Bosch Professional Automotive Information

BOSCH
Invented for life

Konrad Reif *Ed.*

Fundamentals of Automotive and Engine Technology

Standard Drives · Hybrid Drives · Brakes · Safety Systems

Springer Vieweg

Bosch Professional Automotive Information

Bosch Professional Automotive Information is a definitive reference for automotive engineers. The series is compiled by one of the world´s largest automotive equipment suppliers. All topics are covered in a concise but descriptive way backed up by diagrams, graphs, photographs and tables enabling the reader to better comprehend the subject.
There is now greater detail on electronics and their application in the motor vehicle, including electrical energy management (EEM) and discusses the topic of intersystem networking within vehicle. The series will benefit automotive engineers and design engineers, automotive technicians in training and mechanics and technicians in garages.

Konrad Reif
Editor

Fundamentals of Automotive and Engine Technology

Standard Drives, Hybrid Drives, Brakes, Safety Systems

Springer Vieweg

Prof. Dr.-Ing. Konrad Reif
Duale Hochschule Baden-Württemberg
Friedrichshafen, Germany
reif@dhbw-ravensburg.de

ISBN 978-3-658-03971-4 ISBN 978-3-658-03972-1 (eBook)
DOI 10.1007/978-3-658-03972-1

Library of Congress Control Number: 2014942447

Springer Vieweg

Printed on acid-free paper

Springer is part of Springer Science+Business Media
www.springer.com

Foreword

Hybrid drives and the operation of hybrid vehicles are characteristic of contemporary automotive technology. Together with the electronic driver assistant systems, hybrid technology is of the greatest importance and both cannot be ignored by today's car drivers. This technical reference book provides the reader with a firsthand comprehensive description of significant components of automotive technology. All texts are complemented by numerous detailed illustrations.

Complex technology of modern motor vehicles and increasing functions need a reliable source of information to understand the components or systems. The rapid and secure access to these informations in the field of Automotive Electrics and Electronics provides the book in the series "Bosch Professional Automotive Information" which contains necessary fundamentals, data and explanations clearly, systematically, currently and application-oriented. The series is intended for automotive professionals in practice and study which need to understand issues in their area of work. It provides simultaneously the theoretical tools for understanding as well as the applications.

Contents

2 History of the automobile
2 Development history
4 Pioneers of automotive technology
6 Robert Bosch's life's work (1861–1942)

8 History of the diesel engine
9 Rudolf Diesel
10 Mixture formation in the first diesel engines
11 Use of the first vehicle diesel engines
14 Bosch diesel fuel injection

18 Areas of use for diesel engines
18 Suitability criteria
18 Applications
21 Engine characteristic data

22 Basic principles of the diesel engine
22 Method of operation
25 Torque and power output
26 Engine efficiency
29 Operating statuses
33 Operating conditions
35 Fuel-injection system
36 Combustion chambers

40 Basic principles of diesel fuel injection
40 Mixture distribution
42 Fuel-injection parameters
51 Nozzle and nozzle holder designs

52 Basics of the gasoline (SI) engine
52 Method of operation
56 Cylinder charge
60 Torque and power
62 Engine efficiency
64 Specific fuel consumption
66 Combustion knock

68 Inductive ignition system
68 Design
69 Function and method of operation
71 Ignition parameters
75 Voltage distribution
76 Ignition driver stage
77 Connecting devices and interference suppressors

78 Transmissions for Motor Vehicles
78 Transmission in the Drivetrain
80 Transmission Requirements
81 Manual Transmission
82 Automated Shift Transmission (AST)
86 Dual-Clutch Transmission (DCT)
88 Automatic Transmission (AT)
96 Continuously Variable Transmission (CVT)
102 Toroid Transmission

104 Motor-vehicle safety
104 Safety systems
106 Basics of vehicle operation

114 Basic principles of vehicle dynamics
114 Tires
117 Forces acting on a vehicle
124 Dynamics of linear motion
126 Dynamics of lateral motion
128 Definitions

130 Car braking systems
130 Overview
132 History of the brake
138 Classification of car braking systems
140 Components of a car braking system
141 Brake-circuit configuration

142 Vehicle electrical systems
142 Electrical energy supply in the passenger car
146 Electrical energy management
148 Two-battery vehicle electrical system
149 Vehicle electrical systems for commercial vehicles
152 Wiring harnesses
154 Plug-in connections

158 Overview of electrical and electronic systems in the vehicle
158 Overview

161 Control of gasoline engines

172 Control of Diesel engines

180 Lighting technology

194 Electronic stability program

202 Adaptive cruise control

210 Occupant-protection systems

218 Hybrid drives
218 Principle
219 Operating modes
221 Start/stop function
222 Degrees of hybridization
224 Drive configurations

231 Operation of hybrid vehicles
231 Hybrid control
232 Operating strategies for hybrid vehicles
234 Operating-point optimization
237 Design of the internalcombustion engine

240 Regenerative braking system
240 Strategies of regenerative braking

244 Workshop technology
244 Workshop business
248 Diagnostics in the workshop
250 Testing equipment
252 Brake testing
258 Fuel-injection pump test benches
260 Testing in-line fuel-injection pumps
264 Testing helix and portcontrolled distributor injection pumps
268 Nozzle tests

270 Index

Authors

History of the automobile
Dipl.-Ing. Karl-Heinz Dietsche
Dietrich Kuhlgatz

History of the diesel engine
Dipl.-Ing. Karl-Heinz Dietsche

Areas of use for diesel engines
Dipl.-Ing. Joachim Lackner,
Dr.-Ing. Herbert Schumacher,
Dipl.-Ing. (FH) Hermann Grieshaber.

Basic principles of the diesel engine
Dr.-Ing. Thorsten Raatz,
Dipl.-Ing. (FH) Hermann Grieshaber.

Basic principles of diesel fuel injection
Dipl.-Ing. Jens Olaf Stein,
Dipl.-Ing. (FH) Hermann Grieshaber.

Basics of the gasoline (SI) engine
Dr. rer. nat. Dirk Hofmann,
Dipl.-Ing. Bernhard Mencher,
Dipl.-Ing. Werner Häming,
Dipl.-Ing. Werner Hess.

Inductive ignition system
Dipl.-Ing. Walter Gollin.

Transmissions for Motor Vehicles
D. Grauman,
Dipl.-Ing. T. Laux,
Dipl.-Ing. T. Müller.

Motor-vehicle safety
Dipl.-Ing. Friedrich Kost.

Basic principles of vehicle dynamics
Dipl.-Ing. Friedrich Kost.

Car braking systems
Dipl.-Ing. Wulf Post.

Vehicle electrical systems
Dipl.-Ing. Clemens Schmucker,
Dipl.-Ing. (FH) Hartmut Wanner,
Dipl.-Ing. (FH) Wolfgang Kircher,
Dipl.-Ing. (FH) Werner Hofmeister,
Dipl.-Ing. Andreas Simmel.

Overview of electrical and electronic systems in the vehicle, Control of gasoline engines, Control of Diesel engines, Lighting technology, Electronic stability program (ESP), Adaptive cruise control (ACC), Occupant-protection systems
Dipl.-Ing. Bernhard Mencher,
Dipl.-Ing. (BA) Ferdinand Reiter,
Dipl.-Ing. Andreas Glaser,
Dipl.-Ing. Walter Gollin,
Dipl.-Ing. (FH) Klaus Lerchenmüller,
Dipl.-Ing. Felix Landhäußer,
Dipl.-Ing. Doris Boebel, Automotive Lighting Reutlingen GmbH,
Dr.-Ing. Michael Hamm, Automotive Lighting Reutlingen GmbH,
Dipl.-Ing. Tilman Spingler, Automotive Lighting Reutlingen GmbH,
Dr.-Ing. Frank Niewels,
Dipl.-Ing. Thomas Ehret,
Dr.-Ing. Gero Nenninger,
Prof. Dr.-Ing. Peter Knoll,
Dr. rer. nat. Alfred Kuttenberger.

Hybrid drives
Dipl.-Ing. Michael Bildstein,
Dr.-Ing. Karsten Mann,
Dr.-Ing. Boyke Richter.

Operation of hybrid vehicles
Regenerative braking system
Strategies of regenerative braking
Dipl.-Ing. Michael Bildstein,
Dr.-Ing. Karsten Mann,
Dr.-Ing. Boyke Richter.

Workshop technology
Dipl.-Wirtsch.-Ing. Stephan Sohnle,
Dipl.-Ing. Rainer Rehage,
Rainer Heinzmann,
Rolf Wörner,
Günter Mauderer,
Hans Binder.

and the editorial team in cooperation with the responsible in-house specialist departments of Robert Bosch GmbH.
Unless otherwise stated, the authors are all employees of Robert Bosch GmbH.

Basics

History of the automobile

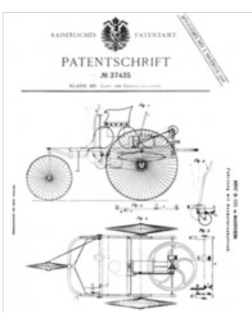
Mobility has always played a crucial role in the course of human development. In almost every era, man has attempted to find the means to allow him to transport people over long distances at the highest possible speed. It took the development of reliable internal-combustion engines that were operated on liquid fuels to turn the vision of a self-propelling "automobile" into reality (combination of Greek: autos = self and Latin: mobilis = mobile).

Development history

It would be hard to imagine life in our modern day without the motor car. Its emergence required the existence of many conditions without which an undertaking of this kind would not have been possible. At this point, some development landmarks may be worthy of note. They represent an essential contribution to the development of the automobile:

- About 3500 B.C.
  The development of the wheel is attributed to the Sumerians
- About 1300
  Further refinement of the carriage with elements such as steering, wheel suspension and carriage springs
- 1770
  Steam buggy by Joseph Cugnot
- 1801
  Étienne Lenoir develops the gas engine
- 1870
  Nikolaus Otto builds the first four-stroke internal-combustion engine

The first journey with an engine-powered vehicle is attributed to Joseph Cugnot (in 1770). His lumbering, steam-powered, wooden three-wheeled vehicle was able to travel for all of 12 minutes on a single tankful of water.



The patent issued to Benz on January 29, 1886 was not based on a converted carriage. Instead, it was a totally new, independent construction (Source: DaimlerChrysler Classic, Corporate Archives)

In 1885 Carl Benz enters the annals of history as the inventor of the first automobile. His patent marks the beginning of the rapid development of the automobile powered by the internal-combustion engine. Public opinion remained divided, however. While the proponents of the new age lauded the automobile as the epitome of progress, the majority of the population protested against the increasing annoyances of dust, noise, accident hazard, and inconsiderate motorists. Despite all of this, the progress of the automobile proved unstoppable.

Daimler Motorized Carriage, 1894 (Source: DaimlerChrysler Classic, Corporate Archives)

In the beginning, the acquisition of an automobile represented a serious challenge. A road network was virtually nonexistent; repair shops were unknown, fuel was purchased at the drugstore, and spare parts were produced on demand by the local blacksmith. The prevailing circumstances made the first long-distance journey by Bertha Benz in 1888 an even more astonishing accomplishment. She is thought to have been the first woman behind the wheel of a motorized vehicle. She also demonstrated the reliability of the automobile by journeying the then enormous distance of more than 100 kilometers (about 60 miles) between Mannheim and Pforzheim in south-western Germany.

In the early days, however, few entrepreneurs – with the exception of Benz – considered the significance of the engine-powered vehicle on a worldwide scale. It was the French who were to help the automobile to greatness. Panhard & Levassor used licenses for Daimler engines to build their own automobiles. Panhard pioneered construction features such as the steering wheel, inclined steering column, clutch pedal, pneumatic tires, and tube-type radiator.

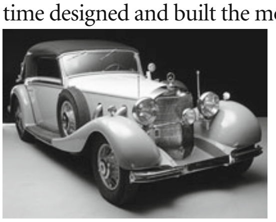
In the years that followed, the industry mushroomed with the arrival of companies such as Peugeot, Citroën, Renault, Fiat, Ford, Rolls-Royce, Austin, and others. The influence of Gottlieb Daimler, who was selling his engines almost all over the world, added significant impetus to these developments.

