival probability of each system element $R_S(t) \leq 1$. The result for system reliability is always a value less than/equal to the reliability of the worst component.

With parallel systems (Figure 16.10b), the system's reliability is derived from the formula

$$R_S(t) = 1 - (1 - R_1(t)) \cdot (1 - R_2(t)) \cdot (1 - R_3(t)) \ldots$$

$$\text{or} \quad R_S(t) = \prod_{i=1}^{r} (1 - R_i(t)) \; , \tag{16.19}$$

with the redundancy level r of the system.

16.1.5 Availability of Systems

Reliability describes the survival probability of components or whole systems until the *first* failure. In the case of *repairable systems* the system can be returned to a functional condition by a repair. Failure and subsequent repair can be frequently repeated in the case of repairable systems. The term "availability" was introduced for such systems. Availability $A(t)$ relates to the probability of a system being in a functional condition at a given time. The stationary availability (long-term availability) is calculated using the formula

$$A(t) = \frac{MTBF}{MTBF + MTTR} \; , \tag{16.20}$$

where $MTBF$ is the Mean Time Between Failures and $MTTR$ is the Mean Time to Repair. Since the reciprocal of $MTBF$ is equal to the failure rate λ, this formula can only be used when it is anticipated that the failure rate will be constant over the whole period under consideration.

The MARKOFF model is frequently used as a system theory to describe state probability [16.6, 16.7]. This model can record the state probability of a repairable system at any desired point in time. The state conditions of the system are described by linear differential equations which in most cases can no longer be analytically solved, but have to be solved numerically using a computer.

If the reliability or availability of systems cannot be determined analytically, they can be determined using the Monte Carlo method as a simulation model. The Monte Carlo method is sometimes the only practically accessible method for investigating complex systems [16.3, 16.8, 16.9].

16.2 Reliability Analysis of Vehicle Transmissions

The main purpose of reliability assurance is to determine the anticipated failure behaviour of a product at the development stage, or to forecast it. Such forecasts are only possible for fatigue and wear failures, i.e. for section 3 of the bathtub curve (Figure 16.5).

In order to be able to dispense with some of the extensive and time-consuming tests, calculation methods are used which are based on the principles of probability theory described in the previous sections. A reliable prognosis can only be achieved if the failure behaviour of the individual components is known in sufficient detail.

The procedure illustrated in Figure 16.11 [16.3, 16.10] has proved its value in determining system reliability. In system analysis, first all components and their functions are determined. In order to guarantee a complete analysis, it is usually appropriate to divide up the components into groups according to their function or design. In the following second stage, qualitative reliability analysis, the system elements relevant to reliability, and their effect on the functionality of the system, are determined and evaluated. In the final stage, quantitative reliability analysis, the failure behaviour of the system is determined with the principles of probability theory discussed in the preceding stages.

These three stages of reliability analysis are examined below using examples from vehicle transmission engineering.

16.2.1 System Analysis

Here the product is initially delineated as a system within its environment or within its superordinate system. In order to gain an overview of the whole system, all the elements arising are then determined.

Elements in this case include both the components and the component interfaces. Component interfaces are for example shrink connections, welded connections, etc., which also represent reliability-critical components of a system as well as the components themselves. To illustrate the functions of the system and components, it is helpful to further subdivide complex products into functional groups or sub-assemblies.

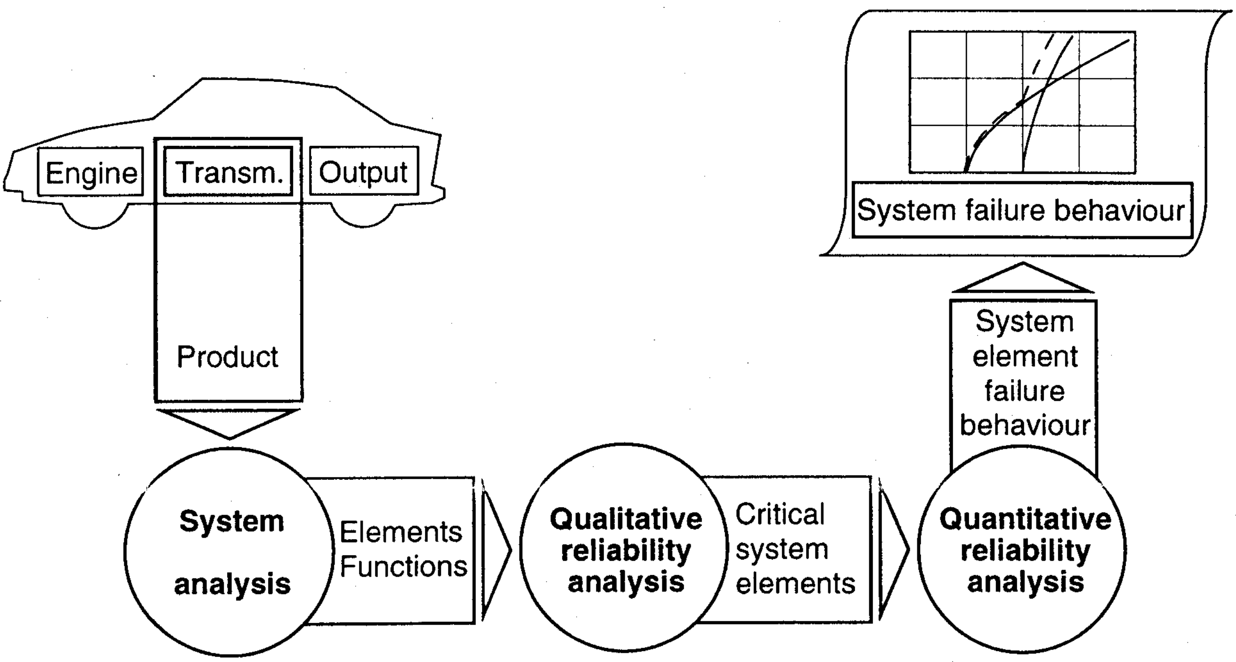


Figure 16.11. Procedure for reliability analysis

System Delineation

To describe the effect of failures of a product on neighbouring systems, it is necessary to establish a system boundary. This then involves determining all interactions across system boundaries. The system boundary relates both to mechanical and hydraulic as well as electrical connections. For an vehicle transmission for example this is the input and output, the suspension, the gearshift and the data circuit for engine speed measurement (Figure 16.12).

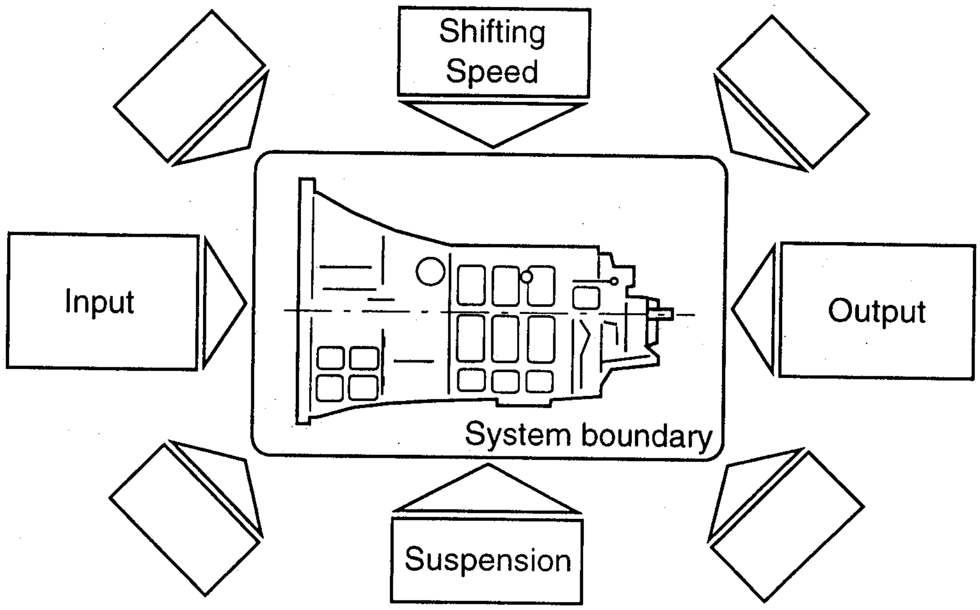


Figure 16.12. System boundary for a passenger car selector gearbox

Interactions between the Components

The links and interactions of the individual components are shown in the so-called "functional block diagram". It is important that this diagram is clear and complete, since the functional block diagram is the starting point and basis for the qualitative reliability analysis described in the following section.

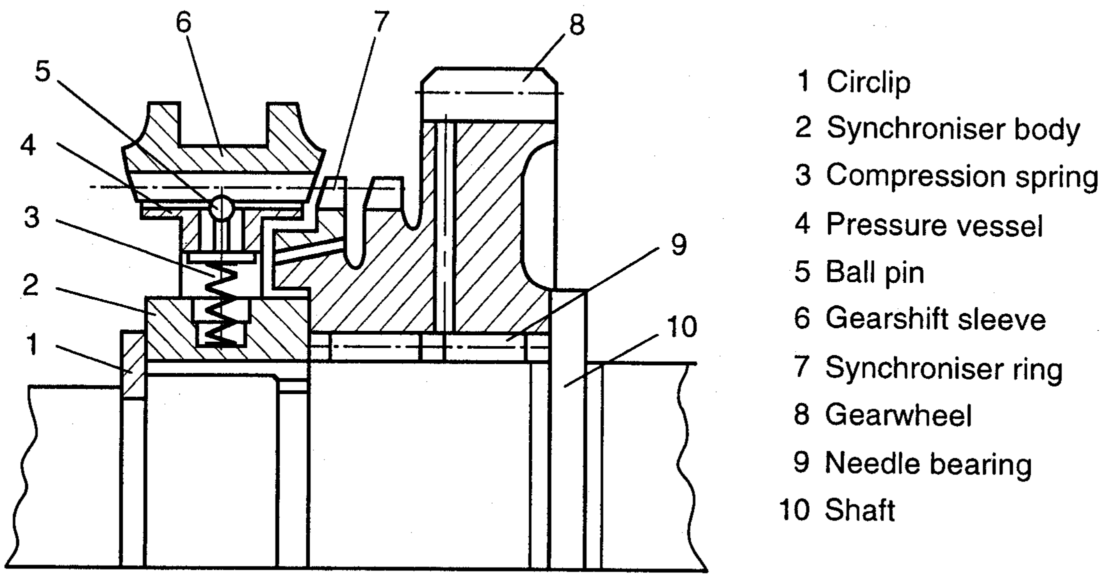


Figure 16.13. Assembly drawing of a single-taper synchroniser

The functional block diagram is derived directly from the design drawing (Figure 16.13). The arrangement of the components in the functional block diagram should correspond as far as possible to the structure of the design, so that the output and power flows can be directly recognised (Figure 16.14).

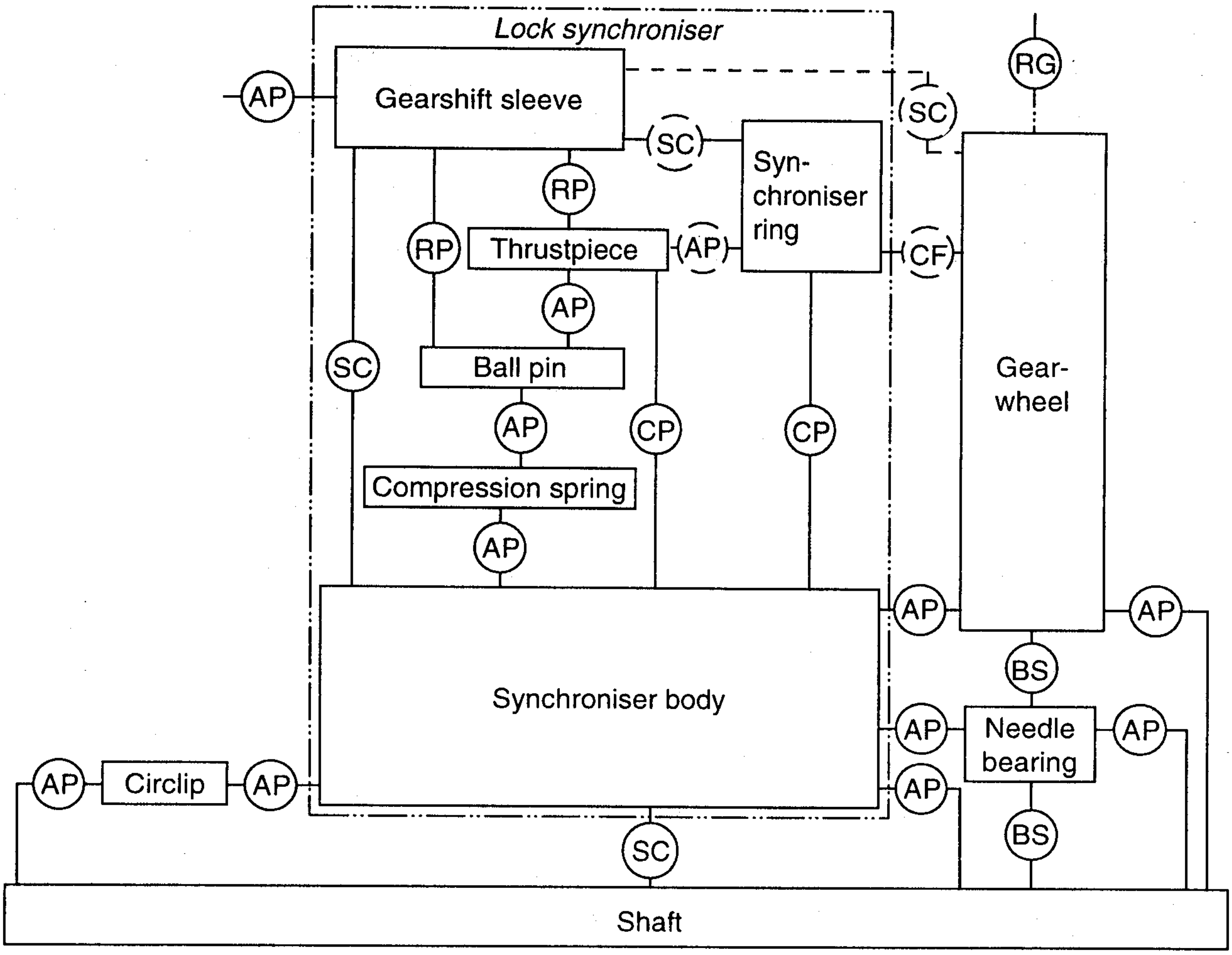


Figure 16.14. Functional block diagram of a single-taper synchroniser, see Figure 16.15

Component	Description			
Component interface	XY			
Universal elements	**Direction of action**			**Type of connection**
	axial	A	P	positive locking
	radial	R	F	frictional
	circumference	C	M	material locking
Special elements	Bearing surface			BS
	Spline shaft connection			SC
	Running gears			RG
	Temporary connection			( )

Figure 16.15. Elements of the functional block diagram for the single-taper synchroniser

The component interfaces should be marked in the functional block diagram by indicating the type of link. The links which cannot be described through a linking element, such as the meshing of running gears or rolling contact of rolling bearings, must be introduced as special elements, and marked accordingly or listed separately.

16.2.2 Qualitative Reliability Analysis

Qualitative reliability analysis involves investigating the subsystems relevant to reliability in terms of their failure potential, and assessing them in terms of their effect on the functionality of the system as a whole. The assessment should be based on the functions of the product determined in the course of systems analysis, the interrelations between components, calculations, results of tests, fault statistics and knowledge gathered from experience. The results of qualitative reliability analysis can be represented as a fault tree (FTA), block diagram (BD) or in the failure mode and effect analysis (FMEA) worksheets.

To document that some elements can fail in different ways, the elements are broken down into system elements according to the type of fault. In the example described of the single-taper synchroniser, the component, e.g. gearwheel, must be broken down into the system elements fracture, pitting or corrosion. The system elements resulting from this breakdown naturally fulfil different functions, and thus make a different contribution to system reliability. This means it is neither relevant nor reliable to regard all system elements as of equal value. It is therefore necessary to pre-select elements that are relevant to reliability and those that are not before assessing the elements. This can for example be carried out using the so-called "A, B, C" analysis.

"A, B, C" Analysis

A helpful technique for pre-selecting elements that are relevant to reliability is subdividing them into three categories, as shown in Figure 16.16. The system elements are classified on the basis of the effect of the system elements on system reliability and predictability of their B_x service life. Whilst the B_x service life of "A" system elements that are critical to reliability can be predicted, in the case of "B" system elements that are also critical to reliability, one has to rely on data derived from experience or from test results. The "C" system elements that are neutral in terms of reliability are not taken into account in subsequent analysis. After this classification, the necessary calculations or tests can be carried out for the rest of the reliability analysis. The "A, B, C" analysis can be regarded as a highly simplified form of the failure mode and effect analysis (FMEA) described in the next section.

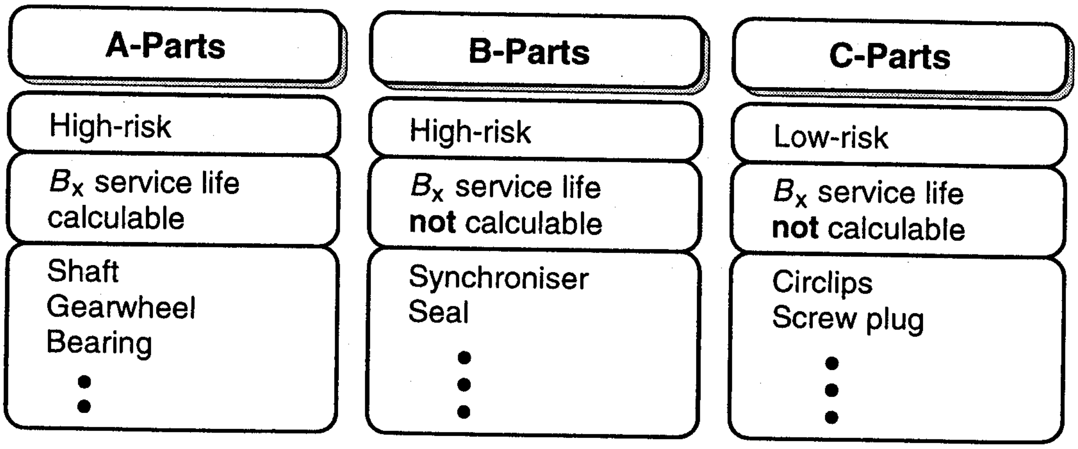


Figure 16.16. "A, B, C" analysis

FMEA, FTA, BD

In the case of failure mode and effect analysis (FMEA) potential failures are systematically identified and assessed. It should always be carried out for new and important products in parallel with the design process [16.11], in order to immediatelly take account of the necessary design improvements. The FMEA is based on the functional block diagram described in the previous section, which illustrates the interaction of the components. Starting from the elements, the FMEA is broken down into three steps *risk analysis, risk assessment and concept optimisation* (Figure 16.17).