K. Reif (Ed.), *Fundamentals of Automotive and Engine Technology*,
DOI 10.1007/978-3-658-03972-1_1, © Springer Fachmedien Wiesbaden 2014

Taking their original design from coachbuilding, the motor cars of the time would soon evolve into the automobiles as we know them today. However, it should be noted that each automobile was an individual product of purely manual labor. A fundamental change came with the introduction of the assembly line by Henry Ford in 1913. With the Model T, he revolutionized the automobile industry in the United States. It was exactly at this juncture that the automobile ceased to be an article of luxury. By producing large numbers of automobiles, the price of an automobile dropped to such a level that it became accessible to the general public for the first time. Although Citroën and Opel were among the

first to bring the assembly line to Europe, it would gain acceptance only in the mid-1920s.

Automobile manufacturers were quick to realize that, in order to be successful in the marketplace, they had to accommodate the wishes of their customers. Automobile racing victories were exploited for commercial advertising. With ever-advancing speed records, professional race drivers left indelible impressions of themselves and the brand names of their automobiles in the minds of spectators. In addition, efforts were made to broaden the product line. As a result, the following decades produced a variety of automobile designs based on the prevailing zeitgeist, as well as the economic and political influences of the day. E.g., streamlined vehicles were unable to gain acceptance prior to WWII due to the demand for large and representative automobiles. Manufacturers of the time designed and built the most exclusive au-



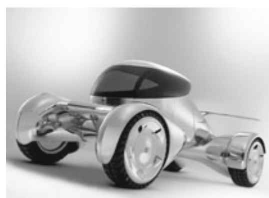
tomobiles, such as the Mercedes-Benz 500 K, Rolls-Royce Phantom III, Horch 855, or Bugatti Royale.

Mercedes-Benz 500 K Convertible C, 1934 (Source: DaimlerChrysler Classic, Corporate Archives)

More than 15 million units were produced of the Model T, affectionately called "Tin Lizzie". This record would be topped only by the Volkswagen Beetle in the 1970s

(Photos: Ford, Volkswagen AG)

WWII had a significant influence on the development of smaller cars. The Volkswagen model that came to be known as the "Beetle" was designed by Ferdinand Porsche and was manufactured in Wolfsburg. At the end of the war, the demand for cars that were small and affordable was prevalent. Responding to this demand, manufacturers produced automobiles such as the Goliath GP 700, Lloyd 300, Citroën 2CV, Trabant, Isetta, and the Fiat 500 C (Italian name: Topolino = little mouse). The manufacture of automobiles began to evolve new standards; there was greater emphasis on technology and integrated accessories, with a reasonable price/performance ratio as a major consideration.



Contemporary studies indicate what automobiles of tomorrow might look like (Photo: Peugeot)

Today, the emphasis is on a high level of occupant safety; the ever-rising traffic volumes and significantly higher speeds compared with yesteryear are making the airbag, ABS, TCS, ESP, and intelligent sensors virtually indispensable. The ongoing development of the automobile has been powered by innovative engineering on the part of the auto industry and by the constant rise in market demands. However, there are fields of endeavor that continue to present a challenge well into the future. One example is the further reduction of environmental burdens through the use of alternative energy sources (e.g., fuel cells).

One thing, however, is not expected to change in the near future – it is the one concept that has been associated with the automobile for more than a century, and which had inspired its original creators – it is the enduring ideal of individual mobility.

In 1899 the Belgian Camille Jenatzy was the first human to break the 100 km/h barrier. Today, the speed record stands at 1227.9 km/h.

Pioneers of automotive technology

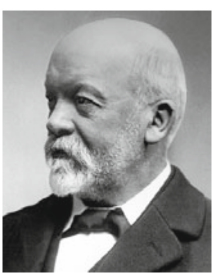
Owing to the large number of people who contributed to the development of the automobile, this list makes no claim to completeness

1866: Nikolaus August Otto (Photo: Deutz AG) acquires the patent for the atmospheric gas machine

Nikolaus August Otto (1832–1891), born in Holzhausen (Germany), developed an interest in technical matters at an early age. Beside his employment as a traveling salesman for food wholesalers, he was preoccupied with the functioning of gas-powered engines.

From 1862 onward he dedicated himself totally to engine construction. He managed to make improvements to the gas engine invented by the French engineer, Étienne Lenoir. For this work, Otto was awarded the gold medal at the 1867 Paris World Fair. Together with Daimler and Maybach, he developed an internal-combustion engine based on the four-stroke principle he had formulated in 1861. The resulting engine is known as the “Otto engine” to this day. In 1884 Otto invented magneto ignition, which allowed engines to be powered by gasoline. This innovation would form the basis for the main part of Robert Bosch’s life’s work.

Otto’s singular contribution was his ability to be the first to build the four-stroke internal-combustion engine and demonstrate its superiority over all its predecessors.



Gottlieb Daimler (Photo: DaimlerChrysler Classic, Corporate Archives)

Gottlieb Daimler (1834–1900) hailed from Schorndorf (Germany). He studied mechanical engineering at the Polytechnikum engineering college in Stuttgart. In 1865 he met the highly talented engineer Wilhelm Maybach. From that moment on, the two men would be joined in a lasting relationship of mutual cooperation. Besides inventing the first motorcycle, Daimler mainly worked on developing a gasoline engine suitable for use in road vehicles. In 1889 Daimler and Maybach introduced the first “steel-wheeled vehicle” in Paris featuring a two-cylinder V-engine. Scarcely one year later, Daimler was marketing his fast-running Daimler engine on an international scale. In 1891, for example, Armand Peugeot successfully entered a vehicle he had engineered himself in the Paris-Brest-Paris long-distance trial. It proved both the worth of his design and the dependability of the Daimler engine he was using.

Daimler’s merits lie in the systematic development of the gasoline engine and in the international distribution of his engines.

Wilhelm Maybach (Photo: MTU Friedrichshafen GmbH)

Wilhelm Maybach (1846–1929), a native of Heilbronn (Germany), completed his apprenticeship as a technical draftsman. Soon after, he worked as a design engineer. Among his employers was the firm of Gasmotoren Deutz AG (founded by Otto). He already earned the nickname of “king of engineers” during his own lifetime.

Maybach revised the gasoline engine and brought it to production. He also developed water cooling, the carburetor, and the dual-ignition system. In 1900 Maybach built a revolutionary, alloy-based racing car. This vehicle was developed in response to a suggestion by an Austrian businessman named Jellinek. His order for 36 of these cars was given on condition that the model was to be named after his daughter Mercedes.

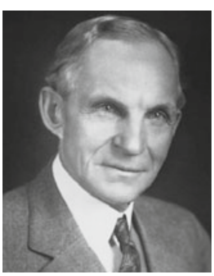
Maybach’s virtuosity as a design engineer pointed the way to the future of the contemporary automobile industry. His death signaled the end of the grand age of the automotive pioneers.

1886: As inventor of the first automobile fitted with an internal-combustion engine, Benz enters the annals of world history (Photo: DaimlerChrysler Classic, Corporate Archives)

Carl Friedrich Benz (1844–1929), born in Karlsruhe (Germany), studied mechanical engineering at the Polytechnikum engineering college in his hometown. In 1871 he founded his first company, a factory for iron-foundry products and industrial components in Mannheim.

Independently of Daimler and Maybach, he also pursued the means of fitting an engine in a vehicle. When the essential claims stemming from Otto's four-stroke engine patent had been declared null and void, Benz also developed a surface carburetor, electrical ignition, the clutch, water cooling, and a gearshift system, besides his own four-stroke engine. In 1886 he applied for his patent and presented his motor carriage to the public. In the period until the year 1900, Benz was able to offer more than 600 models for sale. In the period between 1894 and 1901 the factory of Benz & Co. produced the "Velo", which, with a total output of about 1200 units, may be called the first mass-produced automobile. In 1926 Benz merged with Daimler to form "Daimler-Benz AG".

Carl Benz introduced the first automobile and established the preconditions for the industrial manufacture of production vehicles.



Henry Ford (Photo: Ford)

Henry Ford (1863–1947) hailed from Dearborn, Michigan (USA). Although Ford had found secure employment as an engineer with the Edison Illuminating Company in 1891, his personal interests were dedicated to the advancement of the gasoline engine.

In 1893 the Duryea Brothers built the first American automobile. Ford managed to even the score in 1896 by introducing his own car, the "Quadricycle Runabout", which was to serve as the basis for numerous additional designs. In 1908 Ford introduced the legendary "Model T", which was mass-produced on assembly lines from 1913 onward. Beginning in 1921, with a 55-percent share in the country's industrial production, Ford dominated the domestic automobile market in the USA.

The name Henry Ford is synonymous with the motorization of the United States. It was his ideas that made the automobile accessible to a broad segment of the population.

Rudolf C. K. Diesel (Photo: Historical Archives of MAN AG)

Rudolf Christian Karl Diesel (1858–1913), born in Paris (France), decided to become an engineer at the age of 14. He graduated from the Polytechnikum engineering college in Munich with the best marks the institution had given in its entire existence.

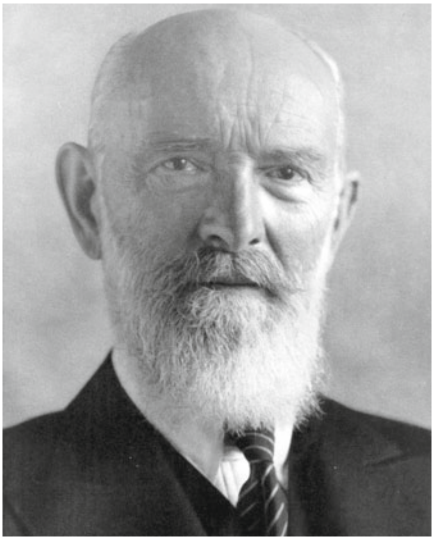
In 1892 Diesel was issued the patent for the "Diesel engine" that was later to bear his name. The engine was quickly adopted as a stationary power plant and marine engine. In 1908 the first commercial truck was powered by a diesel engine. However, its entrance into the world of passenger cars would take several decades. The diesel engine became the power plant for the serial-produced Mercedes 260 D as late as 1936. Today's diesel engine has reached a level of development such that it is now as common as the gasoline engine.

With his invention, Diesel has made a major contribution to a more economical utilization of the internal-combustion engine. Although Diesel became active internationally by granting production licenses, he failed to earn due recognition for his achievements during his lifetime.

Robert Bosch's life's work
(1861–1942)

"It has always been an unbearable thought to me that someone could inspect one of my products and find it inferior in any way. For that reason, I have constantly endeavored to make products that withstand the closest scrutiny – products that prove themselves superior in every respect."

Robert Bosch



Robert Bosch, born on September 23, 1861 in Albeck near Ulm (Germany), was the scion of a wealthy farmer's family. After completing his apprenticeship as a precision fitter, he worked temporarily for a number of enterprises, where he continued to hone his technical skills and expand his merchandising abilities and experience. After six months as an auditor studying electrical engineering at Stuttgart technical university, he traveled to the United States to work for "Edison Illuminating". He was later employed by "Siemens Brothers" in England.

(Photos: Bosch Archives)

In 1886 he decided to open a "Workshop for Precision Mechanics and Electrical Engineering" in the back of a dwelling in Stuttgart's west end. He employed another mechanic and an apprentice. At the beginning, his field of work lay in installing and repairing telephones, telegraphs, lightning


ROB. BOSCH
Rothebühlstr. 75 B.
Telephone, Haustelegraphen.
Fachmännische Prüfung und Anlegung von Blitzableitern. (37
Anlegung und Reparatur elektr. Apparate,
sowie aller Arbeiten der Feinmechanik.

First ad in the Stuttgart daily "Der Beobachter" (The Observer), 1887

conductors, and other light-engineering jobs. His dedication in finding rapid solutions to new problems also helped him gain a competitive lead in his later activities.

To the automobile industry, the low-voltage magneto ignition developed by Bosch in 1897 represented – much unlike its unreliable predecessors – a true breakthrough. This product was the launching board for the rapid expansion of Robert Bosch's business. He always managed to bring the purposefulness of the world of technology and economics into harmony with the needs of humanity. Bosch was a trailblazer in many aspects of social care.