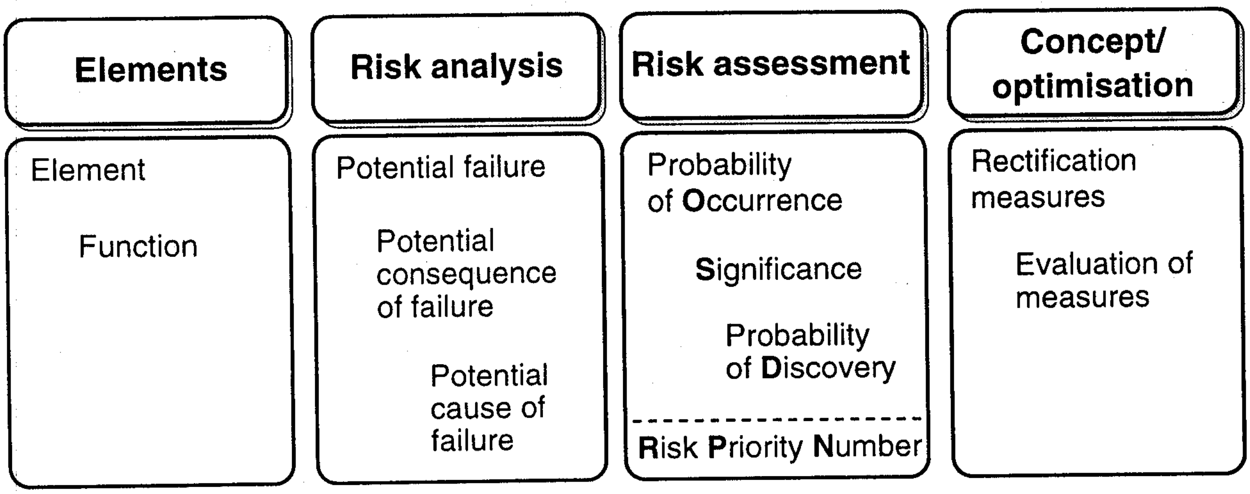


Figure 16.17. Structure of a failure mode and effect analysis (FMEA)

Risk analysis involves recording all potential failures, their consequences and causes. Risk assessment gives the probability of occurrence O, the significance of the consequences of failure S, and the probability of discovery D of the cause of the failure, taking into account the prevention and testing measures in place. The assessment process uses the risk priority number *RPN*.

It is defined as the product of the variables O, S and D. The principle here is that a low risk priority number *RPN* is better in the interests of a reliable product than a high one. For the components and failure causes deemed to be critical, rectification measures are recommended, the persons responsible determined, and the necessary improvement measures taken.

Fault tree analysis (FTA) [16.12] is an analytical procedure, the result of which is displayed as a fault tree. This involves a deductive procedure in which all associated system elements are sought, starting from the system failure behaviour. Fault tree analysis and failure mode and effect analysis (FMEA) lead to substantially the same result. Figure 16.18 shows the fault tree of an analysis relating to the chronological sequence of an FMEA, to illustrate this relationship. This shows that a part of the fault tree analysis is implicitly contained in a component or element FMEA.

In the block diagram (BD) the system elements are linked together to form a reliability structure, which can be either a serial or parallel structure, or a combined structure (Figure 16.10).

Constructing the block diagram involves an inductive procedure, since conclusions are drawn about system failure behaviour from the failures of individual system elements. As in fault tree analysis, the block diagram is in some way contained in the FMEA. In a serial system the block diagram is created by setting out the individual system elements (potential failures) arising from the FMEA risk analysis, one behind the other (Figure 16.18).

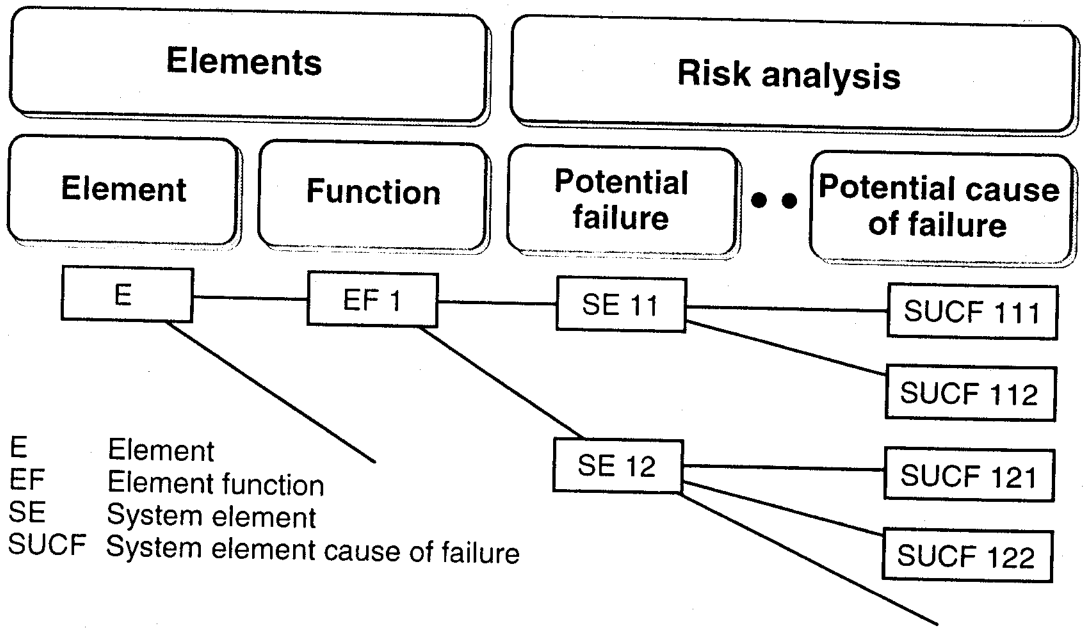


Figure 16.18. Relation between BD, FTA and FMEA

The structure encountered in most engineering products in practice is a serial structure. This means that there is no redundancy, and that the first failure of a system element leads to a failure of the whole system.

16.2.3 Quantitative Reliability Analysis

The aim of quantitative reliability analysis is to determine the failure behaviour of the system elements identified as critical in the qualitative reliability analysis. On this basis the failure behaviour of the system is determined in accordance with the principles of reliability theory discussed in the preceding sections.

Failure Behaviour of System Elements

The failure behaviour of system elements is determined in different ways depending on how the system element concerned is categorised in the "A, B, C" analysis. There are relatively precise duty cycles for the "A" system elements and the associated Wöhler curves. This enables the service life of the system elements to be determined by means of a serviceability calculation. This calculated service life corresponds in most cases to the B_1 or B_{10} service life. For the "B" system elements one has to rely on data derived from experience and test results. If the failure behaviour profile is known, the B_1 or B_{10} service life can be converted into the characteristic service life T using appropriate equations, as in [16.1]. To be able to describe the failure behaviour of a system element fully, the associated distribution function is placed using the point in the Weibull network determined with the B_x service life.

There are different ways of determining the Weibull parameters t_0, T and b needed to describe failure behaviour. The most reliable is to carry out tests. This is however very costly, since a large number of tests are required to produce representative results. If Weibull parameters are available for the same type of failure under comparable conditions, the Weibull parameters required can be estimated using knowledge derived from experience and calculation. There are now also reliability databases containing the Weibull parameters as a function of stress, machining, the material used and the failure mechanism for some machine elements. Using the values stored in the database, and possibly other tests, it is possible to estimate the parameter b (and in certain cases t_0) required.

Failure behaviour depends principally on the failure mechanism (e.g. fracture), on the load (time period or fatigue strength range), the type of machining, and the material. Instead of the time without failures t_0, the relationship t_0 / B_{10} has proved its value in practice. The major influence on the failure behaviour of a component is the failure mechanism. For example components which fail as a result of fracture have higher b or t_0 / B_{10} values than components which fail as a result of pitting (Figure 16.19). Components with a higher level of machining or material quality also have higher b or t_0 / B_{10} values than components of lesser quality. Higher stressing results in higher b and smaller t_0 / B_{10} values. More brittle materials usually have a narrower distribution than resilient ones [16.3, 16.10].

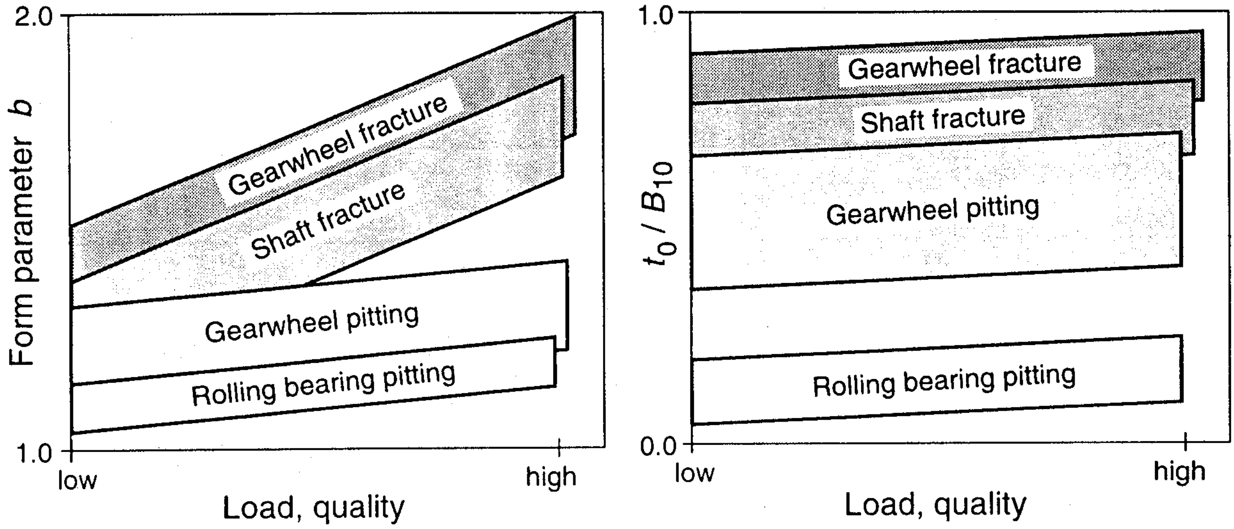


Figure 16.19. Form parameters b and t_0 / B_{10} derived from interpretation of test results

With the Weibull parameters thus determined, the failure behaviour of the particular element can be determined in accordance with Figure 16.20.

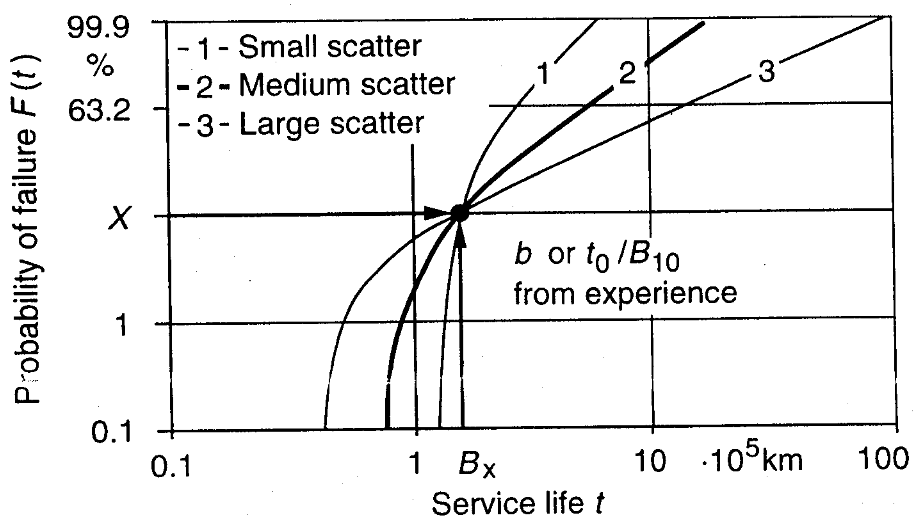


Figure 16.20. Determining failure behaviour from empirical knowledge

Failure behaviour can be displayed on Weibull paper, and also described in tabular form with the B_{10} service life, the form parameter b and the value t_0 / B_{10}. Table 16.2 shows this for the example of the shaft, idler and single-taper synchroniser described above, for comparative purposes.

Table 16.2. Weibull parameters using the example of a lock synchroniser unit

System element	B_{10} (10^5 km)	t_0 / B_{10}	t_0 (10^5 km)	b	T (10^5 km)
Gearwheel Fracture	8.2	0.85	7.0	1.7	11.5
Gearwheel Pitting	1.9	0.5	1.0	1.15	7.4
Shaft Fracture	10.8	0.8	8.6	1.3	21.0
Needle bearing Pitting	14.7	0.15	2.2	1.2	83.7
Synchroniser ring Wear	3.7	0.5	1.9	1.1	15.8

Failure Behaviour of the System

As discussed in Section 16.1.4, the survival probabilities $R_i(t)$ of the critical system elements are linked in accordance with an appropriate system theory to determine the failure behaviour of the system. In the case of vehicle transmissions, the Boolean theory has proved a good approximation.

If the component failure behaviour can be described with a two-parameter Weibull distribution, then the individual components already have a limited reliability $R_i(t) < 1$, even starting from $t > 0$.

This in turn implies that for calculating system reliability all components have to be taken into account, and that each additional component further reduces system reliability. Where there are many components, this leads to very low system reliability.

The failure behaviour of many components can however often, be described better with a three-parameter Weibull distribution. In these cases the components only have to be taken into account for determining system reliability when the running time $t > t_0$. Thus if the early failures (section 1 of the bathtub curve) are prevented by suitable quality assurance measures, then the reliability of the system is determined only by the design of these components in terms of fatigue and wear [16.13].

If the failure probabilities of the various system elements are recorded on Weibull paper, it is generally easy to see which elements largely determine system failure behaviour. Figure 16.21 shows the failure probability for the example already mentioned of the single-taper synchroniser. In this example it is readily apparent that a minimal improvement in tooth flank load capacity (failure of the gearwheel attributable to pitting), can nearly double the B_{10} service life of the system.

Finally it should be noted that the analysis is particularly suited to comparing similar products to be operated under exactly the same operating conditions. Reliability analysis is however also appropriate for new products if the emphasis is not on absolute figures, but the intention is to investigate the effect of the critical system elements on system failure behaviour by means of parameter variations.

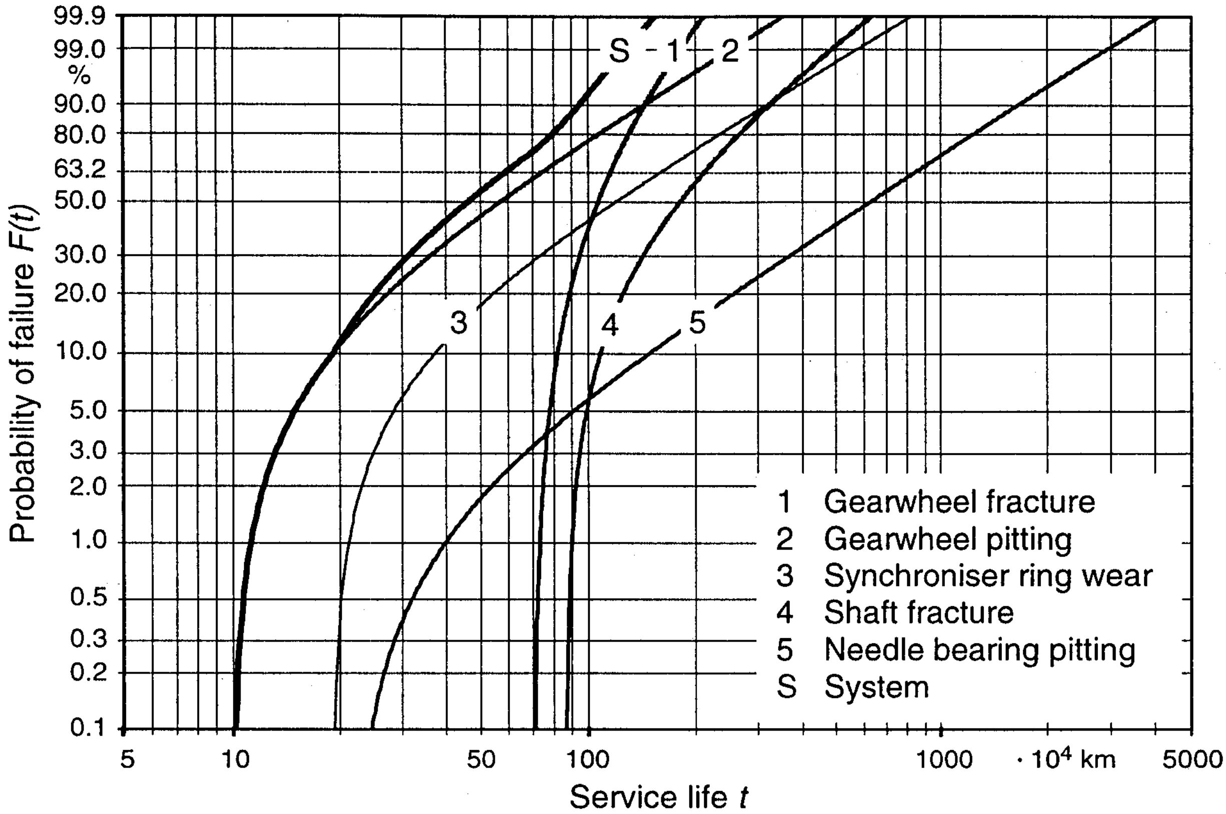


Figure 16.21. System failure behaviour, based on the example of a single-taper synchroniser

16.3 Testing to Ensure Reliability

Each calculation proceeds from idealised conditions (see also Figure 15.4a). Where many parts interact in a complex system under environmental conditions that are not precisely known, the process is only partially susceptible to numerical simulation. It is therefore essential to carry out practical trials on components and complete gear units. In particular weak points and risk components can be identified by testing before production begins. In many cases comprehensive tests are also prescribed by the customer or legislator.