Robert Bosch performed technological pioneering work in developing and bringing the following products to maturity:
- Low-voltage magneto ignition
- High-voltage magneto ignition for higher engine speeds (engineered by his colleague Gottlob Honold)
- Spark plug
- Ignition distributor
- Battery (passenger vehicles and motorcycles)
- Electrical starter
- Generator (alternator)
- Lighting system with first electric headlamp
- Diesel injection pumps
- Car radio (manufactured by "Ideal-Werke", renamed "Blaupunkt" in 1938)
- First lighting system for bicycles
- Bosch horn
- Battery ignition
- Bosch semaphore turn signal (initially ridiculed as being typical of German sense of organization – now the indispensable direction indicator)

At this point, many other achievements, also in the area of social engagement, would be worthy of note. They are clear indicators that Bosch was truly ahead of his time. His forward-thinking mind has given great impetus to advances in automobile development. The rising number of self-driving motorists fostered a corresponding increase in the need for repair facilities. In the 1920s Robert Bosch launched a campaign aimed

First offices in London's Store Street (Photo: Bosch Archives)

at creating a comprehensive service organization. In 1926, within Germany, these service repair centers were uniformly named "Bosch-Dienst" (Bosch Service) and the name was registered as a trademark.

Bosch had similarly high ambitions with regard to the implementation of social-care objectives. Having introduced the 8-hour day in 1906, he compensated his workers with ample wages. In 1910 he donated one million reichsmarks to support technical education. Bosch took the production of the 500,000th magneto as an occasion to introduce the work-free Saturday afternoon. Among other Bosch-induced improvements were old-age pensions, workplaces for the severely handicapped, and the vacation scheme. In 1913 the Bosch credo, "Occupation and the practice of apprenticeship are more knowledgeable educators than mere theory" resulted in the inauguration of an apprentice workshop that provided ample space for 104 apprentices.

In mid-1914 the name of Bosch was already represented around the world. But the era of great expansion between 1908 and 1940 would also bring the strictures of two world wars. Prior to 1914, 88 % of the products made in Stuttgart were slated for export. Bosch was able to continue expansion with the aid of large contingents destined for the military. However, in light of the atrocities of the war years, he disapproved of the resulting profits. As a result, he donated 13 million reichsmarks for social-care purposes.

After the end of WWI it was difficult to regain a foothold in foreign markets. In the United States, for example, Bosch factories, sales offices, and the corporate logo and symbol had been confiscated and sold to an American company. One of the consequences was that products appeared under the "Bosch" brand name that were not truly Bosch-made. It would take until the end of the 1920s before Bosch had reclaimed all of his former rights and was able to reestablish himself in the United States. The Founder's unyielding determination to overcome any and all obstacles returned the company to the markets of the world and, at the same time, imbued the minds of Bosch employees with the international significance of Bosch as an enterprise.

"To each his own automobile"
Such was the Bosch claim in a 1931 issue of the Bosch employee magazine "Bosch-Zünder" (Bosch Ignitor).

A closer look at two specific events may serve to underscore the social engagement of Robert Bosch. In 1936 he donated funds to construct a hospital that was officially opened in 1940. In his inaugural speech, Robert Bosch emphasized his personal dedication in terms of social engagement: "Every job is important, even the lowliest. Let no man delude himself that his work is more important than that of a colleague."

With the passing of Robert Bosch in 1942, the world mourned an entrepreneur who was a pioneer not only in the arena of technology and electrical engineering, but also in the realm of social engagement. Until this day, Robert Bosch stands as an example of progressive zeitgeist, of untiring diligence, of social improvements, of entrepreneurial spirit, and as a dedicated champion of education. His vision of progress culminated in the words, "Knowledge, ability, and will are important, but success only comes from their harmonious interaction."

In 1964 the Robert Bosch Foundation was inaugurated. Its activities include the promotion and support of health care, welfare, education, as well as sponsoring the arts and culture, humanities and social sciences. The Foundation continues to nurture the founder's ideals to this day.

History of the diesel engine

As early as 1863, the Frenchman Etienne Lenoir had test-driven a vehicle which was powered by a gas engine which he had developed. However, this drive plant proved to be unsuitable for installing in and driving vehicles. It was not until Nikolaus August Otto's four-stroke engine with magneto ignition that operation with liquid fuel and thereby mobile application were made possible. But the efficiency of these engines was low. Rudolf Diesel's achievement was to theoretically develop an engine with comparatively much higher efficiency and to pursue his idea through to readiness for series production.

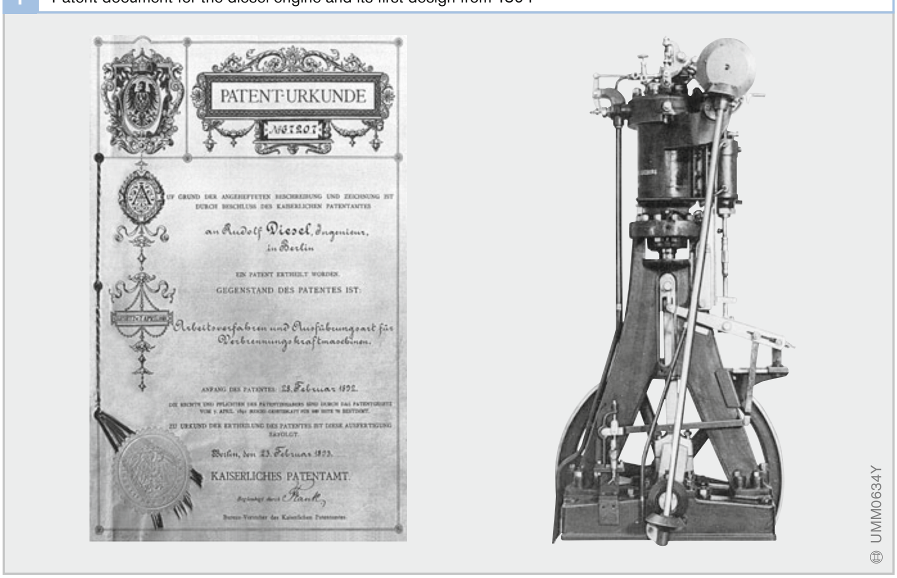
In 1897, in cooperation with Maschinenfabrik Augsburg-Nürnberg (MAN), Rudolf Diesel built the first working prototype of a combustion engine to be run on inexpensive heavy fuel oil. However, this first diesel engine weighed approximately 4.5 tonnes and was three meters high. For this reason, this engine was not yet considered for use in land vehicles.

However, with further improvements in fuel injection and mixture formation, Diesel's invention soon caught on and there were no longer any viable alternatives for marine and fixed-installation engines.

"It is my firm conviction that the automobile engine will come, and then I will consider my life's work complete." (Quotation by Rudolf Diesel shortly before his death)

2 Rudolf Diesel

1 Patent document for the diesel engine and its first design from 1894



K. Reif (Ed.), *Fundamentals of Automotive and Engine Technology*,
DOI 10.1007/978-3-658-03972-1_2, © Springer Fachmedien Wiesbaden 2014

Rudolf Diesel

Rudolf Diesel (1858–1913), born in Paris, decided at 14 that he wanted to become an engineer. He passed his final examinations at Munich Polytechnic with the best grades achieved up to that point.

Idea for a new engine

Diesel's idea was to design an engine with significantly greater efficiency than the steam engine, which was popular at the time. An engine based on the isothermal cycle should, according to the theory of the French physicist Sadi Carnot, be able to be operated with a high level of efficiency of over 90%.

Diesel developed his engine initially on paper, based on Carnot's models. His aim was to design a powerful engine with comparatively small dimensions. Diesel was absolutely convinced by the function and power of his engine.

Diesel's patent

Diesel completed his theoretical studies in 1890 and on 27 February 1892 applied to the Imperial Patent Office in Berlin for a patent on "New rational thermal engines". On 23 February 1893, he received patent document DRP 67207 entitled "Operating Process and Type of Construction for Combustion Engines", dated 28 February 1892.

This new engine initially only existed on paper. The accuracy of Diesel's calculations had been verified repeatedly, but the engine manufacturers remained skeptical about the engine's technical feasibility.

Realizing the engine

The companies experienced in engine building, such as Gasmotoren-Fabrik Deutz AG, shied away from the Diesel project. The required compression pressures of 250 bar were beyond what appeared to be technically feasible. In 1893, after many months of endeavor, Diesel finally succeeded in reaching an agreement to work with Maschinenfabrik Augsburg-Nürnberg (MAN). However, the agreement contained concessions by Diesel in respect of the ideal engine. The maximum pressure was reduced from 250 to 90 bar, and then later to 30 bar. This lowering of the pressure, required for mechanical reasons, naturally had a disadvantageous effect on combustibility. Diesel's initial plans to use coal dust as the fuel were rejected.

Finally, in Spring 1893, MAN began to build the first, uncooled test engine. Kerosene was initially envisaged as the fuel, but what came to be used was gasoline, because it was thought (erroneously) that this fuel would auto-ignite more easily. The principle of auto-ignition – i.e. injection of the fuel into the highly compressed and heated combustion air during compression – was confirmed in this engine.

In the second test engine, the fuel was not injected and atomized directly, but with the aid of compressed air. The engine was also provided with a water-cooling system.

It was not until the third test engine – a new design with a single-stage air pump for compressed-air injection – that the breakthrough made. On 17 February 1897, Professor Moritz Schröder of Munich Technical University carried out the acceptance tests. The test results confirmed what was then for a combustion engine a high level of efficiency of 26.2%.

Patent disputes and arguments with the Diesel consortium with regard to development strategy and failures took their toll, both mentally and physically, on the brilliant inventor. He is thought to have fallen overboard on a Channel crossing to England on 29 September 1913.

Mixture formation in the first diesel engines

Compressed-air injection

Rudolf Diesel did not have the opportunity to compress the fuel to the pressures required for spray dispersion, spray disintegration and droplet formation. The first diesel engine from 1897 therefore worked with compressed-air injection, whereby the fuel was introduced into the cylinder with the aid of compressed air. This process was later used by Daimler in its diesel engines for trucks.

The fuel injector had a port for the compressed-air feed (Fig. 1, 1) and a port for the fuel feed (2). A compressor generated the compressed air, which flowed into the valve. When the nozzle (3) was open, the air blasting into the combustion chamber also swept the fuel in and in this two-phase flow generated the fine droplets required for fast droplet vaporization and thus for auto-ignition.

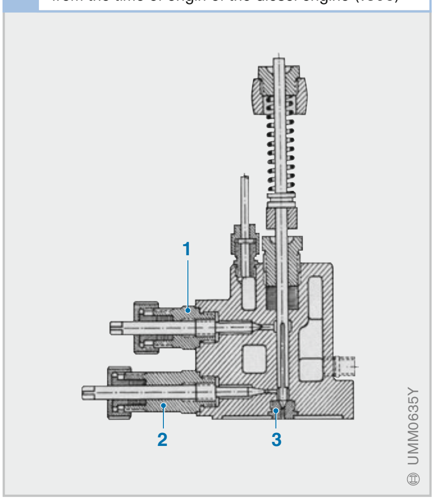
A cam ensured that the nozzle was actuated in synchronization with the crankshaft. The amount of fuel to be injected as controlled by the fuel pressure. Since the injection pressure was generated by the compressed air, a low fuel pressure was sufficient to ensure the efficacy of the process.

The problem with this process was – on account of the low pressure at the nozzle – the low penetration depth of the air/fuel mixture into the combustion chamber. This type of mixture formation was therefore not suitable for higher injected fuel quantities (higher engine loads) and engine speeds. The limited spray dispersion prevented the amount of air utilization required to increase power and, with increasing injected fuel quantity, resulted in local over-enrichment with a drastic increase in the levels of smoke. Furthermore, the vaporization time of the relatively large fuel droplets did not permit any significant increase in engine speed. Another disadvantage of this engine was the enormous amount of space taken up by the compressor. Nevertheless, this principle was used in trucks at that time.

Precombustion-chamber engine

The Benz diesel was a precombustion-chamber engine. Prosper L'Orange had already applied for a patent on this process in 1909. Thanks to the precombustion-chamber principle, it was possible to dispense with the complicated and expensive system of air injection. Mixture formation in the main com-

1 Fuel injector for compressed-air injection from the time of origin of the diesel engine (1895)



2 Principle of the precombustion-chamber engine

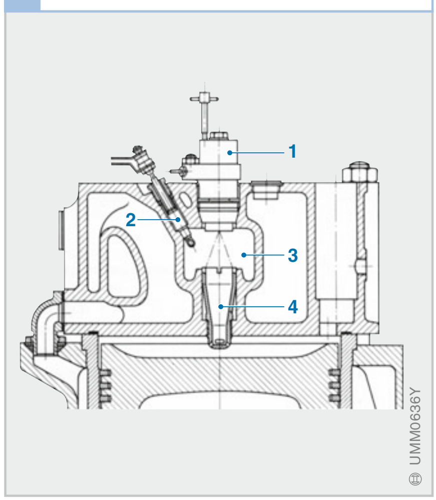


Fig. 1
1 Compressed-air feed
2 Fuel feed
3 Nozzle

Fig. 2
(Picture source: DaimlerChrysler)
1 Fuel valve
2 Glow filament for heating precombustion chamber
3 Precombustion chamber
4 Ignition insert

bustion chamber of this process, which is still used to this day, is ensured by partial combustion in the precombustion chamber. The precombustion-chamber engine has a specially shaped combustion chamber with a hemispherical head. The precombustion chamber and combustion chamber are interconnected by small bores. The volume of the precombustion chamber is roughly one fifth of the compression chamber.

The entire quantity of fuel is injected at approximately 230 to 250 bar into the precombustion chamber. Because of the limited amount of air in the precombustion chamber, only a small amount of the fuel is able to combust. As a result of the pressure increase in the precombustion chamber caused by the partial combustion, the unburned or partially cracked fuel is forced into the main combustion chamber, where it mixes with the air in the main combustion chamber, ignites and burns.

The function of the precombustion chamber here is to form the mixture. This process – also known as indirect injection – finally caught on and remained the predominant process until developments in fuel injection were able to deliver the injection pressures required to form the mixture in the main combustion chamber.

Direct injection

The first MAN diesel engine operated with direct injection, whereby the fuel was forced directly into the combustion chamber via a nozzle. This engine used as its fuel a very light oil, which was injected by a compressor into the combustion chamber. The compressor determined the huge dimensions of the engine.

In the commercial-vehicle sector, direct-injection engines resurfaced in the 1960s and gradually superseded precombustion-chamber engines. Passenger cars continued to use precombustion-chamber engines because of their lower combustion-noise levels until the 1990s, when they were swiftly superseded by direct-injection engines.

Use of the first vehicle diesel engines

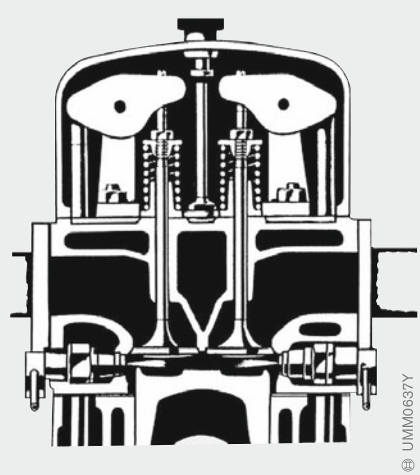
Diesel engines in commercial vehicles

Because of their high cylinder pressures, the first diesel engines were large and heavy and therefore wholly unsuitable for mobile applications in vehicles. It was not until the beginning of the 1920s that the first diesel engines were able to be deployed in commercial vehicles.

Uninterrupted by the First World War, Prosper L'Orange – a member of the executive board of Benz & Cie – continued his development work on the diesel engine. In 1923 the first diesel engines for road vehicles were installed in five-tonne trucks. These four-cylinder precombustion-chamber engines with a piston displacement of 8.8 *l* delivered 45...50 bhp. The first test drive of the Benz truck took place on 10 September with brown-coal tar oil serving as the fuel. Fuel consumption was 25% lower than benzene engines. Furthermore, operating fluids such as brown-coal tar oil cost much less than benzene, which was highly taxed.

The company Daimler was already involved in the development of the diesel engine prior to

3 First vehicle diesel with direct injection (MAN, 1924)



4 The most powerful diesel truck in the world from 1926 from MAN with 150 bhp (110 kW) for a payload of 10 t

the First World War. After the end of the war, the company was working on diesel engines for commercial vehicles. The first test drive was conducted on 23 August 1923 – at virtually the same time as the Benz truck. At the end of September 1923, a further test drive was conducted from the Daimler plant in Berlin to Stuttgart and back.

The first truck production models with diesel engines were exhibited at the Berlin Motor Show in 1924. Three manufacturers were represented, each with different systems, having driven development of the diesel forward with their own ideas:

- The Daimler diesel engine with compressed-air injection
- The Benz diesel with precombustion chamber
- The MAN diesel engine with direct injection

Diesel engines became increasingly powerful with time. The first types were four-cylinder units with a power output of 40 bhp. By 1928, engine power-output figures of more than 60 bhp were no longer unusual. Finally, even more powerful engines with six and eight cylinders were being produced for heavy commercial vehicles. By 1932, the power range stretched up to 140 bhp.

The diesel engine's breakthrough came in 1932 with a range of trucks offered by the company Daimler-Benz, which came into being in 1926 with the merger of the automobile manufacturers Daimler and Benz. This range was led by the Lo2000 model with a payload of 2 t and a permissible total weight of almost 5 t. It housed the OM59 four-cylinder engine with a displacement of 3.8 *l* and 55 bhp. The range extended up to the L5000 (payload 5 t, permissible total weight 10.8 t). All the vehicles were also available with gasoline engines of identical power output, but these engines proved unsuccessful when up against the economical diesel engines.

To this day, the diesel engine has maintained its dominant position in the commercial-vehicle sector on account of its economic efficiency. Virtually all heavy goods vehicles are driven by diesel engines. In Japan, large-displacement conventionally aspirated engines are used almost exclusively. In the USA and Europe, however, turbocharged engines with charge-air cooling are favored.

Diesel engines in passenger cars

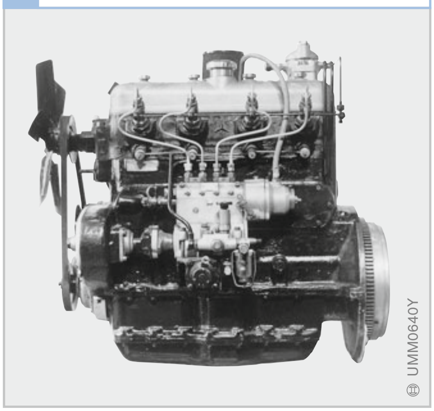
A few more years were to pass before the diesel engine made its debut in a passenger car. 1936 was the year, when the Mercedes 260D appeared with a four-cylinder diesel engine and a power output of 45 bhp.

The diesel engine as the power plant for passenger cars was long relegated to a fringe existence. It was too sluggish when compared with the gasoline engine. Its image was to change only in the 1990s. With exhaust-gas turbocharging and new high-pressure fuel-injection systems, the diesel engine is now on an equal footing with its gasoline counterpart. Power output and environmental performance are comparable. Because the diesel engine, unlike its gasoline counterpart, does not knock, it can also be turbocharged in the lower speed range, which results in high torque and very good driving performance. Another advantage of the diesel engine is, naturally, its excellent efficiency. This has led to it becoming increasingly accepted among car drivers – in Europe, roughly every second newly registered car is a diesel.

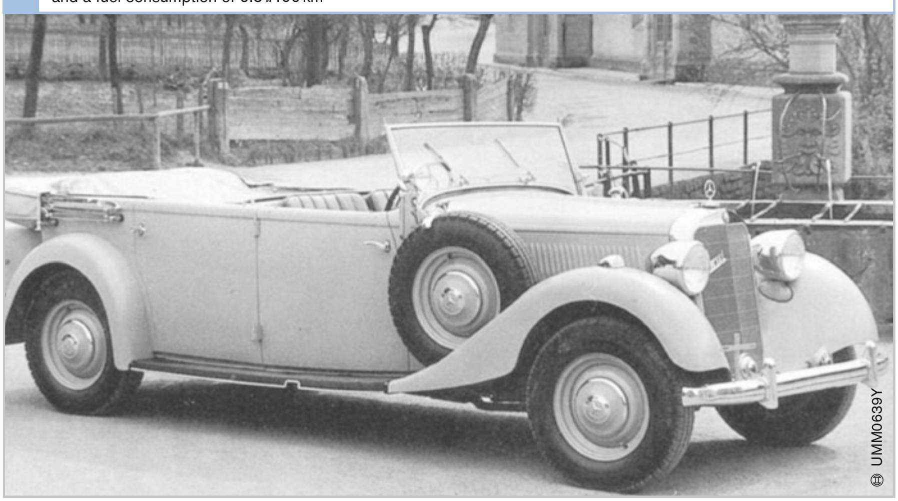
Further areas of application

When the era of steam and sailing ships crossing the oceans came to an end at the beginning of the 20th century, the diesel engine also emerged as the drive source for this mode of transport. The first ship to be fitted with a 25-bhp diesel engine was launched in 1903. The first locomotive to be driven by a diesel engine started service in 1913. The engine power output in this case was 1,000 bhp. Even the pioneers of aviation showed interest in the diesel engine. Diesel engines provided the propulsion on board the Graf Zeppelin airship.

6 Bosch in-line fuel-injection pump on the engine of the Mercedes 260D

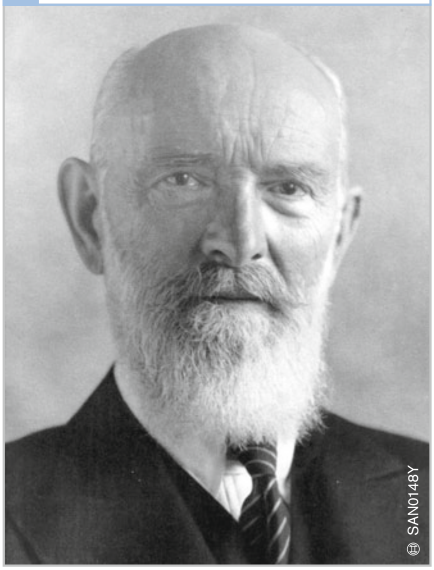


5 First diesel car: Mercedes-Benz 260D from 1936 with an engine power output of 45 bhp (33 kW) and a fuel consumption of 9.5 l/100 km



Bosch diesel fuel injection

1 Robert Bosch



Bosch's emergence onto the diesel-technology stage

In 1886, Robert Bosch (1861–1942) opened a "workshop for light and electrical engineering" in Stuttgart. He employed one other mechanic and an apprentice. At the beginning, his field of work lay in installing and repairing telephones, telegraphs, lightning conductors, and other light-engineering jobs.

The low-voltage magneto-ignition system developed by Bosch had provided reliable ignition in gasoline engines since 1897. This product was the launching board for the rapid expansion of Robert Bosch's business. The high-voltage magneto ignition system with spark plug followed in 1902. The armature of this ignition system is still to this day incorporated in the logo of Robert Bosch GmbH.

In 1922, Robert Bosch turned his attention to the diesel engine. He believed that certain accessory parts for these engines could similarly make suitable objects for Bosch high-volume precision production like magnetos and spark plugs. The accessory parts in question for diesel engines were fuel-injection pumps and nozzles.

Even Rudolf Diesel had wanted to inject the fuel directly, but was unable to do this because the fuel-injection pumps and nozzles needed to achieve this were not available. These pumps, in contrast to the fuel pumps used in compressed-air injection, had to be suitable for back-pressure reactions of up to several hundred atmospheres. The nozzles had to have quite fine outlet openings because now the task fell upon the pump and the nozzle alone to meter and atomize the fuel.