It is essential that the systematic development process and the measures to ensure a product's reliability are co-ordinated before development starts (Figure 16.22). Component and material tests are necessary even at the planning and concept stage. It is also necessary to determine practical load profiles for rating at this point in time. After design detailing and prototype production, comprehensive testing is carried out on test-beds. The final stage is testing in the vehicle under realistic conditions – the most realistic and decisive test. If a defect is discovered in the course of a test or trial, this can sometimes lead to an extremely costly and time-consuming re-run of the development process.

In order to be able to take account in the development process of the requirements for designing a reliable product, it is necessary to consider reliability in addition to serviceability design of the power transmitting and other risk components. The ideal product development process in this regard is one where the individual phases are arranged in a closed loop. Integrating a constant reliability monitoring function at the testing stage and during practical operation enables this ideal case to be nearly realised (Figure 16.23).

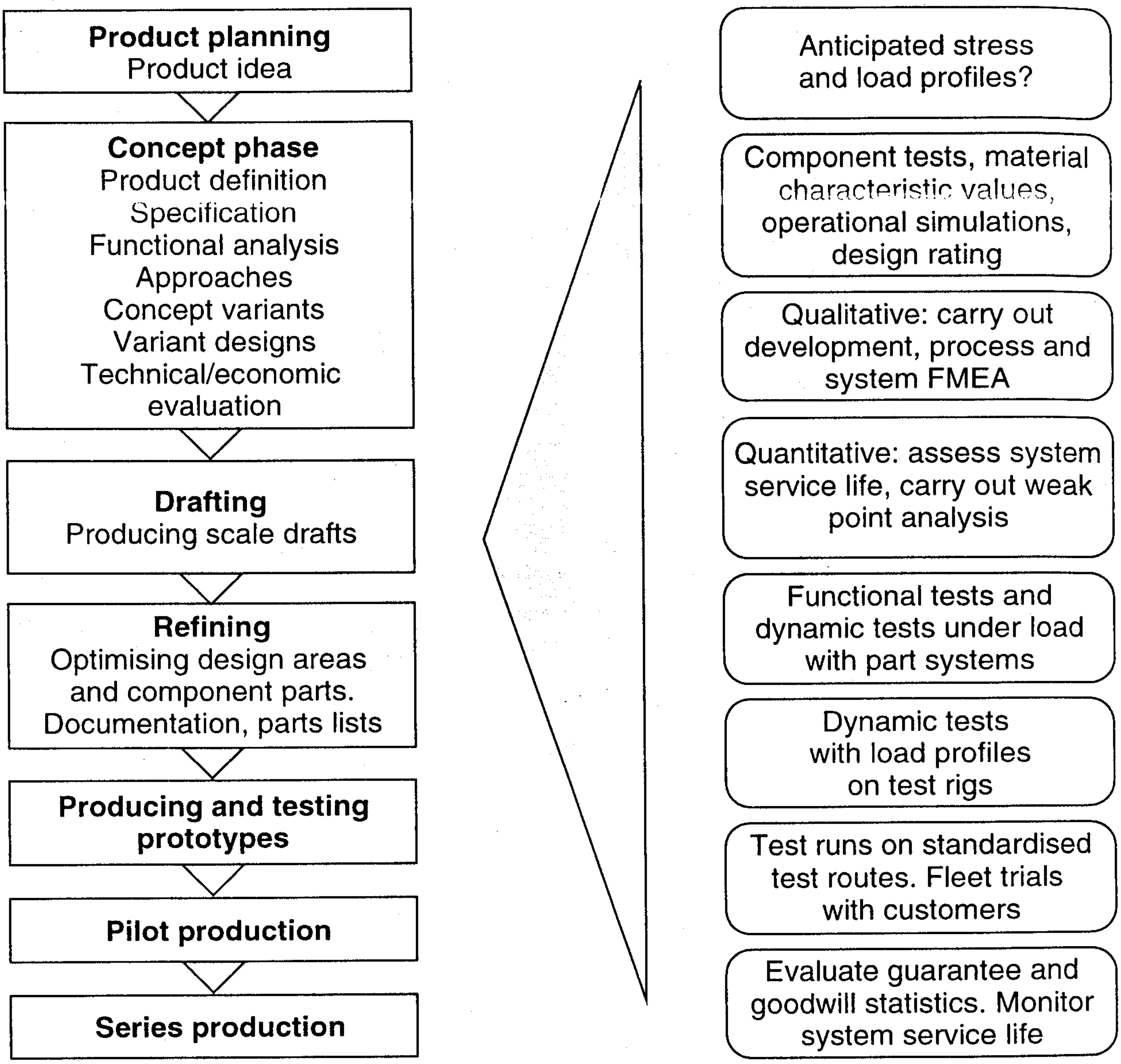


Figure 16.22. Interlinking of the development process and ensuring reliability

16.3.1 Classifying Vehicle Transmission Test Programs

The test programs carried out in the last phase of vehicle transmission development can be classified into three main areas:

1/ component testing (component and analogue tests),
2/ prototype bench tests,
3/ vehicle testing.

1/ Component Testing

Component testing is carried out with individual components or with "analogue test parts", which is the simplest kind of test. However these analogue test parts only give an approximation of the failure behaviour of the components for a particular type of fault, e.g. notch impact test for gearwheel fracture. Precise indications of service life or other types of failure will be derived from the actual component under test. A distinction is made in component testing between static and dynamic tests (Figure 16.24).

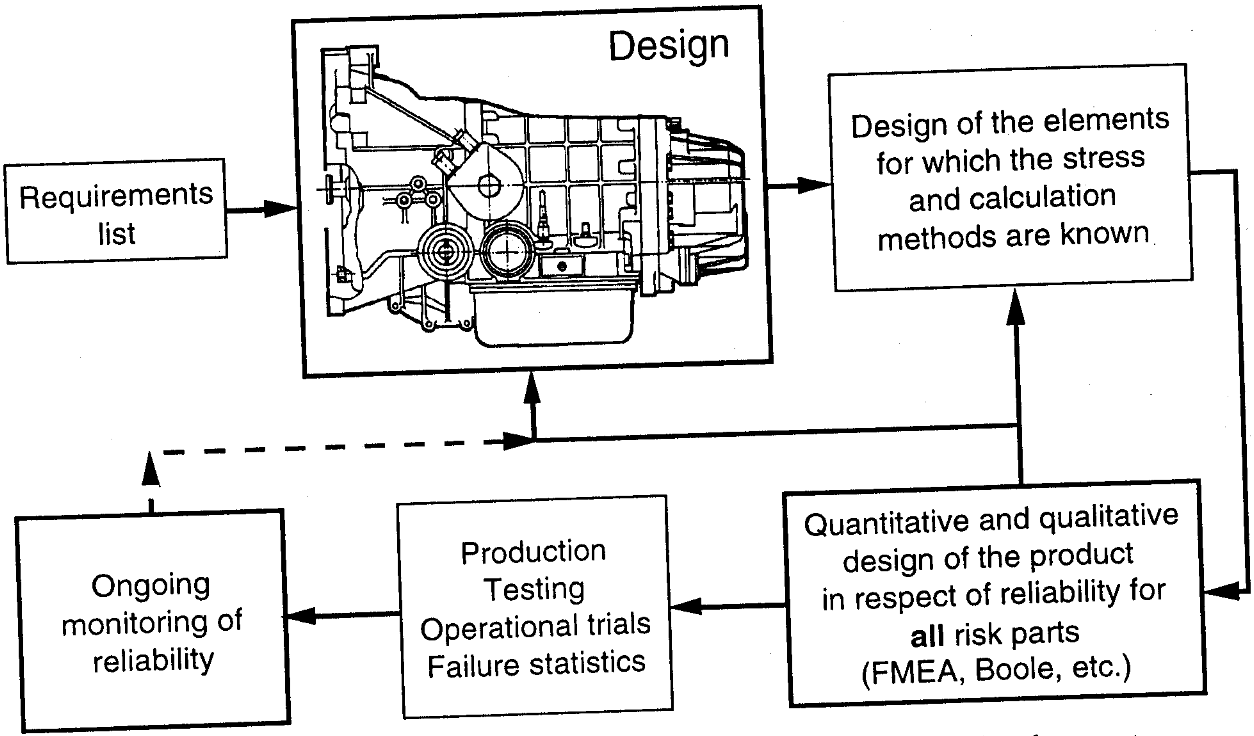


Figure 16.23. Designing for reliability as part of a systematic product development process

Whilst component testing provides principally information for dimensioning components, the prototype tests described in the next section, provide initial indications of the whole system behaviour.

2/ Prototype Bench Tests

For this testing variant a distinction is made between pure functional testing and continuous testing (Figure 16.25).

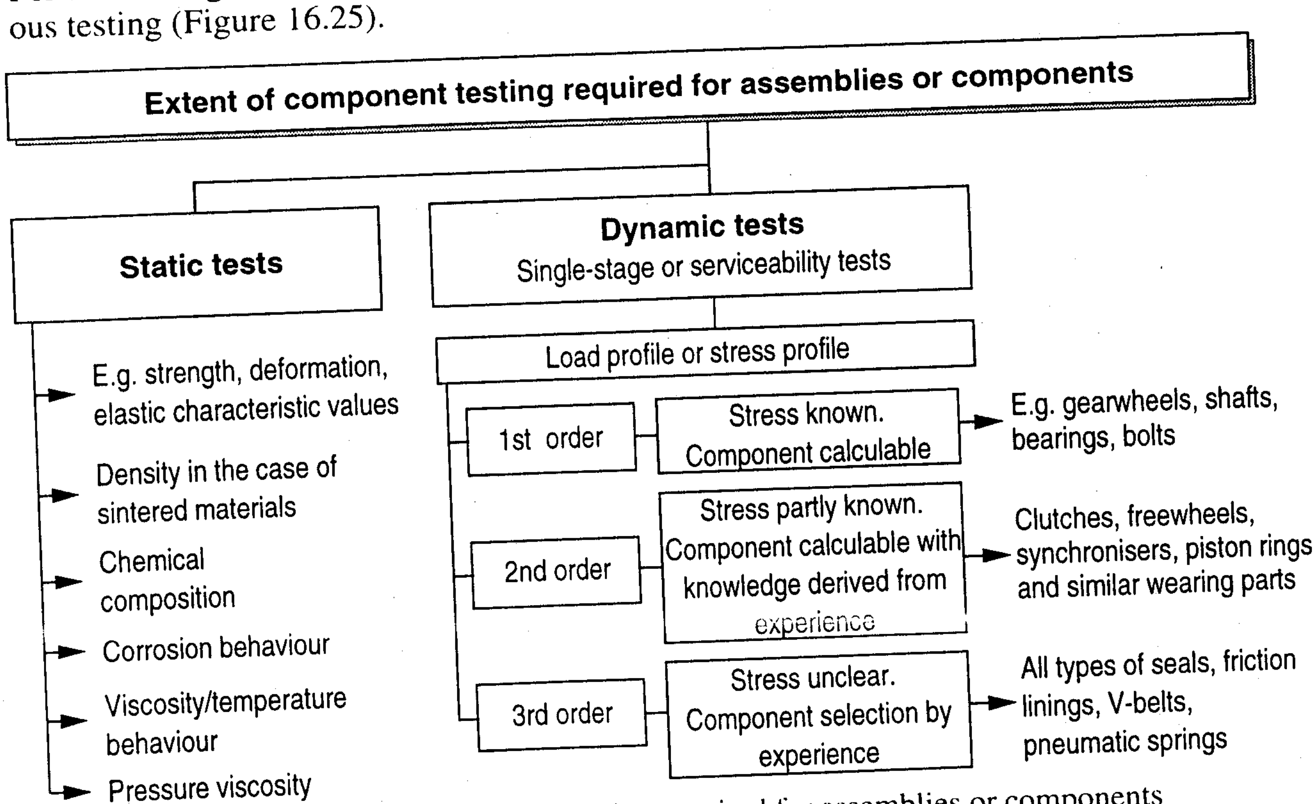


Figure 16.24. Extent of component testing required for assemblies or components

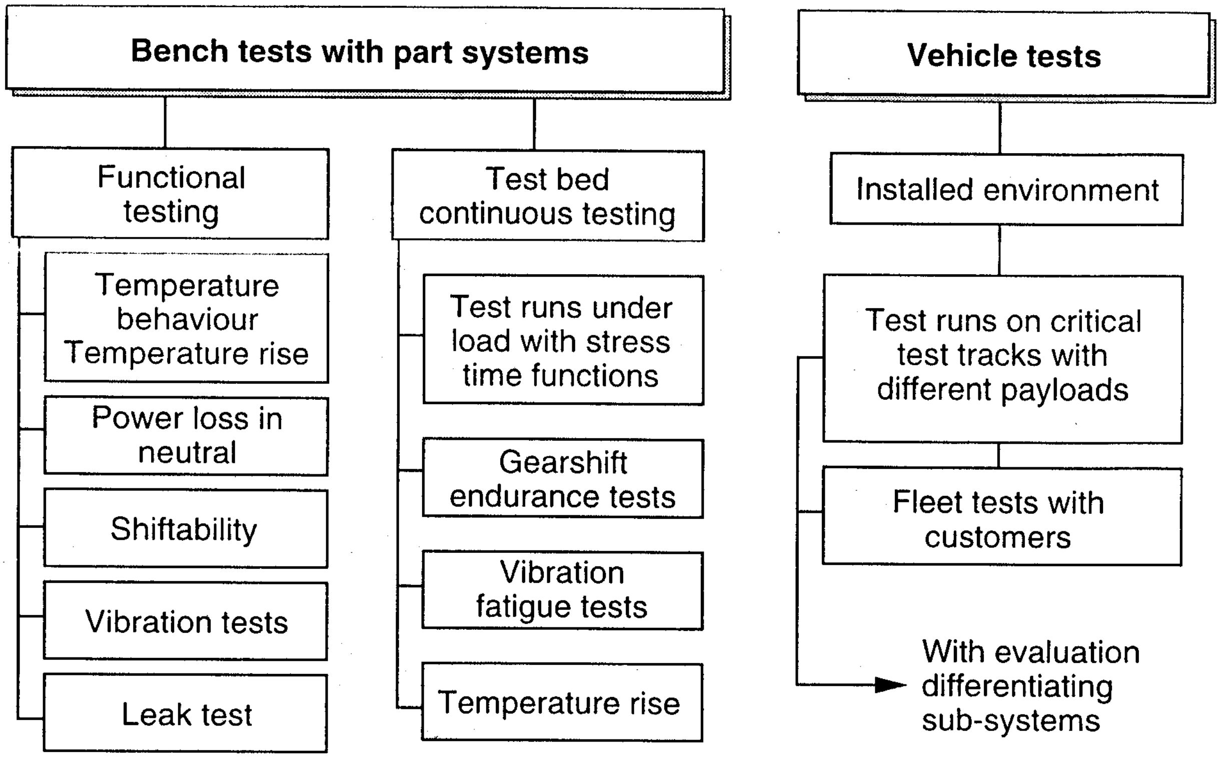


Figure 16.25. Prototype bench trials with part systems and vehicle testing

With functional testing, the power conducting parts are initially tested with low stress. This type of test is commonly used for transmissions, for example to determine their shifting characteristics or oil supply at different lateral and longitudinal inclinations.
With test bed continuous testing, testing is carried out as a function of the possibilities provided by the test bed, e.g. particular load profiles as load runs with precisely defined stress/time functions, or continuous shifting performance.

3/ Vehicle Testing

Vehicle testing involves testing the system as the last part of the testing phase. In this phase the reliability of the transmission is determined on various test routes and under different conditions, in addition to testing the in situ relations (Figure 16.25).

The test routes are characterised by the height profiles, the gradient distribution and the speed distribution, and have a "time lapse effect" as regards transmission wear. The routes are so demanding as to produce faults after short running times. In the case of vehicles, the test routes are often so designed as to include a mixture of motorway, hilly stretches and mountain roads (see also Table 2.9 "Typical vehicle use").

The test routes are specified at the planning phase of development, since their data, in combination with the vehicle data to be derived, give the load cycles which will be needed for rating right from the development phase. The stress/time profiles recorded during the test runs are transferred to the load profiles by an enumeration process (see also Section 7.4.2 "Load Profile and Enumeration"). Figure 16.26 shows causes of the stress/time function.

In the context of testing for motor vehicles, the trend is now increasingly towards "fleet trials" in co-operation with taxi companies or hauliers. These fleet trials provide an extraordinarily realistic profile in respect of various types of stress. This fact then implies that wear arising in the course of these fleet trials will also occur in actual day-to-day use.