The injection pumps which Bosch wanted to develop should match not only the requirements of all the heavy-oil low-power engines with direct fuel injection which existed at the time but also future motor-vehicle diesel engines. On 28 December 1922, the decision was taken to embark on this development.

Demands on the fuel-injection pumps

The fuel-injection pump to be developed should be capable of injecting even small amounts of fuel with only quite small differences in the individual pump elements. This would facilitate smoother and more uniform engine operation even at low idle speeds. For full-load requirements, the delivery quantity would have to be increased by a factor of four or five. The required injection pressures were at that time already over 100 bar. Bosch demanded that these pump properties be guaranteed over 2,000 operating hours.

These were exacting demands for the then state-of-the-art technology. Not only did some feats of fluid engineering have to be achieved, but also this requirement represented a challenge in terms of production engineering and materials application technology.

Development of the fuel-injection pump
Firstly, different pump designs were tried out. Some pumps were spool-controlled, while others were valve-controlled. The injected fuel quantity was regulated by altering the plunger lift. By the end of 1924, a pump design was available which, in terms of its delivery rate, its durability and its low space requirement, satisfied the demands both of the Benz precombustion-chamber engine presented at the Berlin Motor Show and of the MAN direct-injection engine.

In March 1925, Bosch concluded contracts with Acro AG to utilize the Acro patents on a diesel-engine system with air chamber and the associated injection pump and nozzle. The Acro pump, developed by Franz Lang in Munich, was a unique fuel-injection pump. It had a special valve spool with helix, which was rotated to regulate the delivery quantity. Lang later moved this helix to the pump plunger.

The delivery properties of the Acro injection pump did not match what Bosch's own test pumps had offered. However, with the Acro engine, Bosch wanted to come into contact with a diesel engine which was particularly suitable for small cylinder units and high speeds and in this way gain a firm foothold for developing injection pumps and nozzles. At the same time, Bosch was led by the idea of granting licenses in the Acro patents to engine factories to promote the spread of the vehicle diesel engine and thereby contribute to the motorization of traffic.

After Lang's departure from the company in October 1926, the focus of activity at Bosch was again directed toward pump development. The first Bosch diesel fuel-injection pump ready for series production appeared soon afterwards.

2 Design of a Bosch fuel-injection pump from 1923/1924

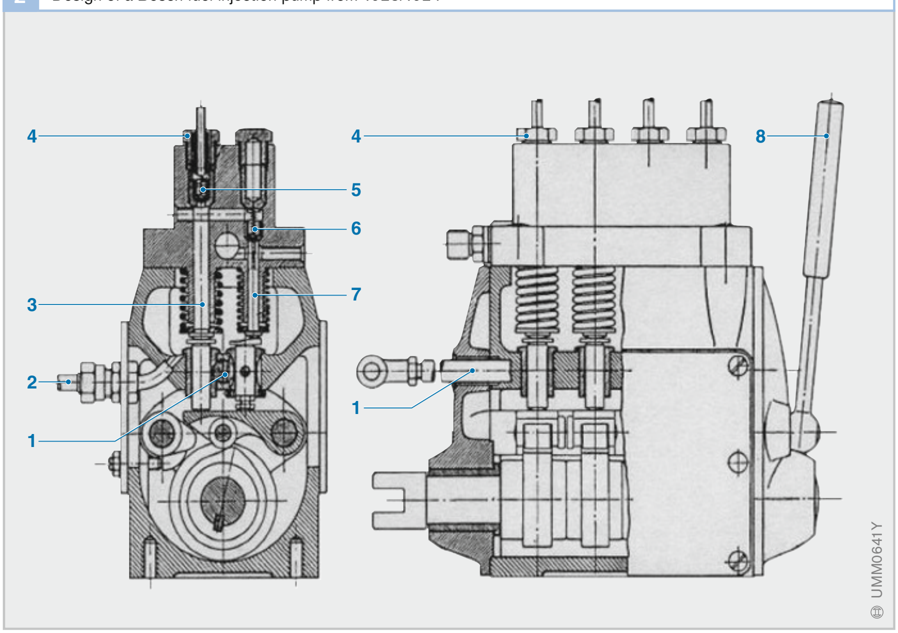


Fig. 2
1 Control rack
2 Inlet port
3 Pump plunger
4 Pressure-line port
5 Delivery valve
6 Suction valve
7 Valve tappet
8 Shutdown and pumping lever

Bosch diesel fuel-injection pump ready for series production

In accordance with the design engineer's plans of 1925 and like the modified Acro pump, the Bosch fuel-injection pump featured a diagonal helix on the pump plunger. Apart from this, however, it differed significantly from all its predecessors.

The external lever apparatus of the Acro pump for rotating the pump plunger was replaced by the toothed control rack, which engaged in pinions on control sleeves of the pump elements.

In order to relieve the load on the pressure line at the end of the injection process and to prevent fuel dribble, the delivery valve was provided with a suction plunger adjusted to fit in the valve guide. In contrast to the load-relieving techniques previously used, this new approach achieved increased steadiness of delivery at different speeds and quantity settings and significantly simplified and shortened the adjustment of multicylinder pumps to identical delivery by all elements.

The pump's simple and clear design made it easier to assemble and test. It also significantly simplified the replacement of parts compared with earlier designs. The housing conformed first and foremost to the requirements of the foundry and other manufacturing processes. The first specimens of this Bosch fuel-injection pump really suitable for volume production were manufactured in April 1927. Release for production in greater batch quantities and in versions for two-, four- and six-cylinder engines was granted on 30 November 1927 after the specimens had passed stringent tests at Bosch and in practical operation with flying colors.

3 First series-production diesel fuel-injection pump from Bosch (1927)

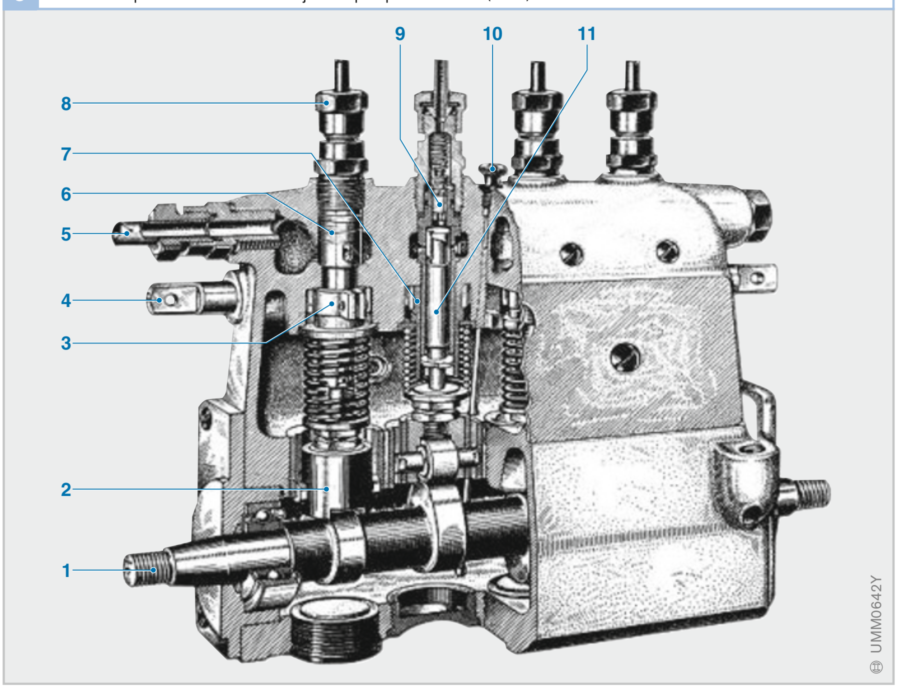


Fig. 3
1 Camshaft
2 Roller tappet
3 Control-sleeve gear
4 Control rack
5 Inlet port
6 Pump cylinder
7 Control sleeve
8 Pressure-line port
9 Delivery valve with plunger
10 Oil level
11 Pump plunger

Nozzles and nozzle holders

The development of nozzles and nozzle holders was conducted in parallel to pump development. Initially, pintle nozzles were used for precombustion-chamber engines. Hole-type nozzles were added at the start of 1929 with the introduction of the Bosch pump in the direct-injection diesel engine.

The nozzles and nozzle holders were always adapted in terms of their size to the new pump sizes. It was not long before the engine manufacturers also wanted a nozzle holder and nozzle which could be screwed into the cylinder head in the same way as the spark plug on a gasoline engine. Bosch adapted to this request and started to produce screw-in nozzle holders.

Governor for the fuel-injection pump

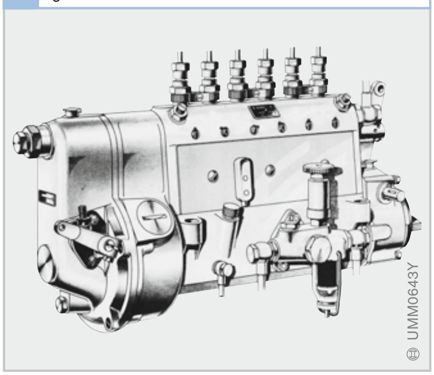
Because a diesel engine is not self-governing like a gasoline engine, but needs a governor to maintain a specific speed and to provide protection against overspeed accompanied by self-destruction, vehicle diesel engines had to be equipped with such devices right from the start. The engine factories initially manufactured these governors themselves. However, the request soon came to dispense with the drive for the governor, which was without exception a mechanical governor, and to combine it with the injection pump. Bosch complied with this request in 1931 with the introduction of the Bosch governor.

Spread of Bosch diesel fuel-injection technology

By August 1928, one thousand Bosch fuel-injection pumps had already been supplied. When the upturn in the fortunes of the vehicle diesel engine began, Bosch was well prepared and fully able to serve the engine factories with a full range of fuel-injection equipment. When the Bosch pumps and nozzles proved their worth, most companies saw no further need to continue manufacturing their own accessories in this field.

Bosch's expertise in light engineering (e.g., in the manufacture of lubricating pumps) stood it in good stead in the development of diesel fuel-injection pumps. Its products could not be manufactured "in accordance with the pure principles of mechanical engineering". This helped Bosch to obtain a market advantage. Bosch had thus made a significant contribution towards enabling the diesel engine to develop into what it is today.

4 Bosch fuel-injection pump with mounted mechanical governor



5 Billboard advertisement for Bosch diesel fuel injection

Areas of use for diesel engines

No other internal-combustion engine is as widely used as the diesel engine[1]. This is due primarily to its high degree of efficiency and the resulting fuel economy.

The chief areas of use for diesel engines are
- Fixed-installation engines
- Cars and light commercial vehicles
- Heavy goods vehicles
- Construction and agricultural machinery
- Railway locomotives and
- Ships

Diesel engines are produced as inline or V-configuration units. They are ideally suited to turbocharger or supercharger aspiration as – unlike the gasoline engine – they are not susceptible to knocking (refer to the chapter “Cylinder-charge control systems”).

[1]) Named after Rudolf Diesel (1858 to 1913) who first applied for a patent for his “New rational thermal engines” in 1892. A lot more development work was required, however, before the first functional diesel engine was produced at MAN in Augsburg in 1897.

Suitability criteria

The following features and characteristics are significant for diesel-engine applications (examples):
- Engine power
- Specific power output
- Operational safety
- Production costs
- Economy of operation
- Reliability
- Environmental compatibility
- User-friendliness
- Convenience (e.g. engine-compartment design)

The relative importance of these characteristics affect engine design and vary according to the type of application.

Applications

Fixed-installation engines

Fixed-installation engines (e.g. for driving power generators) are often run at a fixed speed. Consequently, the engine and fuel-injection system can be optimized specifically

1 Car diesel engine with unit injector system (example)

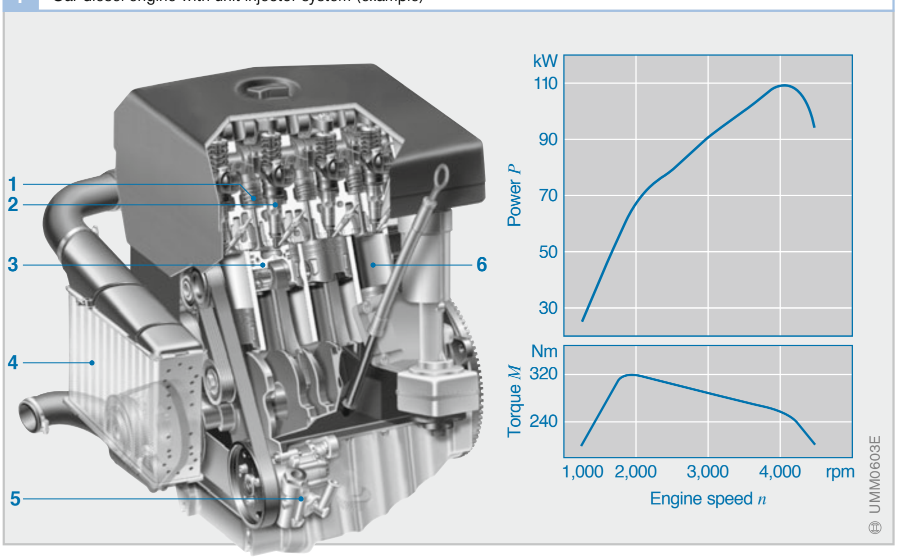


Fig. 1
1 Valve gear
2 Injector
3 Piston with gudgeon pin and conrod
4 Intercooler
5 Coolant pump
6 Cylinder

K. Reif (Ed.), *Fundamentals of Automotive and Engine Technology*,
DOI 10.1007/978-3-658-03972-1_3, © Springer Fachmedien Wiesbaden 2014

for operation at that speed. An engine governor adjusts the quantity of fuel injected dependent on engine load. For this type of application, mechanically governed fuel-injection systems are still used.

Car and commercial-vehicle engines can also be used as fixed-installation engines. However, the engine-control system may have to be modified to suit the different conditions.

Cars and light commercial vehicles

Car engines (Fig. 1) in particular are expected to produce high torque and run smoothly. Great progress has been made in these areas by refinements in engine design and the development of new fuel-injection with Electronic Diesel Control (EDC). These advances have paved the way for substantial improvements in the power output and torque characteristics of diesel engines since the early 1990s. And as a result, the diesel engine has forced its way into the executive and luxury-car markets.

Cars use fast-running diesel engines capable of speeds up to 5,500 rpm. The range of sizes extends from 10-cylinder 5-liter units used in large saloons to 3-cylinder 800-cc models for small subcompacts.

In Europe, all new diesel engines are now Direct-Injection (DI) designs as they offer fuel consumption reductions of 15 to 20% in comparison with indirect-injection engines. Such engines, now almost exclusively fitted with turbochargers, offer considerably better torque characteristics than comparable gasoline engines. The maximum torque available to a vehicle is generally determined not by the engine but by the power-transmission system.

The ever more stringent emission limits imposed and continually increasing power demands require fuel-injection systems with extremely high injection pressures. Improving emission characteristics will continue to be a major challenge for diesel-engine developers in the future. Consequently, further innovations can be expected in the area of exhaust-gas treatment in years to come.

2 Commercial-vehicle diesel engine with common-rail fuel-injection system (example)

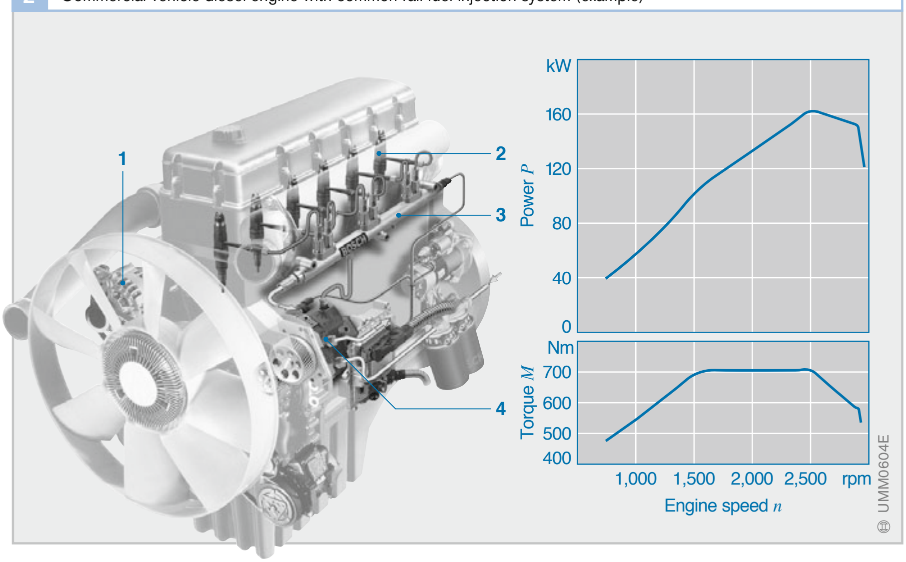


Fig. 2
1 Alternator
2 Injector
3 Fuel rail
4 High-pressure pump

Heavy goods vehicles

The prime requirement for engines for heavy goods vehicles (Fig. 2) is economy. That is why diesel engines for this type of application are exclusively direct-injection (DI) designs. They are generally medium-fast engines that run at speeds of up to 3,500 rpm.

For large commercial vehicles too, the emission limits are continually being lowered. That means exacting demands on the fuel-injection system used and a need to develop new emission-control systems.

Construction and agricultural machinery

Construction and agricultural machinery is the traditional domain of the diesel engine. The design of engines for such applications places particular emphasis not only on economy but also on durability, reliability and ease of maintenance. Maximizing power utilization and minimizing noise output are less important considerations than they would be for car engines, for example.
For this type of use, power outputs can range from around 3 kW to the equivalent of HGV engines.

Many engines used in construction-industry and agricultural machines still have mechanically governed fuel-injection systems. In contrast with all other areas of application, where water-cooled engines are the norm, the ruggedness and simplicity of the air-cooled engine remain important factors in the building and farming industries.

Railway locomotives

Locomotive engines, like heavy-duty marine diesel engines, are designed primarily with continuous-duty considerations in mind. In addition, they often have to cope with poorer quality diesel fuel. In terms of size, they range from the equivalent of a large truck engine to that of a medium-sized marine engine.

Ships

The demands placed on marine engines vary considerably according to the particular type of application. There are out-and-out high-performance engines for fast naval vessels or speedboats, for example. These tend to be 4-stroke medium-fast engines that run at speeds of 400...1,500 rpm and have up to 24 cylinders (Fig. 3). At the other end of

3 Marine diesel engine with single-plunger fuel-injection pumps (example)

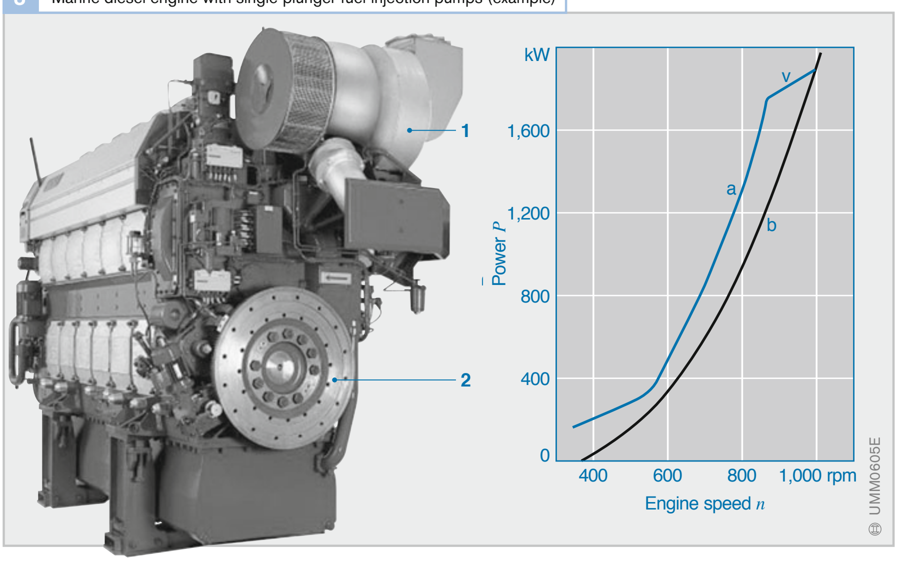


Fig. 3
1 Turbocharger
2 Flywheel

a Engine power output
b Running-resistance curve
v Full-load limitation zone

the scale there are 2-stroke heavy-duty engines designed for maximum economy in continuous duty. Such slow-running engines (< 300 rpm) achieve effective levels of efficiency of up to 55%, which represent the highest attainable with piston engines.

Large-scale engines are generally run on cheap heavy oil. This requires pretreatment of the fuel on board. Depending on quality, it has to be heated to temperatures as high as 160°C. Only then is its viscosity reduced to a level at which it can be filtered and pumped.

Smaller vessels often use engines originally intended for large commercial vehicles. In that way, an economical propulsion unit with low development costs can be produced. Once again, however, the engine management system has to be adapted to the different service profile.

Multi-fuel engines

For specialized applications (such as operation in regions with undeveloped infrastructures or for military use), diesel engines capable of running on a variety of different fuels including diesel, gasoline and others have been developed. At present they are of virtually no significance whatsoever within the overall picture, as they are incapable of meeting the current demands in respect of emissions and performance characteristics.

Engine characteristic data

Table 1 shows the most important comparison data for various types of diesel and gasoline engine.

The average pressure in petrol engines with direct fuel injection is around 10% higher than for the engines listed in the table with inlet-manifold injection. At the same time, the specific fuel consumption is up to 25% lower. The compression ratio of such engines can be as much as 13:1.

1 Comparison of diesel and gasoline engines

Fuel-injection system	Rated speed n_{rated} [rpm]	Compression ratio ε	Mean pressure [1] p_e [bar]	Specific power output $p_{e,\,spec.}$ [kW/l]	Power-to-weight ratio $m_{spec.}$ [kg/kW]	Specific fuel consumption [2] b_e [g/kWh]
Diesel engines						
IDI[3] conventionally aspirated car engines	3,500...5,000	20...24:1	7...9	20...35	1:5...3	320...240
IDI[3] turbocharged car engines	3,500...4,500	20...24:1	9...12	30...45	1:4...2	290...240
DI[4] conventionally aspirated car engines	3,500...4,200	19...21:1	7...9	20...35	1:5...3	240...220
DI[4] turbocharged car engines with i/clr[5]	3,600...4,400	16...20	8...22	30...60	4...2	210...195
DI[4] convent. aspirated comm. veh. engines	2,000...3,500	16...18:1	7...10	10...18	1:9...4	260...210
DI[4] turbocharged comm. veh. engines	2,000...3,200	15...18:1	15...20	15...25	1:8...3	230...205
DI[4] turboch. comm. veh. engines with i/clr[5]	1,800...2,600	16...18	15...25	25...35	5...2	225...190
Construct. and agricultural machine engines	1,000...3,600	16...20:1	7...23	6...28	1:10...1	280...190
Locomotive engines	750...1,000	12...15:1	17...23	20...23	1:10...5	210...200
Marine engines (4-stroke)	400...1,500	13...17:1	18...26	10...26	1:16...13	210...190
Marine engines (2-stroke)	50...250	6...8:1	14...18	3...8	1:32...16	180...160
Gasoline engines						
Conventionally aspirated car engines	4,500...7,500	10...11:1	12...15	50...75	1:2...1	350...250
Turbocharged car engines	5,000...7,000	7...9:1	11...15	85...105	1:2...1	380...250
Comm. veh. engines	2,500...5,000	7...9:1	8...10	20...30	1:6...3	380...270

Table 1

1) The average pressure p_e can be used to calculate the specific torque $M_{spec.}$ [Nm]:

$$M_{spec.} = \frac{25}{\pi \cdot p_e}$$

2) Best consumption
3) Indirect Injection
4) Direct Injection
5) Intercooler

Basic principles of the diesel engine

The diesel engine is a compression-ignition engine in which the fuel and air are mixed inside the engine. The air required for combustion is highly compressed inside the combustion chamber. This generates high temperatures which are sufficient for the diesel fuel to spontaneously ignite when it is injected into the cylinder. The diesel engine thus uses heat to release the chemical energy contained within the diesel fuel and convert it into mechanical force.