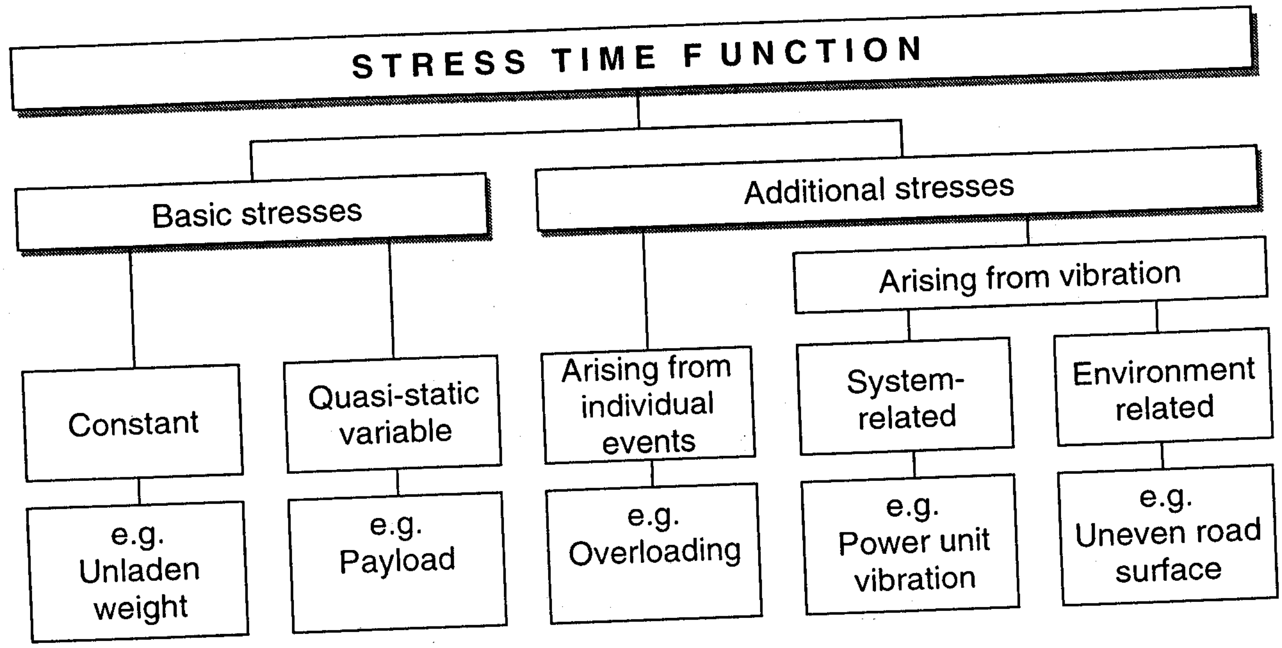


Figure 16.26. Causes of the stress-time function. Source: BUXBAUM

It should be noted that the mathematical principles of reliability calculation have a bearing on all the testing programs referred to.

It is necessary to clarify beforehand how many test units permit a significant result for a given population. It should be mentioned at this point that tests have been carried out to show that when the number of test pieces is increased, their actual test duration of the test pieces can be reduced [16.14]. This is naturally highly desirable with a view to saving time and money when developing new products.

16.3.2 Test Stands for the Test Programs

For component and prototype tests, test beds are needed which can be divided into different categories in line with the test programs:

- component test beds,
- functional test beds,
- test beds with energy flow,
- torque test stands.

The component test beds serve to test individual components or analogue test parts that behave in a similar way to the components under investigation in terms of stress that can be tolerated and in terms of type of failure. The test beds themselves are usually of very simple design, and can therefore only be used for testing parts of similar design (similar geometry).

Such component test beds are for example:

- roller dynamometers for investigating pitting,
- rotating bending machines and torsion testing machines for testing shafts,
- pulsators for recording the Wöhler curves of individual teeth of a gearwheel,
- four-ball tester for lub oil to determine the scuff-limited load of lubricants.

Individual functions of the transmission are tested on functional test beds. In the case of vehicle transmissions these are for example functionality and gear-shift behaviour. The drive motor of the test stand only has to provide the power loss of the components, and can therefore cost-effectively be made compact. Figure 16.27a shows a schematic functional test stand for testing gear shifting and heat generation, both when idling and under power. Test stands using the energy flow method are suitable (Figure 16.27b) for testing under realistic stress. In this case the transmission is driven by an internal combustion engine, as in practical use. The driving resistance is provided by a friction brake. Normal use can be realistically simulated by controlling the engine and brake. Often an electric motor is used instead of a combustion engine to drive the rig. Friction dynamometers are often used to provide braking.

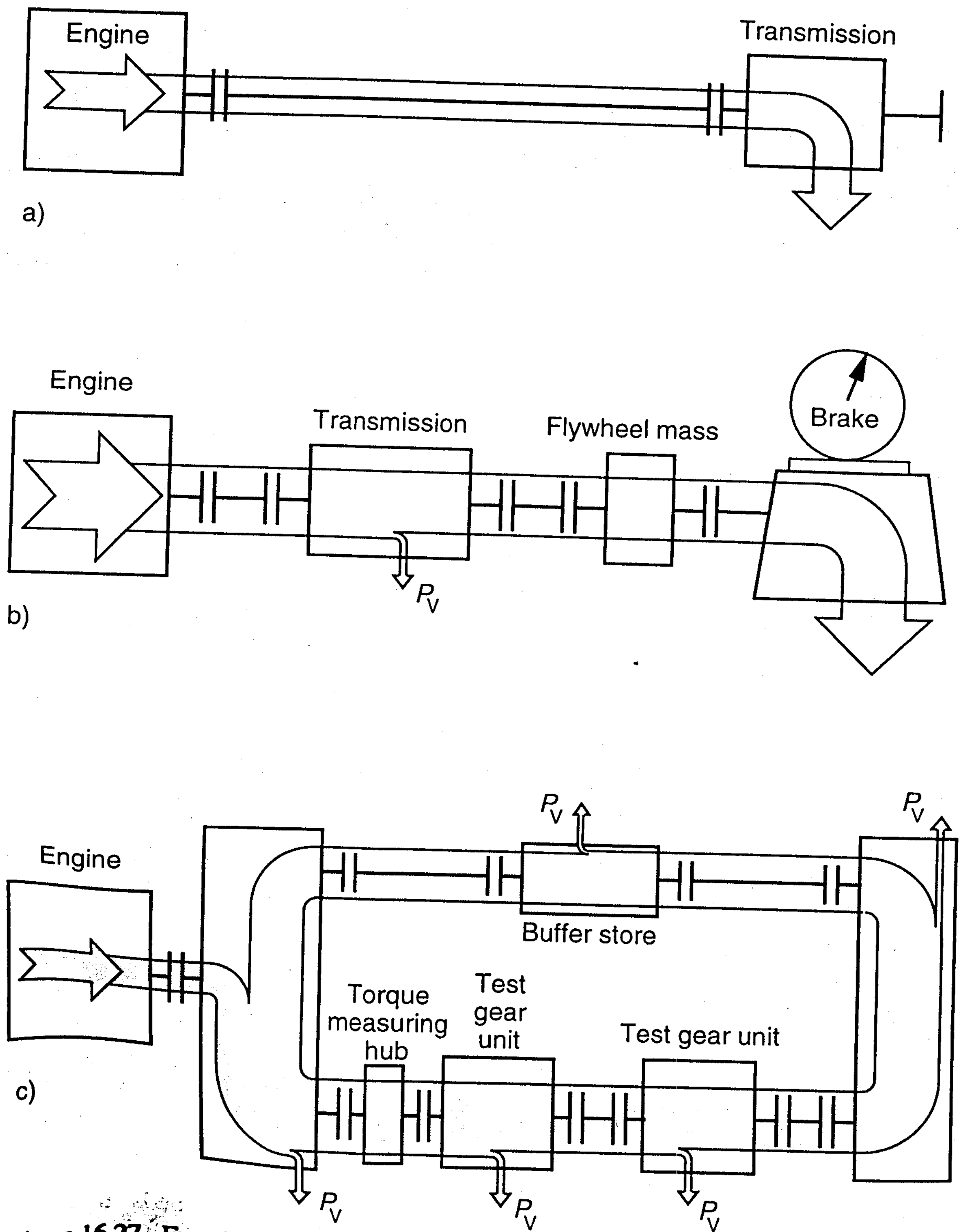


Figure 16.27. **Functional** test stands, basic structure and power flow. **a)** Shifting and **warming up in neutral;** **b)** Energy flow process; **c)** Energy loop process

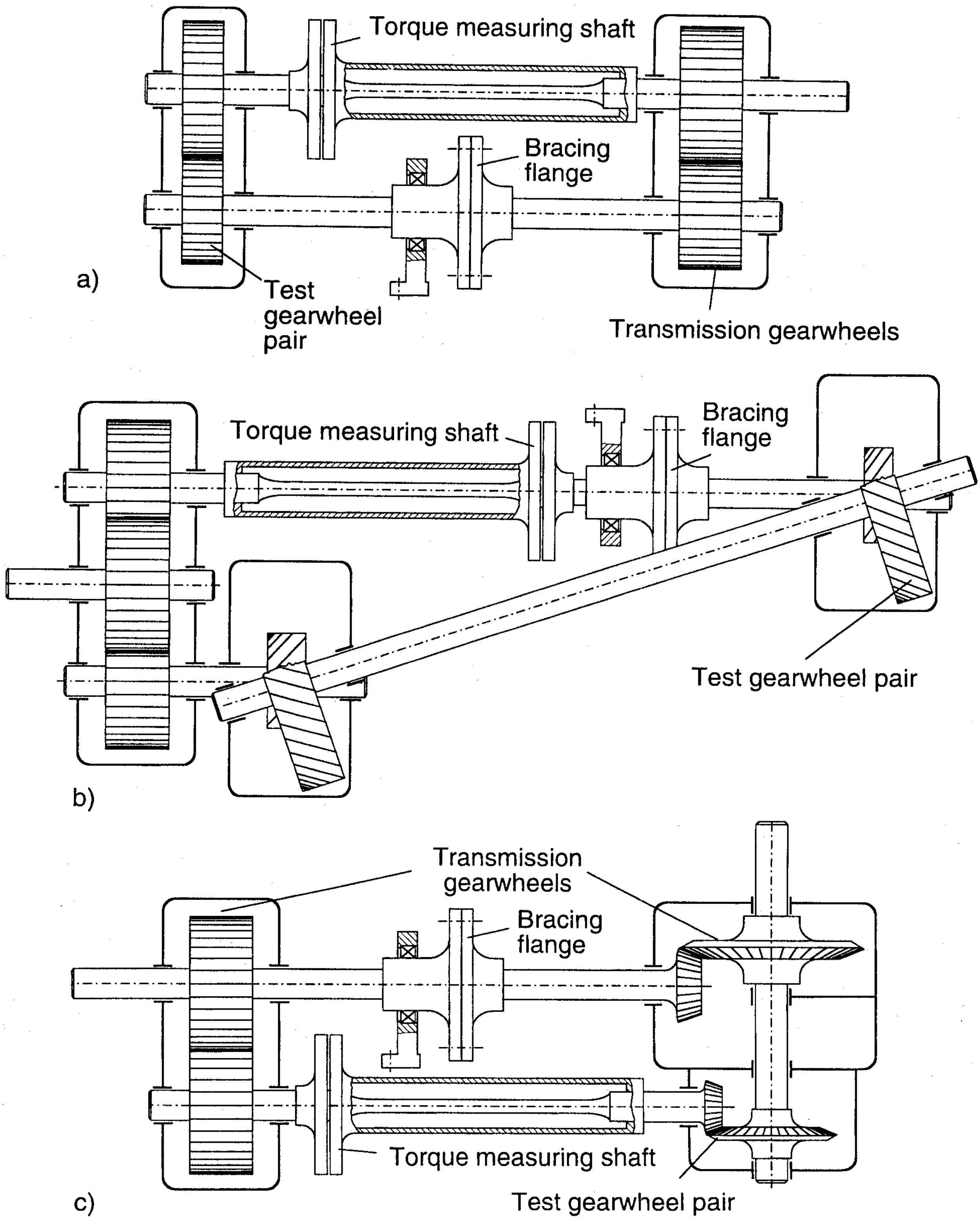


Figure 16.28. Torque test rigs for: **a)** Spur gears; **b)** Helical gears; **c)** Bevel gears

The major disadvantage of the energy flow method is that the drive motor and the braking device have to be designed for full test performance, which involves large plant and consequently high cost of testing.

The disadvantages of the energy flow method can be circumvented by using torque test rigs with an energy circuit (Figure 16.27c). With a torque test rig, the output of the test gear unit is linked to the input of the crossover linkage and shafts. The necessary test power is also split up into generating torque and rotational speed. For producing torque, the circuit is split at a particular point, the two parts turned against each other with the required torque, and then fixed in their turned position. Since most transmissions only have positive locking parts, this is relatively easy to achieve. The transmission parts are statically stressed with the torsional load of torque T. The rotational speed needed for dynamic operation is produced by an electric motor. The electric motor only has to provide the sum of the power loss (10–20% of the test power) and is thus substantially smaller than in the energy flow method.

The torque test unit on torque test stands can be of hydraulic or mechanical design. Figures 16.28a–c show the schematic diagrams of three mechanical torque test rigs designed to test different types of gearwheel. The distortion is produced by combining the hollow shaft as torque absorber and the pre-load flange for bracing it. Figure 16.28a shows the test stand for testing spur gears, Figure 16.28b shows a test stand for testing worm gears, and Figure 16.28c shows a test stand for testing bevel gears. Complete transmissions can also be tested with torque test stands, depending on design (Figure 16.29). In this case a countershaft transmission with two countershafts is used as the torque test device. The housing is turned with a helical gear to produce the torsional stress.

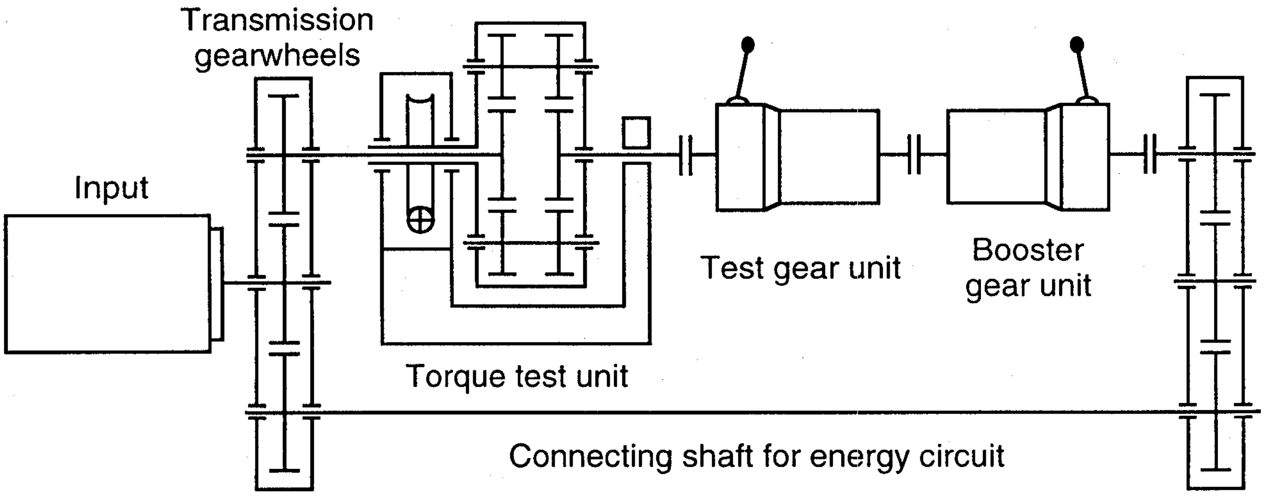


Figure 16.29. Torque test rig for vehicle transmissions

In addition to mechanical torque test stands listed, electrical ones are also frequently used. They generally consist of two electric units, of which one serves as a motor and the other as a generator. The electrical power of the generator is fed back into the power circuit so that only the power loss is consumed.

References

The references are arranged by chapter. Those references marked with an " * " are not used directly in the text, but amplify and complement the topic under discussion.

Chapter 1 "Introduction"

1.1 Lohr, F.W.: *Marktorientierte Kraftfahrzeugentwicklung* (Market-Oriented Motor Vehicle Development) Aachen: 2. Aachener Kolloquium "Fahrzeug- und Motorentechnik", 1989, pp. 1–18

1.2* Kühner, K.: *Geschichtliches zum Fahrzeugantrieb* (Historical Notes on Vehicle Drive Systems) Friedrichshafen: Eigendruck der Zahnradfabrik Friedrichshafen AG, 1965

1.3* Maier, A.: *Vom Kammrad zur Zahnradfabrik. Eine Dokumentation zur Zahnradgeschichte* (From Cogwheel to Gearwheel Factory. A Documentary Record of Gearwheel History) Vol. I–VI. Friedrichshafen: Eigendruck der Zahnradfabrik Friedrichshafen AG, 1975

1.4* Maier, A.: *Geschichtliche Entwicklung von Fahrzeuggetrieben* (Historical Development of Motor Vehicle Transmissions) Friedrichshafen: Sonderdruck der Zahnradfabrik Friedrichshafen AG

1.5* Seherr-Thoss, H.-Chr; Graf v.: *Die Entwicklung der Zahnradtechnik. Zahnformen und Tragfähigkeitsberechnung* (The Development of Gear Engineering. Tooth Formation and Capacity Calculation) Berlin; Heidelberg; New York: Springer-Verlag, 1965

1.6* Sprague de Camp, L.: *Ingenieure der Antike* (Engineers of Ancient Times) Düsseldorf; Wien: ECON-Verlag, 1964

1.7* Priwitzer, G.: *Geschichte der Fahrzeuggetriebe* (History of Automotive Transmissions) Stuttgart. Universität, Institut für Maschinenelemente, Studienarbeit, unveröffentlicht, 1982

1.8* Eckermann, E.: *Vom Dampfwagen zum Auto* (From Steam Passenger Car to Automobile) Rowohlt-Verlag, 1981

1.9* Klinckowstroem, C.; Graf v.: *Knaurs Geschichte der Technik* (Knaurs History of Technology) München; Zürich: Droemersche Verlagsanstalt TH, Knaur Nachfolger, 1959

1.10* Ungerer, H.: *Die Geschichte der Fahrzeugtechnik in sechs Jahrtausenden* (The History of Vehicle Technology over Six Millennia) München: Vortrag auf der 12. Fachtagung (100 Jahre Automobiltechnik) des Münchner Arbeitskreises für Straßenfahrzeuge MAS e.V., 28.02.1986

1.11* Parmee, K.: Gearing up for the Future – A Story of Commercial Vehicle Transmission Development. Proc. Instn. Mech. Engnrs. **202** (D2) (1988), pp. 71–81

1.12* Div. Autoren: *100 Jahre Automobil. Daimler-Benz 1886-1986* (100 Years of the Automobile. Daimler-Benz 1886-1986) München: Sonderbeilage der Süddeutschen Zeitung (24.01.1986), No. 19