The diesel engine is the internal-combustion engine that offers the greatest overall efficiency (more than 50% in the case of large, slow-running types). The associated low fuel consumption, its low-emission exhaust and quieter running characteristics assisted, for example, by pre-injection have combined to give the diesel engine its present significance.

Diesel engines are particularly suited to aspiration by means of a turbocharger or supercharger. This not only improves the engine's power yield and efficiency, it also reduces pollutant emissions and combustion noise.

In order to reduce NO_x emissions on cars and commercial vehicles, a proportion of the exhaust gas is fed back into the engine's intake manifold (exhaust-gas recirculation). An even greater reduction of NO_x emissions can be achieved by cooling the recirculated exhaust gas.

Diesel engines may operate either as two-stroke or four-stroke engines. The types used in motor vehicles are generally four-stroke designs.

Method of operation

A diesel engine contains one or more cylinders. Driven by the combustion of the air/fuel mixture, the piston (Fig. 1, 3) in each cylinder (5) performs up-and-down movements. This method of operation is why it was named the "reciprocating-piston engine".

The connecting rod, or conrod (11), converts the linear reciprocating action of the piston into rotational movement on the part of the crankshaft (14). A flywheel (15) connected to the end of the crankshaft helps to maintain continuous crankshaft rotation and reduce unevenness of rotation caused by the periodic nature of fuel combustion in the individual cylinders. The speed of rotation of the crankshaft is also referred to as engine speed.

1 Four-cylinder diesel engine without auxiliary units (schematic)

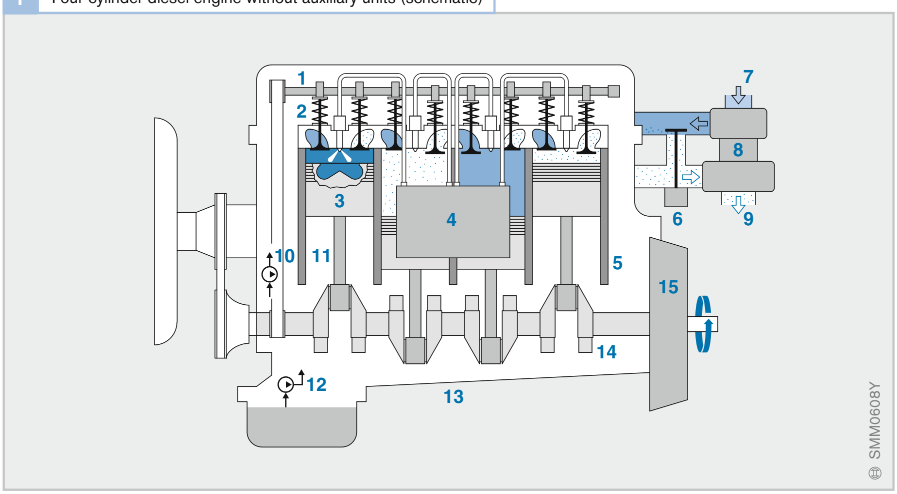


Fig. 1
1 Camshaft
2 Valves
3 Piston
4 Fuel-injection system
5 Cylinder
6 Exhaust-gas recirculation
7 Intake manifold
8 Turbocharger
9 Exhaust pipe
10 Cooling system
11 Connecting rod
12 Lubrication system
13 Cylinder block
14 Crankshaft
15 Flywheel

K. Reif (Ed.), *Fundamentals of Automotive and Engine Technology*,
DOI 10.1007/978-3-658-03972-1_4, © Springer Fachmedien Wiesbaden 2014

2 Operating cycle of a four-stroke diesel engine

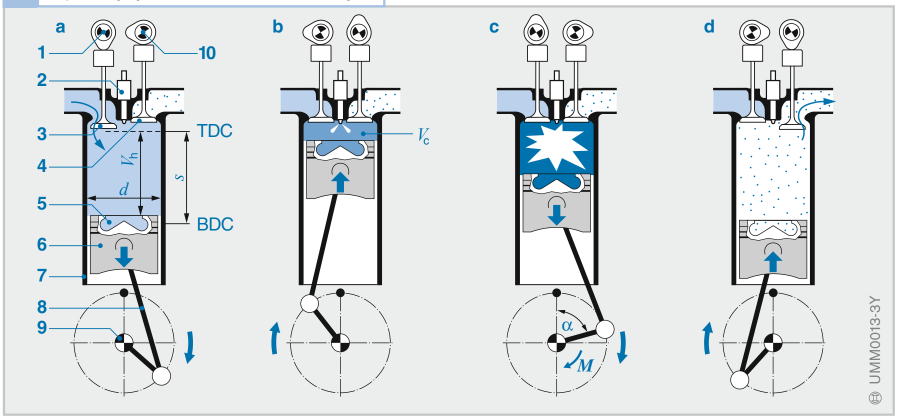


Fig. 2
- a Induction stroke
- b Compression stroke
- c Ignition stroke
- d Exhaust stroke

1. Inlet-valve camshaft
2. Fuel injector
3. Inlet valve
4. Exhaust valve
5. Combustion chamber
6. Piston
7. Cylinder wall
8. Connecting rod
9. Crankshaft
10. Exhaust-valve camshaft

- α Crankshaft angle of rotation
- d Bore
- M Turning force
- s Piston stroke
- V_c Compression volume
- V_h Swept volume

TDC Top dead center
BDC Bottom dead center

Four-stroke cycle

On a four-stroke diesel engine (Fig. 2), inlet and exhaust valves control the intake of air and expulsion of burned gases after combustion. They open and close the cylinder's inlet and exhaust ports. Each inlet and exhaust port may have one or two valves.

1. Induction stroke (a)

Starting from **T**op **D**ead **C**enter (TDC), the piston (6) moves downwards increasing the capacity of the cylinder. At the same time the inlet valve (3) is opened and air is drawn into the cylinder without restriction by a throttle valve. When the piston reaches **B**ottom **D**ead **C**enter (BDC), the cylinder capacity is at its greatest ($V_h + V_c$).

2. Compression stroke (b)

The inlet and exhaust valves are now closed. The piston moves upwards and compresses the air trapped inside the cylinder to the degree determined by the engine's compression ratio (this can vary from 6:1 in large-scale engines to 24:1 in car engines). In the process, the air heats up to temperatures as high as 900°C. When the compression stroke is almost complete, the fuel-injection system injects fuel at high pressure (as much as 2,000 bar in modern engines) into the hot, compressed air. When the piston reaches top dead center, the cylinder capacity is at its smallest (compression volume, V_c).

3. Ignition stroke (c)

After the ignition lag (a few degrees of crankshaft rotation) has elapsed, the ignition stroke (working cycle) begins. The finely atomized and easily combustible diesel fuel spontaneously ignites and burns due to the heat of the compressed air in the combustion chamber (5). As a result, the cylinder charge heats up even more and the pressure in the cylinder rises further as well. The amount of energy released by combustion is essentially determined by the mass of fuel injected (quality-based control). The pressure forces the piston downwards. The chemical energy released by combustion is thus converted into kinetic energy. The crankshaft drive translates the piston's kinetic energy into a turning force (torque) available at the crankshaft.

4. Exhaust stroke (d)

Fractionally before the piston reaches bottom dead center, the exhaust valve (4) opens. The hot, pressurized gases flow out of the cylinder. As the piston moves upwards again, it forces the remaining exhaust gases out.

On completion of the exhaust stroke, the crankshaft has completed two revolutions and the four-stroke operating cycle starts again with the induction stroke.

Valve timing

The cams on the inlet and exhaust camshafts open and close the inlet and exhaust valves respectively. On engines with a single camshaft, a rocker-arm mechanism transmits the action of the cams to the valves.

Valve timing involves synchronizing the opening and closing of the valves with the rotation of the crankshaft (Fig. 4). For that reason, valve timing is specified in degrees of crankshaft rotation.

The crankshaft drives the camshaft by means of a toothed belt or a chain (the timing belt or timing chain) or sometimes by a series of gears. On a four-stroke engine, a complete operating cycle takes two revolutions of the crankshaft. Therefore, the speed of rotation of the camshaft is only half that of the crankshaft. The transmission ratio between the crankshaft and the camshaft is thus 2 : 1.

At the changeover from exhaust to induction stroke, the inlet and exhaust valves are open simultaneously for a certain period of time. This "valve overlap" helps to "flush out" the remaining exhaust and cool the cylinders.

4 Valve-timing diagram for a four-stroke diesel engine

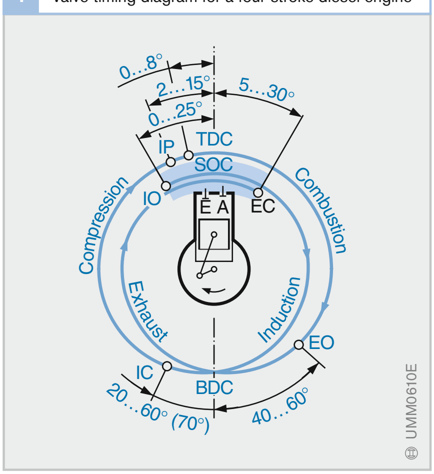


Fig. 4
EO Exhaust opens
EC Exhaust closes
SOC Start of combustion
IO Inlet opens
IC Inlet closes
IP Injection point
TDC Top dead center
BDC Bottom dead center

Valve overlap

3 Temperature rise during compression

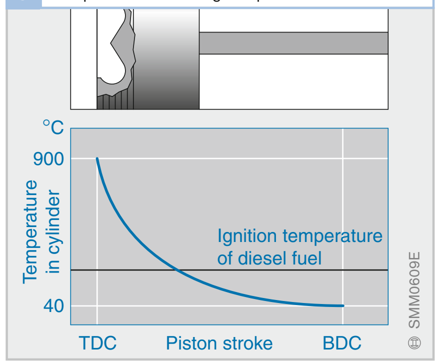


Fig. 3
TDC Top dead center
BDC Bottom dead center

Compression

The compression ratio, ε, of a cylinder results from its swept volume, V_h, and its compression volume, V_c, thus:

$$\varepsilon = \frac{V_h + V_c}{V_c}$$

The compression ratio of an engine has a decisive effect on the following:

- The engine's cold-starting characteristics
- The torque generated
- Its fuel consumption
- How noisy it is and
- The pollutant emissions

The compression ratio, ε, is generally between 16:1 and 24:1 in engines for cars and commercial vehicles, depending on the engine design and the fuel-injection method.
It is therefore higher than in gasoline engines ($\varepsilon = 7:1...13:1$). Due to the susceptibility of gasoline to knocking, higher compression ratios and the resulting higher combustion-chamber temperatures would cause the air/fuel mixture to spontaneously combust in an uncontrolled manner.

The air inside a diesel engine is compressed to a pressure of 30...50 bar (conventionally aspirated engine) or 70...150 bar (turbocharged/supercharged engine). This generates temperatures ranging from 700 to 900°C (Fig. 3). The ignition temperature of the most easily combustible components of diesel fuel is around 250°C.

Torque and power output

Torque

The conrod converts the linear motion of the piston into rotational motion of the crankshaft. The force with which the expanding air/fuel mixture forces the piston downwards is thus translated into rotational force or torque by the leverage of the crankshaft.

The output torque M of the engine is, therefore, dependent on mean pressure p_e (mean piston or operating pressure).
It is expressed by the equation:

$$M = p_e \cdot V_H / (4 \cdot \pi)$$

where
V_H is the cubic capacity of the engine and
$\pi \approx 3.14$.

The mean pressure can reach levels of 8...22 bar in small turbocharged diesel engines for cars. By comparison, gasoline engines achieve levels of 7...11 bar.

The maximum achievable torque, M_{max}, that the engine can deliver is determined by its design (cubic capacity, method of aspiration, etc.). The torque output is adjusted to the requirements of the driving situation essentially by altering the fuel and air mass and the mixing ratio.