1.13* Szabo, I.: *Geschichte der mechanischen Prinzipien* (History of Mechanical Principles) Basel; Stuttgart: Birkhäuser Verlag, 1977

1.14* Schönhaar, P.: *Beitrag zur Entwicklungsgeschichte der Maschinenelemente, ihrer Berechnung und Konstruktion* (Paper on the History of the Development of Machine Elements, their Calculation and Engineering Design) Stuttgart, Universität, Institut für Maschinenelemente, Studienarbeit, unveröffentlicht, 1984

1.15* Lauster, E.: *Untersuchungen und Berechnungen zum Wärmehaushalt mechanischer Getriebe* (Investigations and Calculations Relating to Heat Management within Mechanical Transmissions) Stuttgart, Universität, Institut für Maschinenelemente, Dissertation, 1980

1.16* Lang, O.R.: *Geschichte des Gleitlagers* (History of the Plain Bearing) Stuttgart: Sonderdruck der Daimler-Benz AG, 1982

1.17* Div. Autoren: *Technik. Kleine Enzyklopädie* (Technology. Small Encyclopaedia) Leipzig: Verlag Enzyklopädie Leipzig, 1956

1.18* Div. Autoren: *Genius der Deutschen. Die großen Forscher, Erfinder, Ärzte* (Genius of the Germans. The Great Researchers, Inventors, Doctors) Auszug aus dem Sammelwerk "Die Großen Deutschen" von Heimpel, Heuss und Reifenberg. Berlin: Propyläen Verlag, 1966

1.19* Looman, J.: *Zahnradgetriebe* (Tooth Gearing Systems) 2nd Edition. Berlin; Heidelberg; New York; London; Paris; Tokyo: Springer-Verlag, 1988

Chapter 2 "Overview of the Traffic – Vehicle – Transmission System"

2.1 Kuhlmann, A.: *Auto und Verkehr bis 2000* (Vehicle and Traffic up to the Year 2000) Heidelberg; New York; Tokyo: Springer-Verlag; Köln: Verlag TÜV Rheinland, 1984

2.2 Förster, H.J.: *Transportsystem Straßenverkehr* (The Road Traffic Transport System) Automobil-Industrie 2 (1974), pp. 35ff

2.3 Helling, J.: *Längsdynamik von Kraftfahrzeugen. Umdruck zur Vorlesung Kraftfahrzeuge I* (Longitudinal Dynamics of Motor Vehicles. Reprint of the Motor Vehicles I Lecture) Aachen: RWTH, Institut für Kraftfahrwesen, ika Schriftenreihe Automobiltechnik, 1989

2.4 Helling, J.: *Umdruck zur Vorlesung Kraftfahrzeuge I* (Reprint from the Motor Vehicles I Lecture) Aachen, RWTH, Institut für Kraftfahrwesen, 1972

2.5 VDA: *Tatsachen und Zahlen aus der Kraftverkehrswirtschaft* (Facts and Figures from the Transport Industry) Frankfurt: Verband der Automobilindustrie e.V., 55., 57., 58., 62. Folge, 1991, 1993, 1994, 1998

2.6 Thyssen Henschel: *Transrapid*

2.7 Leutzbach, W.: *Zukunft des Straßenverkehrs* (The Future of Road Traffic) Aachen: 2. Aachener Kolloquium "Fahrzeug- und Motorentechnik", 1989, pp. 19–30

2.8 Lohr, F.W.: *Marktorientierte Kraftfahrzeugentwicklung* (Market-Oriented Motor Vehicle Development) Aachen: 2. Aachener Kolloquium "Fahrzeug- und Motorentechnik", 1989, pp. 1–18

2.9 *Entwicklungslinien in der Kraftfahrzeugtechnik* (Lines of Development in Automotive Engineering) Berlin: 2. Jahrestagung der VDI-Gesellschaft Fahrzeugtechnik, 1976

2.10 Report 1/83: *Perspektiven des zukünftigen Verkehrs. Einflüsse des Umfelds, mögliche Entwicklungen und realisierbare Gestaltungsspielräume* (Perspectives on Future Traffic. The Effects of Contextual Factors, Possible Developments and Achievable Scope for Future Development) Schriftenreihe der Daimler-Benz AG. Düsseldorf: VDI-Verlag, 1983

2.11 Daimler-Benz AG: *Verkehr, Umwelt, Zukunft* (Traffic, Environment, Future) Seminar der Forschungsgruppe Berlin, 1979

2.12 Lippl, E.: *Die Zukunft im Schnellverkehr. Eine reale Lösung* (The Future for Express Traffic. A Realistic Solution) VDI-Zeitschrift 115 (1973), pp. 1132–1135

2.13 Daimler-Benz AG: *Die O-Bahn* (The O railway) WZ20847/00/00/0479

2.14 *Magnetbahn-Schnellverbindung zwischen Köln und Frankfurt* (Maglev Rapid Transit Link between Cologne and Frankfurt) VDI-Nachrichten, 1985

2.15 Christ, H.: *Strategische Forschung und Entwicklung in einem Unternehmen der Kraftfahrzeug-Zulieferindustrie* (Strategic Research and Development in an Automotive Equipment Supply Company) Konstruktion 41 (1989), pp. 389–394

2.16 *Kundenbefragung der ZF Friedrichshafen AG* (Customer Survey by ZF Friedrichshafen AG)

2.17 Looman, J.: *Zahnradgetriebe* (Gear Mechanisms) 2nd Edition. Berlin; Heidelberg; New York; London; Paris; Tokyo: Springer-Verlag, 1988

2.18 Müller, H.W.: *Die Umlaufgetriebe* (Epicyclic Gears) Berlin; Heidelberg; New York: Springer-Verlag, 1971

2.19 Wolf, A.: *Die Grundgesetze der Umlaufgetriebe* (The Fundamental Principles of Epicyclic Gears) Braunschweig: Vieweg-Verlag, 1968

2.20 Küçükay, F.: *Rechnerunterstützte Getriebedimensionierung mit repräsentativen Lastkollektiven* (Computer-Aided Gear Rating with Representative Load Profiles) ATZ Automobiltechnische Zeitschrift 92 (1960), pp. 328–333

2.21 Lauster, E.: *Untersuchungen und Berechnungen zum Wärmehaushalt mechanischer Getriebe* (Investigations and Calculations Relating to Heat Management within Mechanical Transmissions) Stuttgart, Universität, Institut für Maschinenelemente, Dissertation, 1980

Chapter 3 "Mediating the Power Flow"

3.1 Mitschke, M.: *Dynamik der Kraftfahrzeuge. Band A: Antrieb und Bremsung.* (Dynamics of Motor Vehicles. Vol. A: Drive and Braking) 2nd Edition. Berlin; Heidelberg; New York: Springer-Verlag, 1982

3.2 Helling, J.: *Längsdynamik von Kraftfahrzeugen. Umdruck zur Vorl. Kraftfahrzeuge I* (Longitudinal Dynamics of Motor Vehicles. Reprint from the Motor Vehicles I Lecture) Aachen, RWTH, Inst. für Kraftfahrwesen, ika Schriftenreihe Automobiltechnik, 1989

3.3 BOSCH: *Kraftfahrtechnisches Taschenbuch* (Automotive Handbook) 20th Edition, 1987

3.4 Bussien: *Automobiltechnisches Handbuch* (Automobile Engineering Handbook) Vol. 2. 18th Edition. Berlin: Technischer Verlag Herbert Cram, 1965

3.5 Michelin: *Betriebsanleitung für Michelin-Reifen* (Operating Instructions for Michelin Tyres) 28th Edition, 1991

3.6 Wagner, G.: *Berechnung der Verlustleistung von Kfz-Vorgelegegetrieben* (Calculating Power Loss from Motor Vehicle Countershaft Transmissions) VDI-Berichte (1992), No. 977, pp. 175–198

3.7 Förster, H.J.: *Die Kraftübertragung im Fahrzeug vom Motor bis zu den Rädern. Handgeschaltete Getriebe* (Power Transmission from the Engine to the Wheels in Motor Vehicles. Manual Transmissions) Köln: TÜV Rheinland GmbH, 1987

3.8 Regar, K.-N.; Braess, H.-H.; Reister, D.: *Der neue Elektro-3er von BMW – Glied einer langen Entwicklungskette* (BMW's New Electric 3 Series – One Link in a Long Chain of Development) Aachen: 3. Aachener Kolloquium "Fahrzeug- und Motorentechnik", 1991, pp. 551–598

3.9 Nö.: *Elektro-Aufschwung dank Schwungrad-Batterie?* (Boost for Electric Power from the Flywheel Battery?) Automobil Revue (27.01.1994), No. 4

Chapter 4 "Power Conversion: Selecting the Ratios"

4.1 VDI-Richtlinie 2240: *Wellenkupplungen. Systematische Einteilung nach ihren Eigenschaften* (Shaft Couplings. Systematic Classification by their Characteristics) Berlin; Köln: Beuth-Verlag, 1971

4.2 Winkelmann, S.; Harmuth, H.: *Schaltbare Reibkupplungen* (Shiftable Friction Clutches) Berlin; Heidelberg; New York; Tokyo: Springer-Verlag, 1985

4.3 Höhn, B.-R.: *Warum stufenlose Getriebe im Kraftfahrzeug?* (Why Continuously Variable Transmissions in Motor Vehicles?) VDI-Berichte (1990), No. 803, pp. 121–147

4.4 Theissen, M.; Petra, H.; Kern, J.; Klüting, M.: *Künftige Anforderungen an den Fahrzeugantrieb* (Future Requirements to be Met by Vehicle Drives) ATZ Automobiltechnische Zeitschrift 94 (1992), No. 11, pp. 562–570

4.5 Stall, E.: *Entwicklung einer elektronischen Regelung für kraftstoffsparendes Motor-Getriebe-Management im Forschungs-Pkw Uni-Car* (Development of an Electronic Control for Fuel-Saving Engine/Transmission Management in the Uni-Passenger Car Research Vehicle) Aachen, RWTH, Dissertation, 1984

Chapter 5 "Matching Engine and Transmission"

5.1 Mitschke, M.: *Dynamik der Kraftfahrzeuge. Band A: Antrieb und Bremsung* (Dynamics of Motor Vehicles. Vol. A: Drive and Braking) 2nd Edition. Berlin; Heidelberg; New York: Springer-Verlag, 1982

5.2 Fischer, R.; Jürgens, G.: *Einfluß von Pkw-Automatgetrieben auf den Verbrauch* (The Influence of Passenger Car Automatic Transmissions on Consumption) VDI-Berichte (1992), No. 977, pp. 527–544

5.3 DIN 70 030: *Ermittlung des Kraftstoffverbrauchs* (Determining Fuel Consumption) Part 1–3. Beuth-Verlag GmbH, 1990

5.4 Kern, M.: *Fahrstil und Ökonomie* (Driving Style and Economy) Lastauto und Omnibus (1990), No. 10

5.5 Fränkle, G.J.; Haase, F.W.; Woschee, D.; Sommer, H.: *Moderne Mercedes-Benz Nutzfahrzeug-Dieselmotoren für den amerikanischen Markt* (Modern Mercedes-Benz Commercial Vehicle Diesel Engines for the American Market) Aachen: 3. Aachener Kolloquium "Fahrzeug- und Motorentechnik", 1991, pp. 147–196

5.6 Weidner, G.: *Klappern und Rasseln von Fahrzeuggetrieben* (Rattling and Clattering in Vehicle Transmissions) Stuttgart, Universität, Institut für Maschinenelemente, Dissertation, 1991

5.7 Laschet, A.: *Simulation von Antriebssystemen* (Simulating Drive Systems) Berlin; Heidelberg; New York: Springer-Verlag, 1988

5.8 Küçükay, F.: *Dynamik der Zahnradgetriebe* (Dynamics of Gear Mechanisms) Berlin; Heidelberg; New York: Springer-Verlag, 1987

Chapter 6 "Vehicle Transmission Systems: Basic Design Principles"

6.1 Möllers, W.; Müller, R.: *Der Antriebsstrang allradgetriebener Personenkraftwagen* (The Power Train of All-Wheel Drive Passenger Cars) VDI-Berichte (1986), No. 579, pp. 189–213

6.2 Witte, L.; Gorissen, W.: *Beeinflussung der Fahreigenschaften durch den Antrieb* (The Influence of the Drive System on Handling Characteristics) VDI-Berichte (1987), No. 650, pp. 259–282

6.3 Bensinger, J.; Heißling, B.: *Auslegung und Konzeption des Allradantriebs im Audi 80 quattro* (Design and Construction of All-Wheel Drive in the Audi 80 Quattro) ATZ Automobiltechnische Zeitschrift 89 (1987), pp. 247–251

6.4 Heißling, B.; Müller, E.; Stockmar, J.: *Die Entwicklungsstufen von Pkw-Allradantrieben der Audi AG* (The Development Stages in Audi Passenger Car All-Wheel Drives) Aachen: 2. Aachener Kolloquium "Fahrzeug- und Motorentechnik", 1989, pp. 651–685

6.5 Schmidt, R.; Diessner, G.; Uhlig, G.; Otto, H.: *Antriebskonzept, Bauteile und Eigenschaften des VW Golf syncro* (Drive Concept, Components and Characteristics of the VW Golf Syncro) ATZ Automobiltechnische Zeitschrift 88 (1986), pp. 477–485

6.6 Brügel, E.; Klein, H.; Willibald, K.: *Die Visko-Kupplung als Bauelement in Antriebssystemen* (The Visco-Clutch as a Component in Drive Systems) VDI-Berichte (1986), No. 579, pp. 321–343

6.7 Peschke, W.: *Die Wirkungsweise der Visko-Kupplung und ihr Einfluß auf die Traktion eines Allradfahrzeugs* (The Principle of Operation of the Visco-Clutch and its Effect on the Traction of an All-Wheel Drive Vehicle) Hannover, Universität, Dissertation, 1989

6.8 Looman, J.: *Zahnradgetriebe* (Gear Mechanisms) 2nd Edition. Berlin; Heidelberg; New York; London; Paris; Tokyo: Springer-Verlag, 1988

6.9 Lechner, G.: *Getriebelehre II. Angewandte Getriebelehre* (Theory of Gears II. Applied Theory of Gears) Berlin: Walter de Gruyter & Co, 1970

6.10 Ott, A.: *Zur systematischen Synthese mehrgängiger Umlaufräder-Schaltgetriebe* (On the Systematic Synthesis of Multiple Planetary Gear Manual Transmissions) ZF-Sonderdruck: ATZ Automobiltechnische Zeitschrift 70 (1968), No. 1, 3, 4

6.11 Blumenstock, K.-U.: *Von Sonnen und Planeten* (Of Suns and Planets) mot Spezial, 1990, pp. 72–77

6.12 *ZF-Firmenprospekt* (ZF Company Brochure)

6.13 Förster, H.J.: *Automatische Fahrzeuggetriebe* (Automatic Automotive Transmissions) Berlin; Heidelberg; New York; Hong Kong; Barcelona; Budapest: Springer-Verlag, 1990

6.14 Jarchow, F.: *Stufenlos wirkende hydrostatisch-mechanische Lastschaltgetriebe* (Infinitely Variable Hydrostatic/Mechanical Powershift Transmissions) VDI-Berichte (1991), No. 878, pp. 623–647

6.15 Jarchow, F.: *Die Wirkungsweise von Schubgliederketten* (The Mode of Operation of Thrust Link Chains) Konstruktion 41 (1989), pp. 73–81

6.16 *Firmenschrift Van Doorne's Transmissie b.v.* (Van Doorne's Transmissie b.v. Company Publication)

6.17 Köpf, P.: *Entwicklungstendenzen bei Nkw-Getrieben* (Development Trends in Commercial Vehicle Transmissions) VDI-Berichte (1991), No. 878, pp. 21–40

6.18 *ZF-Firmenprospekt "Ecomat"* (ZF "Ecomat" Company Brochure)

6.19 Wycklicky, R.: *1000 PS-Lastschalt-Automatgetriebe für Nutz- und Spezialfahrzeuge* (1000 hp Powershift Automatic Transmissions for Commercial Vehicles and Special-Purpose Vehicles) VDI-Berichte (1991), No. 878, pp. 173–188

6.20 *Voith-Firmenprospekt "SHL"* (Voith "SHL" Company Brochure)

6.21 Ehrhard, P.: *Das elektrische Getriebe von Magnet-Motor für Pkw und Omnibusse* (The Electrical Reluctance Motor Transmission for Passenger Cars and Buses) VDI-Berichte (1991), No. 878, pp. 611–622

6.22 Altmann, F.G.: *Ausgleichsgetriebe für Kraftfahrzeuge* (Differential Gears for Motor Vehicles) VDI-Zeitung 84 (1940), No. 31, pp. 545–551

6.23* Behles, F.: *Die Triebwerksanordnung beim Frontantrieb* (Power Unit Layout with Front-Wheel Drive) ATZ Automobiltechnische Zeitschrift 71 (1969), pp. 35–39

6.24* Behles, F.; Barske, H.: *Die Frontantriebsentwicklung bei Audi, NSU, Auto Union und DKW* (The Development of Front-Wheel Drive at Audi, NSU, Auto Union and DKW) ATZ Automobiltechnische Zeitschrift 83 (1981), pp. 487–495 Part 1; ATZ Automobiltechnische Zeitschrift 83 (1981), pp. 557–564 Part 2

6.25* Behles, F.; Schneeweiss, M.: Front Wheel Drive Power Trains and Suspensions. SAE-Paper 810422

6.26* Lorenz, R.: *Variable Verteilung von Längs- und Querkräften beim Bremsen mit Allradantrieb* (Variable Distribution of Axial and Transverse Forces when Braking with All-Wheel Drive) 20. Mitarbeiterveröffentlichung Daimler-Benz AG 1991, pp. 1–20

6.27 Elser, W.: *Optimierung stufenloser Toroidgetriebe für Pkw* (Optimising Infinitely Variable Toroidal Mechanisms for Passenger Cars) VDI-Berichte (1998), No. 1393, pp. 513–526

6.28 James, I.B.; Greenwood, C.J.: Pressure Control Issues for Higher Viscosity Traction Fluids in a Torque Controlled IVT. VDI-Berichte (1998), No. 1393, pp. 695–702

6.29 Tenberge, P.: *Toroidgetriebe mit verbesserten Kennwerten* (Toroidal Mechanisms with Improved Parameters) VDI-Berichte (1998), No. 1393, pp. 703–724

Chapter 7 "Design of Gearwheel Transmissions for Vehicles"

7.1 Konzernnorm ZFN 201: *Zahnradschäden, Begriffsbestimmung* (Gearwheel Damage, Definition) 1990

7.2 DIN 3990: *Tragfähigkeitsberechnung von Stirnrädern* (Capacity Calculation for Spur Gear Drives) Part 1–5, 12, 21, 31, 41. Berlin: Beuth-Verlag GmbH, Dezember 1987

7.3 Lechner, G.: *Berechnung der Freßtragfähigkeit von Stirn- und Kegelrädern.* (Calculating the Scuffing Resistance of Spur Gears and Bevel Gears) Technisch-wissenschaftliche Veröffentlichung der ZF AG. 1973, Heft 8

7.4 Lechner, G.: *Die Freß-Grenzlast bei Stirnrädern aus Stahl* (The Scuff-Limited Load for Steel Spur Gears) München, Technische Universität, Dissertation, 1966

7.5 Buxbaum, O.: *Betriebsfestigkeit* (Operational Integrity) Düsseldorf: Verlag Stahleisen GmbH, 1986

7.6 Haibach, E.: *Betriebsfestigkeit* (Operational Integrity) Düsseldorf: VDI-Verlag, 1989

7.7 Borenius, G.: *Zur rechnerischen Schädigungsakkumulation in der Erprobung von Kraftfahrzeugteilen bei stochastischer Belastung mit variabler Mittellast* (Numerical Damage Accumulation in the Testing of Motor Vehicle Parts with Stochastic Loading with Variable Centre Load) Stuttgart, Universität, Dissertation, 1990

7.8 Miner, M.A.: Cumulative Damage in Fatigure Trans. New York: ASME (American Society of Mechanical Engineers), Journal of Applied Mechanics 4 (1937), pp. A160–A162

7.9 Palmgren, A.: *Die Lebensdauer von Kugellagern* (The Service Life of Ball Bearings) Düsseldorf: VDI-Zeitschrift 68 (1924), No. 14, pp. 339–341

7.10 Hirschmann, K.H.: *Methodische Lebensdauerberechnung bei Maschinenelementen. Vortrag Nr. 3. Lehrgang Nr. 17019/60.175: Zuverlässigkeitsanalyse von Systemen* (Methodical Calculation of Service Life for Machine Elements. Lecture No. 3, Course No. 17019/60.175: Reliability analysis of systems) TAE-Technische Akademie Esslingen, Weiterbildungszentrum, 1993

7.11 Köpf, P.: *Entwicklungstendenzen bei Nkw-Getrieben* (Development Trends in Commercial Vehicle Transmissions) VDI-Berichte (1991), No. 878, pp. 21–40

7.12 Niemann, G.; Winter, H.: *Maschinenelemente. Band II: Getriebe* (Machine Elements Vol. II: Gear Mechanisms) Berlin; Heidelberg; New York: Springer-Verlag, 1983

7.13 Nill, R.: *Das Schwingungsverhalten loser Bauteile in Fahrzeuggetrieben* (The Vibration Behaviour of Loose Components in Vehicle Transmissions) Stuttgart, Universität, Dissertation, 1986

7.14 Weidner, G.; Lechner, G.: *Klapper- und Rasselgeräusche in Fahrzeuggetrieben* (Rattling and Clattering Noise in Vehicle Transmissions) ATZ Automobiltechnische Zeitschrift 92 (1990), No. 6, pp. 320–326

7.15 Weidner, G.: *Klappern und Rasseln von Fahrzeuggetrieben* (Rattling and Clattering in Vehicle Transmissions) Stuttgart, Universität, Dissertation, 1991

7.16 Reik, W.: *Torsionsschwingungen und Getriebegeräusche* (Torsional Vibration and Transmission Noise) Automobil-Industrie 32 (1987), No. 1, pp. 37–42

7.17 Gosdin, M.: *Analyse und Optimierung des dynamischen Verhaltens eines Pkw-Antriebsstrangs* (Analysis and Optimisation of the Dynamic Behaviour of a Passenger Car Power Train) München, Technische Universität, Dissertation, 1985

7.18 Schöpf, H.-J.; Jürgens, G.; Fischer, R.: *Optimierung der Komforteigenschaften des Triebstrangs von Mercedes-Benz-Fahrzeugen mit Schaltgetriebe* (Optimising the Comfort Aspects of the Power Train of Mercedes-Benz Manual Transmission Vehicles) ATZ Automobiltechnische Zeitschrift 91 (1989), No. 10, pp. 568–575

7.19 Goto, Y.; Yagi, Y.; Morimoto, Y.; Kawasaki, M.: Shift Feel in Manual Transmission – An Analysis of Unsmooth Shifting and Gear Clashing. JSAE Review **9** (4) (1988)

7.20 Finsterhölzl, H.; Kelle, W.: *Leerlaufqualität in der Fahrzeugbeurteilung* (Idling Quality in Vehicle Assessment) ATZ Automobiltechnische Zeitschrift 92 (1990), No. 5, pp. 268–282

7.21* VDI-Richtlinie 2159: *Emissionskennwerte technischer Schallquellen – Getriebegeräusche* (Emission Characteristic Values of Technical Noise Sources – Transmission Noise) Düsseldorf: VDI-Verlag GmbH, 1985

7.22* Küçükay, F.: *Dynamik der Zahnradgetriebe* (Dynamics of Gear Mechanisms) Berlin; Heidelberg; New York: Springer-Verlag, 1987

7.23* Henn, H.; Sinambari, G.R.; Falle, M.: *Ingenieurakustik* (Engineering Acoustics) Wiesbaden; Braunschweig: Vieweg und Sohn, 1984

Chapter 8 "Specification and Design of Shafts"

8.1 Truong, H.: *Strukturorientiertes Modellieren, Optimieren und Identifizieren von Mehrkörpersystemen* (Structure-Oriented Modelling, Optimising and Identifying Multi-Body Systems) Stuttgart, Universität, Institut für Maschinenelemente, Dissertation

8.2 DRESP, *Schwingungsberechnungsprogramm* (Vibration Calculation Program) Aachen, RWTH

8.3 Niemann, G.: *Maschinenelemente, Band I* (Machine Elements, Vol. I.) 2nd Edition. Berlin; Heidelberg: Springer-Verlag, 1975

8.4 Wellinger, K.; Dietmann, H.: *Festigkeitsberechnung, Grundlagen und technische Anwendung* (Strength Calculation, Basic Elements and Technical Application) Stuttgart: Kröner Verlag, 1976

8.5 Kußmaul, K.: *Festigkeitslehre I. Vorlesungsmanuskript* (Strength Theory I. Lecture Manuscript) Stuttgart, Universität, 1985

8.6 Rühl, K.H.: *Die Tragfähigkeit metallischer Baukörper in Bautechnik und Maschinenbau* (The Load-Bearing Capacity of Metal Structures in Construction and Machine Construction) Berlin: Verlag Wilhelm Ernst & Sohn, 1952

8.7 Siebel, E.; Ludwig, N.: *Handbuch der Werkstoffprüfung. Band 2: Die Prüfung der metallischen Werkstoffe.* (Materials Testing Handbook. Volume 2: Testing Metal Materials) 2nd Edition. Berlin: Springer-Verlag, 1955

8.8 Thum, A.; Peterson, C.; Svenson, O.: *Verformung, Spannung und Kerbwirkung* (Deformation, Stress and Stress Concentration) Düsseldorf: VDI-Verlag, 1960

8.9 Matek, W.; Muhs, D.; Wittel, H.: *Roloff/Matek Maschinenelemente* (Roloff/Matek Machine Elements) 11th Edition. Braunschweig; Wiesbaden: Fried. Vieweg & Sohn, 1987

8.10 Dietrich; Stahl: *Matrizen und Determinanten und ihre Anwendungen in Technik und Ökonomie* (Matrices and Determinants and their Applications in Engineering and Economics) Leipzig: VEB, 1978

8.11 DIN 17200: *Werkstoffkennwerte* (Material Characteristic Values) Berlin: Beuth-Verlag GmbH

8.12* DIN 3960: *Verzahnungsterminologie* (Toothing Terminology) Berlin: Beuth-Verlag GmbH

8.13* Göldner, H.; Holzweißig, F.: *Leitfaden der Technischen Mechanik. Statik – Festigkeitslehre – Kinematik – Dynamik* (Guide to Engineering Mechanics. Statics – Strength Theory – Kinematics – Dynamics) 8th Edition. Darmstadt: Steinkopff-Verlag, 1984

8.14* Hähnchen, R.; Decker, K.-H.: *Neue Festigkeitsberechnung für den Maschinenbau* (New Strength Calculation for Machine Construction) München: Carl Hanser Verlag, 1967

8.15* Kleinbach, K.: *Qualitätsbeurteilung von Kegelradsätzen durch integrierte Prüfung von Tragbild, Einflankenwälzabweichung und Spielverlauf* (Quality Assessment of Bevel Gear Sets by Integrated Inspection and Testing of Contact Pattern, Single Flank Working Devitation and Backlash Pattern) Stuttgart, Universität, Institut für Maschinenelemente, Dissertation, 1989

8.16* Magnus, K.; Müller, H.H.: *Grundlagen der Technischen Mechanik* (Basic Elements of Engineering Mechanics) Stuttgart: Teubner Studienbücher, 1974

8.17* Schmidt, F.: *Berechnung und Gestaltung von Wellen* (Calculating and Designing Shafts) 2nd Edition. Berlin; Heidelberg; New York: Springer-Verlag, 1967

8.18* Tochtermann, W.; Bodenstein, F.: *Konstruktionselemente des Maschinenbaues* (Engineering Design Elements in Machine Construction) Part 2. 9th Edition. Berlin; Heidelberg: Springer-Verlag, 1979

Chapter 9 "Gear Shifting Mechanisms, Layout and Design of Synchronisers"

9.1 Looman, J.: *Zahnradgetriebe* (Gear Mechanisms) 2nd Edition. Berlin; Heidelberg; New York; London; Paris; Tokyo: Springer-Verlag, 1988

9.2 Paulsen, L.: *Entwicklungen und Perspektiven des Nfz-Triebstrangmanagements* (Developments and Perspectives in Commercial Vehicle Power-Train Management) VDI-Berichte (1991), No. 878, pp. 97–115

9.3 Förster, H.J.: *Automatische Fahrzeuggetriebe* (Automatic Vehicle Transmissions) Berlin; Heidelberg; New York; Hong Kong; Barcelona; Budapest: Springer-Verlag, 1990

9.4 Köpf, P.: *Die Synchronisierung von Fahrzeuggetrieben* (Synchronising Vehicle Transmissions) Automobil Revue (24.04.1980), No. 18, pp. 61–67

9.5 Goto, Y.; Yagi, Y.; Morimoto, Y.; Kawasaki, M.: Shift Feel in Manual Transmission – An Analysis of Unsmooth Shifting and Gear Clashing. JSAE Review **9** (4) (1988)

9.6 Schröper, R.: *Temperaturmessungen an Synchronisierungen* (Temperature Measurements and Synchronisers) Stuttgart, Universität, Institut für Maschinenelemente, Diplomarbeit, unveröffentlicht, 1981

9.7 Winkelmann, S.; Harmuth, H.: *Schaltbare Reibkupplungen* (Shiftable Friction Clutches) Berlin: Springer-Verlag, 1985

9.8 *Berechnungsmappe der Zahnradfabrik Friedrichshafen* (Calculation Reference Folder of Zahnradfabrik Friedrichshafen)

9.9 Wagner, D.: *Neue Reibmaterialien und Konzepte für Einfach- und Mehrfachsynchronisierungen* (New Friction Materials and Concepts for Single and Multiple Synchronisers) ATZ Automobiltechnische Zeitschrift 95 (1993), No. 7/8, pp. 380–387

9.10 Rastinger, B.: *Die neuen Handschaltgetriebe in den BMW-Baureihen 8, 5 und 3* (The New Manual Transmissions in the BMW Series 8, 5 and 3) VDI-Berichte (1991), No. 878, pp. 75–95

9.11 Drott, H.; Oppertshäuser, E.: *3-Konus-Synchronisierungen in Opel-PKW-Schaltgetrieben* (3-Cone Synchronisers in Opel Passenger Car Manual Transmissions) VDI-Berichte (1991), No. 878, pp. 293–309

9.12 Winter, H.; Pflaum, H.: *Synchronisierungen in Kfz-Schaltgetrieben – Einfluß der Werkstoffpaarung auf das Reibungsverhalten* (Synchronisers in Motor Vehicle Manual Transmissions – Influence of Material Mating on Friction Behaviour) Antriebstechnik 30 (1991), No. 8, pp. 50–58

9.13 Goll, S.; Schröper, R.: *Das Schaltelement "Synchronisierung" – eine hochentwickelte Baugruppe bei PKW- und NKW-Getrieben* (The "Synchroniser" Shifting Element – a Sophisticated Assembly in Passenger Car and Commercial Vehicle Transmissions) VDI-Bericht (1986), No. 579, pp. 287–300

9.14 Wulff, G.; Reusing, G.; Schiebold, S.; Bauch, D.: *Integration von elektrischen und mechanischen Antriebskomponenten* (Integrating Electrical and Mechanical Drive Components) Aachen: 4. Aachener Kolloquium "Fahrzeug- und Motorentechnik", 1993, pp. 697–710

9.15 Ersoy, M.: *Entwicklungstendenzen für Getriebe-Außenschaltungen* (Development Trends in Transmission External Shift Systems) VDI-Berichte (1998), No. 1393, pp. 273–286

Chapter 10 "Hydrodynamic Clutches and Torque Converters"

10.1 Stüper, J.: *Automatische Automobilgetriebe* (Automatic Automobile Transmissions) Berlin; Göttingen; Heidelberg: Springer-Verlag, 1965

10.2 Voith (Hrsg.): *Hydrodynamik in der Antriebstechnik* (Hydrodynamics in Drive Engineering) Vereinigte Fachverlage Krauskopf Ingenieur Digest, 1987

10.3 Förster, H.J.: *Automatische Fahrzeuggetriebe* (Automatic Vehicle Transmissions) Berlin; Heidelberg; New York; Hong Kong; Barcelona; Budapest: Springer-Verlag, 1990

10.4 Jauch, F.: *Beitrag zur Optimierung des Antriebsstranges von Kraftfahrzeugen unter dem Aspekt von Kraftstoffverbrauch, Transportleistung und Lastkollektiven* (Conribution to Optimising the Power Train of Motor Vehicles in Terms of Fuel Consumption, Transportation Efficiency and Load Profiles) Stuttgart, Universität, Institut für Maschinenelemente, Dissertation, 1989

10.5* Kickbusch, E.: *Föttinger-Kupplungen und Föttinger-Getriebe* (Föttinger Clutches and Föttinger Transmissions) Berlin; Göttingen; Heidelberg: Springer-Verlag, 1963

10.6* Wolf, M.: *Strömungskupplungen und Strömungswandler* (Fluid Clutches and Hydrodynamic Transmissions) Berlin; Göttingen; Heidelberg: Springer-Verlag, 1962

10.7* Looman, J.: *Zahnradgetriebe* (Gear Mechanisms) 2nd Edition. Berlin; Heidelberg; New York; London; Paris; Tokyo: Springer-Verlag, 1988

10.8* VDI-Richtlinie 2153: *Hydrodynamische Getriebe* (Hydrodynamic Transmissions) Berlin: Beuth-Verlag GmbH, 1974

Chapter 11 "Notes on the Design and Configuration of Further Vehicle Transmission Design Elements"

11.1 Freund, H.; Stubenrauch, A.: *Entwicklungen von Wälzlagerungen in Nutzfahrzeug-Schaltgetrieben* (Developments in Roller Bearings in Commercial Vehicle Manual Transmissions) Antriebstechnik (1987), No. 4, pp. 28–39

11.2 *FAG Wälzlager Standardprogramm* (FAG Standard Range Roller Bearings) Katalog WL 41510/2 DB, FAG Kugelfischer Georg Schäfer KGaA, Schweinfurt, 1989

11.3 Lorösch, H.-K.: *Wirtschaftliche, umweltschonende Wälzlagerungen durch Vermeiden von Überdimensionierung* (Economic, Environmentally Acceptable Roller Bearings by Avoiding Over-Dimensioning) Antriebstechnik 32 (1993), No. 3, pp. 89–93

11.4 Brändlein, J.; Lorösch, H.-K.: *Schmutzgeschützte Lager in Kraftfahrzeuggetrieben* (Clean Bearings in Vehicle Transmissions) VDI-Berichte (1986), No. 579, pp. 253–268

11.5 Pittroff, H.: *Riffelbildung bei Wälzlagern infolge Stillstandserschütterungen* (Grooving in Roller Bearings as a Result of Shocks at Rest) Schweinfurt: Eigenverlag SKF-Kugellagerfabriken GmbH, 1961

11.6 *Wälzlager INA 305* (Roller Bearing INA 305) INA Wälzlager Schaeffler KG, Herzogenaurach, 1989

11.7 Lundberg, S.: *Anstellen von Kegelrollenlagern in Pkw-Getrieben mit Leichtmetallgehäuse* (Adjusting Tapered Roller Bearings in Passenger Car Transmissions with Light-Alloy Housings) SKF-Kugellager-Zeitschrift 193, Jahrgang 52

11.8 Dornhöfer, G.: *Lebensdauerschmierung von Kfz-Aggregaten unter Berücksichtigung der Umweltschonung* (Life-Time Lubrication of Motor Vehicle Units Taking Into Account Environmental Considerations) Tribologie und Schmierungstechnik (1994), No. 2, pp. 86–90

11.9 Lechner, G.: *Untersuchungen zur Schmierfilmbildung an Zahnrädern* (Investigations into Lubricant Film Formation on Gearwheels) VDI-Zeitschrift 111 (1969), No. 4, pp. 269–274

11.10 Lechner, G.: *Berechnung der Freßtragfähigkeit von Stirn- und Kegelrädern* (Calculating the Scuffing Resistance of Spur Gears and Bevel Gears) Technisch-wissenschaftliche Veröffentlichungen der Zahnradfabrik Friedrichshafen AG, 1973, Heft 8

11.11 Block, H.: Fundamental Mechanical Aspects of Boundary Lubrication. JSAE **46** (2) (1940)

11.12 Lane, T.B.; Hughes, J.K.: A Practical Application of the Flash Temperature Hydropotesis of Gear Lubrication. Third World Petrol Congr., 1951, pp. 7

11.13 Niemann, G.: *Schmierfilmbildung, Verlustleistung und Schadensgrenzen bei Zahnrädern mit Evolventenverzahnung* (Lubricant Film Formation, Power Loss and Damage Limits in Gearwheels with Involute Toothing) VDI-Zeitschrift 97 (1955), No. 10

11.14 Eckhardt, F.: *Stationäre Zahnradgetriebe. Schmierung und Wartung* (Stationary Gear Mechanisms. Lubrication and Maintenance) Hamburg: Eigenverlag der Mobil Oil AG

11.15 Khosrawi, M.A.: *Untersuchung über die Reaktionsschichtbildung bei hochbelastbaren Schmierölen* (Investigation Into Reaction Layer Formation in Highly Stressed Lubricants) Stuttgart, Universität, Staatliche Materialprüfungsanstalt (MPA), Dissertation, 1983

11.16 Uetz, H.; Föhl, J.; Sommer, K.: *Einfluß v. Reaktionsschichten auf das tribologische Verhalten langsamlaufender Wälzpaarungen* (The Influence of Reaction Layers on the Tribological Behaviour of Slow-Running Roller Pairs) Antriebstechnik 28 (1989), No. 10, pp. 57–62

11.17 *Additive für Schmierstoffe. Reihe Tribotechnik* (Additives for Lubricants. Tribology Series) Vol. 2. Hannover: Curt R. Vincentz-Verlag

11.18 Collenberg, F.: *Untersuchungen zur Freßtragfähigkeit schnellaufender Stirnradgetriebe* (Investigations Into the Scuffing Resistance of Fast-Running Spur Gear Mechanisms) München, Technische Universität, Lehrstuhl Maschinenelemente, Dissertation, 1991

11.19 Kürzeder, H.: *Verlängerung der Lebensdauer neuer und bereits beschädigter Getriebe durch Spezialschmierstoffe* (Extending the Service Life of New and Damaged Transmissions by Using Special Lubricants) Antriebstechnik 26 (1987), No. 7, pp. 18–25

11.20 DIN 51509: *Auswahl von Schmierstoffen für Zahnradgetriebe* (Selecting Lubricants for Gear Mechanisms) Berlin: Beuth-Verlag GmbH, Juni 1976

11.21 Oster, P.: *Beanspruchung der Zahnflanken unter Bedingungen der Elastohydrodynamik* (Stressing Tooth Faces Under Conditions of Elasto-Hydrodynamics) München, Technische Universität, Lehrstuhl Maschinenelemente, Dissertation, 1982

11.22 Sonntag, W.: *Vermeidung von Zahnradschäden durch geeignete Schmieröle* (Avoiding Gearwheel Damage by Using Suitable Lubricants) Berlin: Tribologie und Schmierungstechnik 19 (1988), No. 10, pp. 292–295

11.23 Joachim, F.J.; Thies, H.E.; Brügel, E.: *Lebensdauerschmierung bei Fahrzeuggetrieben* (Life-Time Lubrication for Vehicle Transmissions) Berlin: Tribologie und Schmierungstechnik 39 (1992), No. 6, pp. 317–324

11.24 Meyer, K.; Kloss, H.; Reichelt, F.: *Übertragbarkeit von Freßverschleißergebnissen aus Modellapparaturen auf praxisnahe Bedingungen* (Transferability of Gear Scuffing Results from Model Apparatus to Practical Conditions) Berlin: Tribologie und Schmierungstechnik. Part I: 21 (1990), No. 10, pp. 299–302; Part II: 22 (1991), No. 3, pp. 68–71; Part III: 22 (1991), No. 4, pp. 105–107

11.25 Winter, H.; Michaelis, K.: *Freßtragfähigkeit von Stirnradgetrieben. Teil I: Grundlagen zum Ansatz einer Integraltemperatur* (Scuffing Resistance of Spur Gear Mechanisms. Part I: Basic Principles for Estimating an Integral Temperature) Antriebstechnik 14 (1975), No. 7, pp. 405–409

11.26 Niemann, G.; Lechner, G.: *Die Freßgrenzlast bei Stirnrädern aus Stahl* (Maximum Scuffing Load for Steel Spur Gears) Erdöl und Kohle 20 (1967), No. 2, pp. 96–106

11.27 *Getriebeöl im Kraftfahrzeug* (Gear Oil in Motor Vehicles) Mitteilung des SHELL Technischen Dienstes, No. 01.08.63

11.28 *Molykote-Handbuch* (Molykote Handbook) 1991

11.29 Wender, B.; Lecong, S.: *Gestaltung von Versteifungsrippen und Bohrungen an Getriebegehäusen* (Designing Bracing Ribs and Drill Holes in Transmission Housings) Konstruktion 43 (1991), pp. 365–370

11.30 Jäckle, M.: *Entlüftung von Getrieben* (Venting transmissions) Stuttgart, Universität, Institut für Maschinenelemente, Dissertation, 1994

11.31 Krieg, W.-E.: *Untersuchungen an Gehäuseabdichtungen von hochbelasteten Getrieben* (Investigations into Housing Seals for Highly Stressed Transmissions) Stuttgart, Universität, Institut für Maschinenelemente, Dissertation, 1993

11.32 Hettich, V.; v. Eiff, H.; Lechner, G.: *Untersuchung von Flächendichtungen unter dynamischer Scherbeanspruchung* (Investigation of Gaskets under Dynamic Shearing Stress) Antriebstechnik 31 (1992), No. 4, pp. 74–79

11.33 Müller, H.-K.; *Abdichtung bewegter Maschinenteile* (Sealing Moving Machine Parts) Waiblingen: Medienverlag Müller, 1990

11.34 BOSCH – *Kraftfahrtechnisches Taschenbuch* (Pocket Book of Automotive Engineering) 20th Edition. Düsseldorf: VDI-Verlag, 1987

11.35 Göhring, E.: *Bremsleistung schwerer Nutzfahrzeuge* (Braking Performance of Heavy Commercial Vehicles) Aachen: 2. Aachener Kolloquium "Fahrzeug- und Motorentechnik", 1989, pp. 397–425

11.36 Förster, H.J.: *Automatische Fahrzeuggetriebe und Retarder für Nutzfahrzeuge* (Automatic Vehicle Transmissions and Retarders for Commercial Vehicles) ATZ Automobiltechnische Zeitschrift 76 (1974), No. 5, pp. 142–149

11.37* Schwab, M.; Härdtle, W.; Heinzelmann, K.-F.: *Der ZF-Intarder – Ein integrierter hydrodynamischer Retarder für die neue Ecosplit-Getriebereihe* (The ZF Intarder – An Integrated Hydrodynamic Retarder for the New Ecosplit Transmission Range) ATZ Automobiltechnische Zeitschrift 95 (1993), No. 5, pp. 250–255

11.38* N.N: *Was können die Retarder?* (What Can Retarders Do?) Nutzfahrzeug (1990), No. 9, pp. 29–30

11.39* DIN 51512: *SAE-Viskositätsklassen für Schmieröle für Kraftfahrzeuggetriebe* (SAE Viscosity Grades for Motor Vehicles Transmission Lubricants) Berlin: Beuth-Verlag GmbH, Mai 1988

11.40* Michaelis, K.: *Sonderkapitel Getriebe-Tribologie des Zahnrades. Skriptum zur Wahlvorlesung* (Gearwheel Transmission Tribology – Special Chapter. Script of Selective Lecture) München, Technische Universität, Lehrstuhl für Maschinenelemente

11.41* Schmid, E.: *Handbuch der Dichtungstechnik* (Handbook of Sealing Technology) Grafenau/Württemberg: Expert-Verlag, 1981

Chapter 12 "Typical Designs of Vehicle Transmissions"

12.1 Rastinger, B.: *Die neuen Schaltgetriebe für die BMW-Baureihen 8, 5 und 3* (The New Manual Transmissions for the BMW Series 8, 5 and 3) ATZ Automobiltechnische Zeitschrift 93 (1991), No. 11, pp. 698–705

12.2 Barnbeck, A.; Drott, H.; Helms, G.: *Das neue Opel-Sechsgang-Getriebe* (The New Opel 6-Speed Transmission) ATZ Automobiltechnische Zeitschrift 95 (1993), No. 6, pp. 288–292

12.3 Schöpf, H.-J.; Jürgens, G.; Pickard, J.: *Das neue Fünfgang-Automatgetriebe von Mercedes-Benz* (The New Mercedes-Benz 5-Speed Automatic Transmission) ATZ Automobiltechnische Zeitschrift 91 (1989), No. 9, pp. 444–453

12.4 Eggert, U.: *CVT – Elektronische Regelung und Fahrdynamik* (CVT – Electronic Control and Vehicle Dynamics) VDI-Berichte (1990), No. 803, pp. 197–234

12.5 Simon, E.: *Serienerfahrungen mit dem CTX-Getriebeautomaten im Ford Fiesta* (Production Experience with the CTX Automatic Transmissions in the Ford Fiesta) VDI-Berichte (1990), No. 803, pp. 263–280

12.6 Ledwinka, E.: *Der Steyr-Puch-"Pinzgauer"-Geländewagen* (The Steyr-Puch "Pinzgauer" Off-Road Vehicle) ATZ Automobiltechnische Zeitschrift 74 (1972), pp. 228–235

12.7 Jaskiewicz, Z.: *Mosty Napedowe* (Drive axles) Warszawa: Wydawnictwo komunikacji i lacnosci, 1977

12.8 Looman, J.: *Zahnradgetriebe* (Gear Mechanisms) 2nd Edition. Berlin; Heidelberg; New York; London; Paris; Tokyo: Springer-Verlag, 1988

12.9 Langenbeck, K.; Körner, T.: *Hydrostatisch selbstsperrende Stirnraddifferentiale* (Hydrostatically Self-Locking Spur Gear Differentials) VDI-Berichte (1988), No. 672, pp. 359–378

12.10 Langenbeck, K.; Lenzen, O.: *Hydrostatisch selbstsperrende Planetengetriebe – Auslegung, Konstruktion, Temperaturkompensation* (Hydrostatically Self-Locking Planetary Gears – Design, Engineering, Temperature Compensation) VDI-Berichte (1991), No. 905, pp. 37–56

12.11 Langenbeck, K.; Lenzen, O.: *Temperaturkompensation hydrostatisch selbstsperrender Planetengetriebe* (Temperature Compensation of Hydrostatically Self-Locking Planetary Gear Mechanisms) Antriebstechnik 290 (1990), No. 9, pp. 88–92

12.12 Schmidt, R.; Diessner, G.; Uhlig, G.; Otto, H.: *Antriebskonzept, Bauteile und Eigenschaften des VW Golf Syncro* (Drive Concept, Components and Characteristics of the VW Golf Syncro) ATZ Automobiltechnische Zeitschrift 90 (1988), No. 10, pp. 535–542

12.13 Bensinger, J.; Heißling, B.: *Auslegung und Konzeption des Allradantriebs im Audi 80 quattro* (Design and Construction of the All-Wheel Drive in the Audi 80 Quattro) ATZ Automobiltechnische Zeitschrift 89 (1987), No. 5, pp. 247–251

12.14 Gaus, H.; Schöpf, H.-J.: ASD, ASR und 4MATIC: *Drei Systeme im "Konzept Aktive Sicherheit" von Daimler-Benz. Teil 1* (Three Systems from Daimler-Benz Contributing to "Active Safety". Part 1) ATZ Automobiltechnische Zeitschrift 88 (1986), No. 5, pp. 273–284

12.15 Gaus, H.; Schöpf, H.-J.: ASD, ASR und 4MATIC: *Drei Systeme im "Konzept Aktive Sicherheit" von Daimler-Benz. Teil 2* (Three Systems from Daimler-Benz Contributing to "Active Safety". Part 2) ATZ Automobiltechnische Zeitschrift 88 (1986), No. 6, pp. 345–349

12.16 Stork, W.; Dietrich, C.; Rittinghaus, H.: *BMW 325i-4 – Die Allradvariante der 3er-Baureihe* (BMW 325i-4 – The All-Wheel Variant of the 3 Series) ATZ Automobiltechnische Zeitschrift 87 (1985), No. 9, pp. 395–400

12.17 Höhn, B.-H.; Nowatschin, K.: *Der Antriebsstrang des Audi V8* (The Power Train of the Audi V8) ATZ Automobiltechnische Zeitschrift 91 (1989), No. 10, pp. 582–587

12.18 Duckstein, H.; Leinfellner, H.; Sommer, H.-D.: *Der VW-Transporter Syncro* (The VW Transporter Syncro) ATZ Automobiltechnische Zeitschrift 87 (1985), No. 9, pp. 405–417

12.19 Pilgrim, R.; Rohardt, H.; Srock, R.: *911 Carrera 4, der Allrad-Porsche. Teil 1* (911 Carrera 4, the All-Wheel Drive Porsche. Part 1) ATZ Automobiltechnische Zeitschrift 90 (1988), No. 11, pp. 615–625

12.20 Söffge, F.; Dietz, M.; Bofinger, G.; Lawitzki, S.: *911 Carrera 4, der Allrad-Porsche. Teil 2* (911 Carrera 4, the All-Wheel Drive Porsche. Part 2) ATZ Automobiltechnische Zeitschrift 91 (1989), No. 1, pp. 15–24

12.21 Manthey, D.; Fleischmann, O.: *Das Allradsystem des Opel Vectra* (The All-Wheel Drive System of the Opel Vectra) Aachen: 2. Aachener Kolloquium "Fahrzeug- und Motorentechnik", 1989, pp. 687–702

12.22 Müller, R.; Witte, L.: *PDS – Ein neues Antriebssystem für Allradfahrzeuge. Teil 1* (PDS – A New Drive System for All-Wheel Drive Vehicles. Part 1) ATZ Automobiltechnische Zeitschrift 89 (1987), No. 11, pp. 595–598

12.23 Müller, R.; Witte, L.: *PDS – Ein neues Antriebssystem für Allradfahrzeuge. Teil 2* (PDS – A New Drive System for All-Wheel Drive Vehicles. Part 2) ATZ Automobiltechnische Zeitschrift 89 (1987), No. 12, pp. 673–678

12.24 Brodbeck, P.: *Konstruktion einer hydrodynamischen Allradkupplung* (Engineering Design of the Hydrodynamic All-Wheel Drive Clutch) Stuttgart, Universität, Institut für Maschinenelemente, Studienarbeit, unveröffentlicht

12.25 Lanzer, H.: *Die Viskomatic – Ein Vierradantrieb der dritten Generation* (The Viskomatic – A Third-Generation Four-Wheel Drive) ATZ Automobiltechnische Zeitschrift 88 (1986), No. 9, pp. 477–485

12.26* Bantle, M.; Bott, H.: *Der Porsche Typ 959-Gruppe B – Ein besonderes Automobil. Teil 2* (The Porsche Type 959 Group B – A Special Automobile. Part 2) ATZ Automobiltechnische Zeitschrift 88 (1986), No. 6, pp. 353–356

12.27* Förster, H.J.: *Automatische Fahrzeuggetriebe* (Automatic Vehicle Transmissions) Berlin; Heidelberg; New York; Hong Kong; Barcelona; Budapest: Springer-Verlag, 1990

12.28* SKF S-Range: *Ritzel- und Differentiallager für Lastkraftwagen* (Pinion Bearings and Differential Bearings for Trucks)

12.29* Göhring, E.; Hopf, W.; Striftler, P.: *Eine neue Achsbaureihe für Nutzfahrzeuge mit Planetengetrieben in den Radnaben* (A New Range of Axles for Commercial Vehicles with Planetary Gears in the Wheel Hubs) ATZ Automobiltechnische Zeitschrift 75 (1973), pp. 308–314

12.30* Bussien: *Automobiltechnisches Handbuch* (Automobile Engineering Handbook) Vol. 2. Berlin: Technischer Verlag Herbert Cram, 1965

12.31* Müller, A.H.: *Neuentwicklung von Daimler-Benz-Nutzfahrzeugen* (New Development of Daimler-Benz Commercial Vehicles) ATZ Automobiltechnische Zeitschrift 69 (1967), pp. 311–318

12.32* Klanner, R.: *Die neue M.A.N.-Nutzfahrzeugreihe* (The New M.A.N. Commercial Vehicle Range) ATZ Automobiltechnische Zeitschrift 74 (1972), pp. 224–228

12.33* Kill, H.: *Die neue Leichtlastwagen-Baureihe von Magirus Deutz* (The New Magirus Deutz Light Truck Range) ATZ Automobiltechnische Zeitschrift 79 (1977), pp. 97–100

12.34* Automobile Engineer, Supplement: New Double Reduction Axle. October 1968, pp. 436

12.35* Leffler, H.: *ABS-integrierte Schlupfregelsysteme, Aufwandsabschätzung und Leistungsvergleich durch den Fahrzeughersteller* (ABS-Integral Slip Control Systems, Estimation of Resources Required and Performance Comparison by the Vehicle Manufacturers) ATZ Automobiltechnische Zeitschrift 93 (1991), No. 1, pp. 4–10

12.36* Egger, G.; Herb, E.; Krusche, H.; Wallentowitz, H.: *Stabilitäts- und Traktionsüberwachung bei allradgetriebenen Personenwagen* (Stability and Traction Monitoring in All-Wheel Drive Passenger Cars) ATZ Automobiltechnische Zeitschrift 91 (1989), No. 2, pp. 83–90

12.37* Behles, F.; Barske, H.: *Die Frontantriebs-Entwicklung bei Audi, NSU, Auto Union und DKW. Teil 1* (Front-Wheel Drive Development at Audi, NSU, Auto Union and DKW. Part 1) ATZ Automobiltechnische Zeitschrift 83 (1981), No. 10, pp. 487–495

12.38* Behles, F.; Barske, H.: *Die Frontantriebs-Entwicklung bei Audi, NSU, Auto Union und DKW. Teil 2* (Front-Wheel Drive Development at Audi, NSU, Auto Union and DKW. Part 2) ATZ Automobiltechnische Zeitschrift 83 (1981), No. 11, pp. 557–564

12.39* Heißling, B.; Rompe, K.; Bensinger, J.: *Untersuchung der Fahreigenschaften des vierradgetriebenen Audi quattro* (Investigation of the Handling Characteristics of the 4-Wheel Drive Audi Quattro) ATZ Automobiltechnische Zeitschrift 85 (1983), No. 11, pp. 681–691

12.40 Berger, R.; Fischer, R.; Salecker, M.: *Von der Automatisierten Kupplung zum Automatisierten Schaltgetriebe* (From the Automated Clutch to the Automated Manual Gearbox) VDI-Berichte (1998), No. 1393, pp. 159–174

12.41 Schaller, K.-V.: *Automatisiertes Getriebeschaltsystem SAMT B. Tagungsband, Symposium: Steuerungssysteme für den Antriebsstrang von Kraftfahrzeugen* (SAMT B Automatic Gearshift System. Conference Publication, Symposium on: Control Systems for Motor Vehicle Power Trains) 18.–19. September 1997, Berlin

12.42 v. Glasner, E.-C.; Povel, R.; Holloh, K.-D.: *Der intelligente Antriebsstrang. Tagungsband, Symposium: Steuerungssysteme für den Antriebsstrang von Kraftfahrzeugen* (The Intelligent Power Train. Conference Publication, Symposium on: Control Systems for Motor Vehicle Power Trains) 18.–19. September 1997, Berlin

12.43 Hirt, M.; Witzenberger, M.: *Modulationskupplung für schwere Nutzfahrzeuggetriebe* (Modulation Clutch for Heavy Commercial Vehicle Transmissions) VDI-Berichte (1995), No. 1175, pp. 407–417

12.44 Wörner, G.; Sperber, R.; Kosik, F.; Schröder, R.; Duckeck, F.: *Das neue Schaltgetriebe FSG 150. Die neue A-Klasse von Daimler-Benz* (The New FSG 150 Manual Gearbox. The New Daimler-Benz A-Class) Sonderausgabe ATZ und MTZ (10/1997), pp. 88–94

12.45 Göddel, T.; Hillenbrand, H.; Hopff, C.; Jud, M.: *Das neue Fünfgang-Automatikgetriebe W5A 180. Die neue A-Klasse von Daimler-Benz* (The New W5A 180 5-Speed Automatic Transmission. The New Daimler-Benz A-Class) Sonderausgabe ATZ und MTZ (10/1997), pp. 96–101

12.46 Rösch, R.; Kürschner, M.; Zaiser, W.: *Das automatische Getriebe W5A 330/580 für Personenwagen von Mercedes-Benz – Aufbau und Betriebsverhalten des mechanischen Getriebeteils* (The Mercedes-Benz W5A 330/580 Automatic Transmission for Passenger Cars – Construction and Operating Behaviour of the Mechanical Part of the Transmission) ATZ Automobiltechnische Zeitschrift 97 (1995), No. 9, pp. 536–545

12.47 Härdtle, W.: *Ein neues automatisiertes Schaltgetriebe für schwere Nutzfahrzeuge* (A New Automated Manual Gearbox for Heavy Commercial Vehicles) ATZ Automobiltechnische Zeitschrift 99 (1997), No. 10, pp. 598–604

Chapter 13 "Engine and Transmission Management, Electronics and Information Networking"

13.1 Schubert, K.: *Aspekte der zukünftigen Nutzfahrzeugtechnik unter besonderer Berücksichtigung der Elektronik* (Aspects of Future Commercial Vehicle Engineering with Special Regard to Electronics) Automobil-Industrie (1989), No. 2, pp. 157–165

13.2 Köpf, P.: *Getriebekonzepte für Straßenfahrzeuge – Entwicklungsstand und künftige Entwicklungen* (Transmission Concepts for Road Vehicles – State of Development and Future Developments) Aachen: 2. Aachener Kolloquium "Fahrzeug- und Motorentechnik", 1989, pp. 521–538

13.3 Westendorf, H.: *EKS – Das elektronische Kupplungssystem von Fichtel & Sachs* (The Fichtel & Sachs Electronic Clutch System) VDI-Berichte (1991), No. 878, pp. 117–135

13.4 Maier, U.; Petersmann, J.; Seidel, W.; Stohwasser, A.; Wehr, T.: *Porsche Tiptronic* (Porsche Tiptronic) ATZ Automobiltechnische Zeitschrift 92 (1990), No. 6, pp. 308–319

Chapter 14 "Overview of the Development Process, Product Planning and Systematic Engineering Design"

14.1 VDI-Richtlinie 2221: *Methodik zum Entwickeln und Konstruieren technischer Systeme und Produkte* (Methodology for Developing and Engineering Technical Systems and Products) Düsseldorf: VDI-Verlag, 1985

14.2 Brankamp, K.: *Planung und Entwicklung neuer Produkte* (Planning and Developing New Products) Berlin: de Gruyter & Co, 1971

14.3 Mellerowicz, K.: *Planung und Kostenrechnung* (Planning and Cost Accounting) Vol. I. Berlin: de Gruyter & Co, 1957

14.4 Koller, R.: *Konstruktionsmethode für den Maschinen-, Geräte- und Apparatebau* (Design Engineering Methods for Machine Building, Equipment Building and Apparatus Construction) Berlin; Heidelberg; New York: Springer-Verlag, 1979

14.5 Ewald, O.: *Lösungssammlungen für das Methodische Konstruieren* (Solution Compendiums for Systematic Design) Düsseldorf: VDI-Verlag GmbH, 1975

14.6 Hansen, F.: *Konstruktionssystematik* (Design Methodology Systematics) Berlin: VEB-Verlag Technik, 1966

14.7 Wolff, J.: *Kreatives Konstruieren* (Creative Engineering Design) Essen: Verlag W. Girardet, 1976

14.8 Conrad, P.; Schiemann, H.; Vömel, G.: *Erfolg durch Methodisches Konstruieren* (Success Through Systematic Design) Grafenau/Württ.: Lexika-Verlag, 1978

14.9 Steinwachs, H.: *Praktische Konstruktionsmethode* (Practical Design Engineering Methods) Würzburg: Vogel-Verlag, 1976

14.10 Hohmann, K.: *Methodisches Konstruieren* (Systematic Design) Essen: Verlag W. Girardet, 1977

14.11 Pahl, G.; Beitz, W.: *Konstruktionslehre* (Engineering Design Theory) Berlin; Heidelberg; New York: Springer-Verlag, 1977

14.12 Rodenacker, W.: *Methodisches Konstruieren* (Systematic Design) Berlin; Heidelberg; New York: Springer-Verlag, 1970

14.13 Hubka, V.: *Theorie der Konstruktionsprozesse* (Theory of Engineering Design Processes) Berlin; Heidelberg; New York: Springer-Verlag, 1976

14.14 Rodenacker, W.; Claussen, U.: *Regeln des Methodischen Konstruierens* (Rules of Systematic Design) Mainz: Krausskopf-Verlag GmbH, 1973

14.15 Roth, K.: *Aufbau und Handhabung von Konstruktionskatalogen* (Compiling and Managing Design Engineering Catalogues) Düsseldorf: VDI-Berichte (1974), No. 219

14.16 Adams, J.: Conceptual Blockbusting. New York; London: W.W. Norton & Company, 1976

14.17 Glegg, G.: The Design of Design. Cambridge: University Press, 1971

14.18 Beitz, W.: *Neue Arbeitstechniken beim Konstruieren* (New Techniques for Design Engineering) Düsseldorf: VDI-Berichte (1983), No. 492

14.19 VDI-Richtlinie 2222 (1): *Konstruktionsmethodik: Konzipieren technischer Produkte* (Design Methodology Systematics: Creating Technical Products) Düsseldorf: VDI-Verlag, 1977

14.20 Zwicky, F.: *Entdecken, Erfinden, Forschen im Morphologischen Weltbild* (Discovering, Inventing, Researching in the Morphological World Picture) München; Zürich: Verlag Droemer-Knaur, 1966/1971

Chapter 15 "Computer-Aided Transmission Development, Driving Simulation"

15.1 Dirschmid, W.: *Einsatz der Simulationstechnik unter dem Gesichtspunkt des Nutzen-Kosten-Aspektes* (Cost/Benefit Aspects of the Use of Simulation Techniques) ATZ Automobiltechnische Zeitschrift 94 (1992), No. 9, pp. 488–491

15.2 Ludmann, J.; Diekamp, R.; Lerner, G.: *PELOPS – Ein Programmsystem zur Untersuchung neuer Längsdynamikkonzepte im Verkehrsfluß* (PELOPS – A Program System for Investigating New Longitudinal Dynamics Concepts in Traffic Flow) VDI-Berichte (1992), No. 1007, pp. 192–209

15.3 Vollmer, D.; Balasubramanian, B.; Siegert, E.: *Fahrsimulation unter realistischen Umfeldbedingungen* (Driving Simulation Under Realistic Conditions) VDI-Berichte (1992), No. 1007, pp. 263–280

15.4 Wiedemann, R.; Schwerdtfeger, T.: *Makroskopisches Simulationsmodell für Schnellstraßennetze mit Berücksichtigung von Einzelfahrzeugen* (Macroscopic Simulation Model for Highway Networks Taking Into Account Individual Vehicles) Forschung Straßenbau und Straßenverkehrstechnik (1987), Heft 500

15.5 Schuh, R.; Baumgart, H.; Proschka, R.; Schieschke, R.: *Moderne Methoden der Systemerprobung am Beispiel der "Hardware in the Loop" Untersuchung von ASR-Systemen* (Modern Methods of System Testing Based on the Example of "Hardware in the Loop" Investigation of ASR Systems) VDI-Berichte (1992), No. 1007, pp. 747–773

15.6 Naunheimer, H.; Lechner, G.: *Fahrsimulation – Parameteruntersuchungen zur Optimierung des Antriebsstranges* (Driving Simulation – Parameter Investigations Into Optimising the Power Train) VDI-Berichte (1991), No. 878, pp. 467–483

15.7 Wolz, U.: *Die Einbindung der technischen Berechnung bei einem Fahrzeugzulieferer in den Entwicklungsprozeß* (Incorporating Engineering Calculation in the Development Process at a Vehicle Supplier) VDI-Berichte (1992), No. 1007, pp. 59–79

15.8 Naunheimer, H.; Zelßmann, H.; Lechner, G.: *Satellitengestützte Aufzeichnung von Streckendaten für Fahrsimulationsprogramme* (Satellite-Based Recording of Test Run Data for Driving Simulation Programs) ATZ Automobiltechnische Zeitschrift 95 (1993), No. 12, pp. 650–654

15.9 Jauch, F.: *Beitrag zur Optimierung des Antriebsstranges von Kraftfahrzeugen unter dem Aspekt von Kraftstoffverbrauch, Transportleistung und Lastkollektiven* (Conribution to Optimising the Power Train of Motor Vehicles in Terms of Fuel Consumption, Transportation Efficiency and Load Profiles) Stuttgart, Universität, Institut für Maschinenelemente, Dissertation, 1989

15.10 Butscher, K.; Kötzle, H.; Müller, K.: *Fahrzeug-Meßsystem zur Erfassung topographischer Straßendaten* (Vehicle Measuring System for Acquiring Topographical Road Data) VDI-Berichte (1988), No. 687, pp. 511–520

15.11 David, W.: *Neuartiges, modular aufgebautes Rechenprogramm zur Längsdynamiksimulation* (Innovative, Modular Computer Program for Longitudinal Dynamics Simulation) Aachen: 2. Aachener Kolloquium "Fahrzeug- und Motorentechnik", 1989, pp. 497–519

15.12 Rauh, J.: *Fahrdynamik-Simulation mit CASCaDE* (Simulation Vehicle Movement Dynamics with CASCaDE) VDI-Berichte (1990), No. 816, pp. 599–608

15.13 Willmerding, G.: *Simulationssytem zur Optimierung des Antriebsstrangs unter Einbeziehung des Verkehrsflusses* (Simulation System for Optimising the Power Train Taking Account of Traffic Flow) Part 1: ATZ Automobiltechnische Zeitschrift 94 (1992), No. 5, pp. 286–291; Part 2: ATZ 94 (1992), No. 6, pp. 342–349

15.14 Westhäuser, P.; Roddewig, H.: *Voith-Antriebsstrang-Optimierungs-Programm (VASOP)* (Voith Power Train Optimisation Program (VASOP)) ATZ Automobiltechnische Zeitschrift 94 (1992), No. 9, pp. 478–480

15.15 Spörl, T.: *Simulation von Hybridantriebssträngen mit dem Fahrsimulationsprogramm FASIMA II* (Simulation of Hybrid Power Trains with the Driving Simulation Program FASIMA II) Stuttgart, Universität, Institut für Maschinenelemente, Auftragssimulation, unveröffentlicht, März 1994

15.16 Keßler, B.: *Bewegungsgleichungen für Echtzeitanwendungen in der Fahrzeugdynamik* (Motion Equations for Real-Time Applications in Vehicle Dynamics) Stuttgart, Universität, Institut B für Mechanik, Dissertation, 1989

15.17 Stroustrup, B.: The C++ Programming Language. Addison-Wesley Publishing Company, 1987

15.18 Hummel, H.; Naunheimer, H.: *Optimierung des Kraftstoffverbrauchs durch rechnerische Fahrsimulation* (Optimising Fuel Consumption by Quantitative Driving Simulation) DER NAHVERKEHR (1991), No. 3, pp. 54–57

Chapter 16 "Reliability and Testing of Vehicle Transmissions"

16.1 Bertsche, B.; Lechner, G.: *Zuverlässigkeit im Maschinenbau: Ermittlung von Bauteil- und System-Zuverlässigkeiten* (Reliability in Machine Construction: Determining Reliability of Components and Systems) Berlin; Heidelberg; New York; London; Paris; Tokyo; Hong Kong; Barcelona: Springer-Verlag, 1990

16.2 VDI-Richtlinie 4001: *Grundbegriffe zum VDI-Handbuch Technische Zuverlässigkeit* (Basic Concepts for the VDI Technical Reliability Handbook) Page 2

16.3 Technische Akademie Esslingen – TAE-Lehrgang No. 1267/60.175, 1993
1/ Lechner, G.: *Zuverlässigkeit und Lebensdauer von Systemen* (Reliability and Service Life of Systems)
2/ Bertsche, B.: *Mathematische Behandlung der Zuverlässigkeit von Komponenten und Systemen* (Mathematical Treatment of Reliability of Components and Systems)
3/ Hirschmann, K.H.: *Methodische Lebensdauerberechnung bei Maschinenelementen* (Systematic Service Life Calculation for Machine Elements)
4/ v. Eiff, H.; 5/ Brodbeck, P.; 6/ Wu, Z.: *Vorgehensweise bei der Zuverlässigkeitsanalyse* (Procedure for Reliability Analysis)
7/ Herchenröder, P.: *Systemanalyse durch Monte-Carlo-Simulation* (Systems Analysis Using the Monte Carlo Simulation)
8/ Lechner, G.: *Zuverlässigkeitssicherung bei Systemen der Fahrzeugtechnik* (Reliability Assurance in Automotive Engineering Systems)

16.4 Bertsche, B.: *Zur Berechnung der System-Zuverlässigkeit von Maschinenbau-Produkten* (Calculating the System Reliability of Machine-Building Products) Stuttgart, Universität, Institut für Maschinenelemente, Dissertation, 1989

16.5 Bronstein, I.N.; Semendjajew K.A.: *Taschenbuch der Mathematik* (Pocket Book of Mathematics) 25th Edition. Thun und Frankfurt/Main: Verlag Harri Deutsch, 1991

16.6 Schäfer, E.: *Zuverlässigkeit, Verfügbarkeit und Sicherheit in der Elektronik* (Reliability, availability and safety in electronics) Würzburg: Vogelverlag, 1979

16.7 Koslow, B.A.; Uschakow, I.A.: *Handbuch zur Berechnung der Zuverlässigkeit für Ingenieure* (Handbook for Calculating Reliability for Engineers) München; Wien: Carl Hanser Verlag, 1979

16.8 Wu, Z.: *Vergleich und Entwicklung von Methoden zur Zuverlässigkeitsanalyse von Systemen* (Comparison and Development of Methods for Reliability Analysis of Systems) Stuttgart, Universität, Institut für Maschinenelemente, Dissertation, 1992

16.9 Dubi, A.: Stochastic Modeling of Realistic Systems. Ben-Gurion University of the Negev, 1990

16.10 Brodbeck, P.; v. Eiff, H.; Lechner, G.: *Zuverlässigkeitsanalyse von Getrieben* (Reliability Analysis of Gear Systems) VDI-Berichte (1991), No. 878, pp. 425–441

16.11 Kersten, G.: *FMEA-Fehlermöglichkeits- und -einflußanalyse in der industriellen Praxis* (FMEA Failure Mode and Effect Analysis in Industrial Practice) Verlag Moderne Industrie, 1987

16.12 DIN 25424: *Fehlerbaumanalyse* (Fault Tree Analysis) Berlin: Beuth-Verlag GmbH, 1981

16.13 Bertsche, B.; Lechner, G.: *Einfluß der Teileanzahl auf die Systemzuverlässigkeit* (The Effect of the Number of Parts on System Reliability) Antriebstechnik 26 (1987), No. 7, pp. 40–43

16.14 *Qualitätskontrolle in der Automobilindustrie – Zuverlässigkeitssicherung bei Automobilherstellern und Lieferanten* (Quality Control in the Automobile Industry – Reliability Assurance for Automobile Manufacturers and Suppliers) 2nd Edition. Frankfurt: Verband der Automobilindustrie e.V. (VDA), 1984

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