Torque increases in relation to engine speed, n, until maximum torque, M_{max}, is reached (Fig. 1). As the engine speed increases beyond that point, the torque begins to fall again (maximum permissible engine load, desired performance, gearbox design).

Engine design efforts are aimed at generating maximum torque at low engine speeds (under 2,000 rpm) because at those speeds fuel consumption is at its most economical and the engine's response characteristics are perceived as positive (good "pulling power").

Power output

The power P (work per unit of time) generated by the engine depends on torque M and engine speed n. Engine power output increases with engine speed until it reaches its maximum level, or rated power P_{rated} at the engine's rated speed, n_{rated}. The following equation applies:

$$P = 2 \cdot \pi \cdot n \cdot M$$

Figure 1a shows a comparison between the power curves of diesel engines made in 1968 and in 1998 in relation to engine speed.

Due to their lower maximum engine speeds, diesel engines have a lower displacement-related power output than gasoline engines. Modern diesel engines for cars have rated speeds of between 3,500 and 5,000 rpm.

1 Torque and power curves for two diesel car engines with a capacity of approx. 2.2 *l* (example)

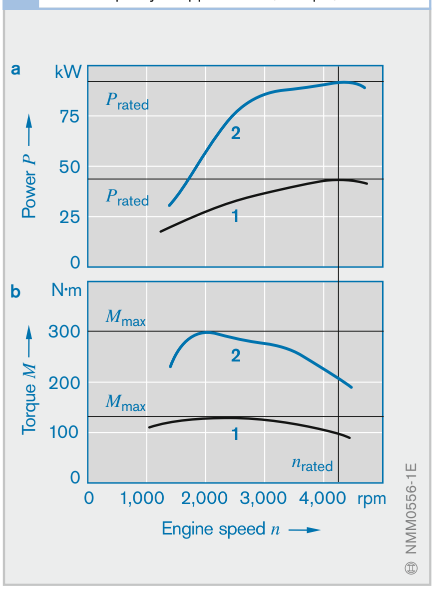


Fig. 1
a Power curve
b Torque curve

1 1968 engine
2 1998 engine

M_{max} Maximum torque
P_{rated} Rated power
n_{rated} Rated speed

Engine efficiency

The internal-combustion engine does work by changing the pressure and volume of a working gas (cylinder charge).

Effective efficiency of the engine is the ratio between input energy (fuel) and useful work. This results from the thermal efficiency of an ideal work process (Seiliger process) and the percentage losses of a real process.

Seiliger process

Reference can be made to the Seiliger process as a thermodynamic comparison process for the reciprocating-piston engine. It describes the theoretically useful work under ideal conditions. This ideal process assumes the following simplifications:

- Ideal gas as working medium
- Gas with constant specific heat
- No flow losses during gas exchange

1 Seiliger process for diesel engines

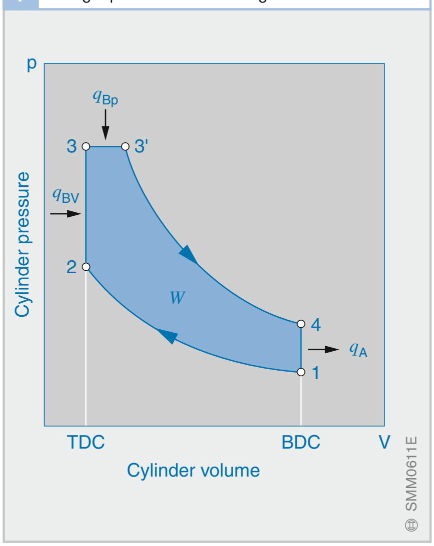


Fig. 1

1-2 Isentropic compression
2-3 Isochoric heat propagation
3-3′ Isobaric heat propagation
3′-4 Isentropic expansion
4-1 Isochoric heat dissipation

TDC Top dead center
BDC Bottom dead center

q_A Quantity of heat dissipated during gas exchange
q_{Bp} Combustion heat at constant pressure
q_{BV} Combustion heat at constant volume
W Theoretical work

The state of the working gas can be described by specifying pressure (p) and volume (V). Changes in state are presented in the p-V chart (Fig. 1), where the enclosed area corresponds to work that is carried out in an operating cycle.

In the Seiliger process, the following process steps take place:

Isentropic compression (1-2)

With isentropic compression (compression at constant entropy, i.e. without transfer of heat), pressure in the cylinder increases while the volume of the gas decreases.

Isochoric heat propagation (2-3)

The air/fuel mixture starts to burn. Heat propagation (q_{BV}) takes place at a constant volume (isochoric). Gas pressure also increases.

Isobaric heat propagation (3-3′)

Further heat propagation (q_{Bp}) takes place at constant pressure (isobaric) as the piston moves downwards and gas volume increases.

Isentropic expansion (3′-4)

The piston continues to move downwards to bottom dead center. No further heat transfer takes place. Pressure drops as volume increases.

Isochoric heat dissipation (4-1)

During the gas-exchange phase, the remaining heat is removed (q_A). This takes place at a constant gas volume (completely and at infinite speed). The initial situation is thus restored and a new operating cycle begins.

p-*V* chart of the real process

To determine the work done in the real process, the pressure curve in the cylinder is measured and presented in the p-V chart (Fig. 2). The area of the upper curve corresponds to the work present at the piston.

2 Real process in a turbocharged/supercharged diesel engine represented by p-V indicator diagram (recorded using a pressure sensor)

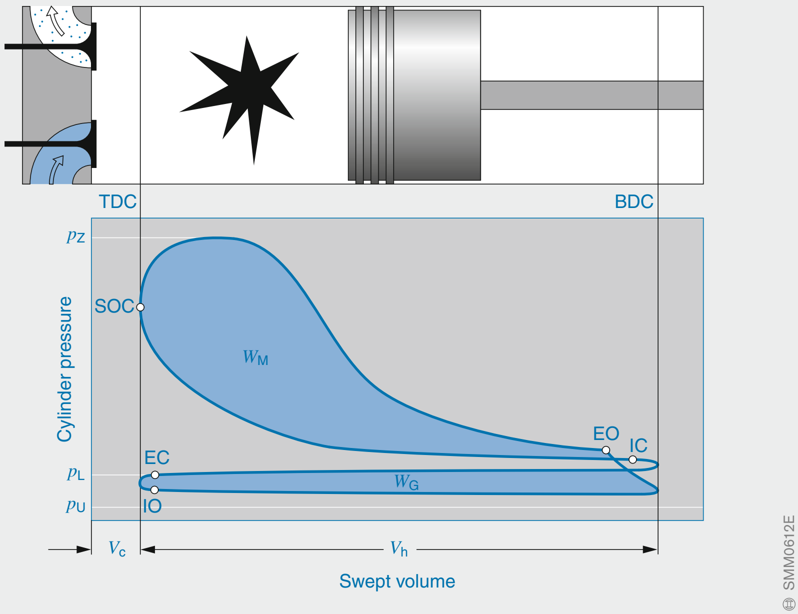


Fig. 2

EO Exhaust opens
EC Exhaust closes
SOC Start of combustion
IO Inlet opens
IC Inlet closes
TDC Top dead center
BDC Bottom dead center

p_U Ambient pressure
p_L Charge-air pressure
p_Z Maximum cylinder pressure
V_c Compression volume
V_h Swept volume
W_M Indexed work
W_G Work during gas exchange (turbocharger/ supercharger)

3 Pressure vs. crankshaft rotation curve (p-α diagram) for a turbocharged/supercharged diesel engine

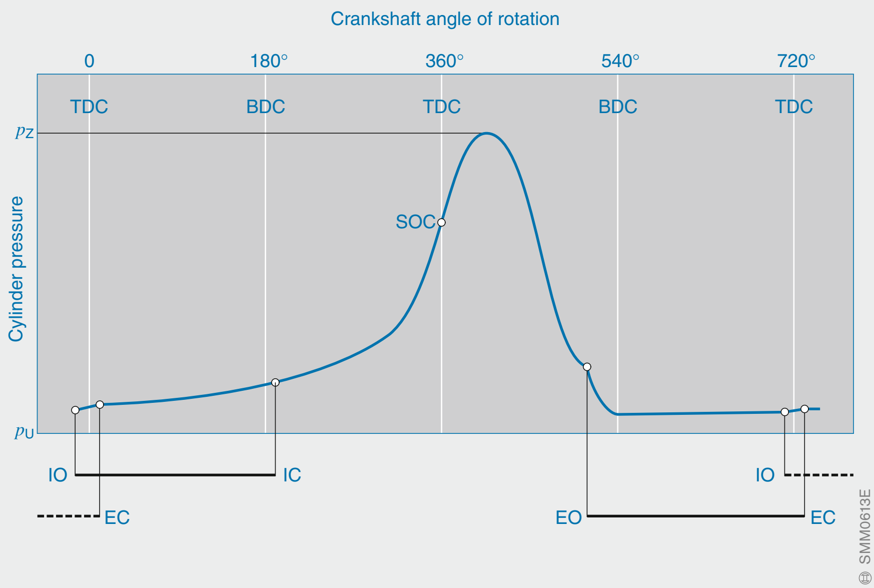


Fig. 3

EO Exhaust opens
EC Exhaust closes
SOC Start of combustion
IO Inlet opens
IC Inlet closes
TDC Top dead center
BDC Bottom dead center

p_U Ambient pressure
p_L Charge-air pressure
p_Z Maximum cylinder pressure

For assisted-aspiration engines, the gas-exchange area (W_G) has to be added to this since the compressed air delivered by the turbocharger/supercharger also helps to press the piston downwards on the induction stroke.

Losses caused by gas exchange are overcompensated at many operating points by the supercharger/turbocharger, resulting in a positive contribution to the work done.

Representation of pressure by means of the crankshaft angle (Fig. 3, previous page) is used in the thermodynamic pressure-curve analysis, for example.

Efficiency

Effective efficiency of the diesel engine is defined as:

$$\eta_e = \frac{W_e}{W_B}$$

W_e is the work effectively available at the crankshaft.
W_B is the calorific value of the fuel supplied.

Effective efficiency η_e is representable as the product of the thermal efficiency of the ideal process and other efficiencies that include the influences of the real process:

$$\eta_e = \eta_{th} \cdot \eta_g \cdot \eta_b \cdot \eta_m = \eta_i \cdot \eta_m$$

η_{th}: thermal efficiency

η_{th} is the thermal efficiency of the Seiliger process. This process considers heat losses occurring in the ideal process and is mainly dependent on compression ratio and excess-air factor.

As the diesel engine runs at a higher compression ratio than a gasoline engine and a high excess-air factor, it achieves higher efficiency.

η_g: efficiency of cycle factor

η_g specifies work done in the real high-pressure work process as a factor of the theoretical work of the Seiliger process.

Deviations between the real and the ideal processes mainly result from use of a real working gas, the finite velocity of heat propagation and dissipation, the position of heat propagation, wall heat loss, and flow losses during the gas-exchange process.

η_b: fuel conversion factor

η_b considers losses occurring due to incomplete fuel combustion in the cylinder.

η_m: mechanical efficiency

η_m includes friction losses and losses arising from driving ancillary assemblies. Frictional and power-transmission losses increase with engine speed. At nominal speed, frictional losses are composed of the following:

- Pistons and piston rings approx. 50%
- Bearings approx. 20%
- Oil pump approx. 10%
- Coolant pump approx. 5%
- Valve-gear approx. 10%
- Fuel-injection pump approx. 5%

If the engine has a supercharger, this must also be included.

η_i: efficiency index

The efficiency index is the ratio between "indexed" work present at the piston W_i and the calorific value of the fuel supplied.

Work effectively available at the crankshaft W_e results from indexed work taking mechanical losses into consideration:
$W_e = W_i \cdot \eta_m$.

Operating statuses

Starting

Starting an engine involves the following stages: cranking, ignition and running up to self-sustained operation.

The hot, compressed air produced by the compression stroke has to ignite the injected fuel (combustion start). The minimum ignition temperature required for diesel fuel is approx. 250°C.

This temperature must also be reached in poor conditions. Low engine speeds, low outside temperatures, and a cold engine lead to relatively low final compression temperatures due to the fact that:

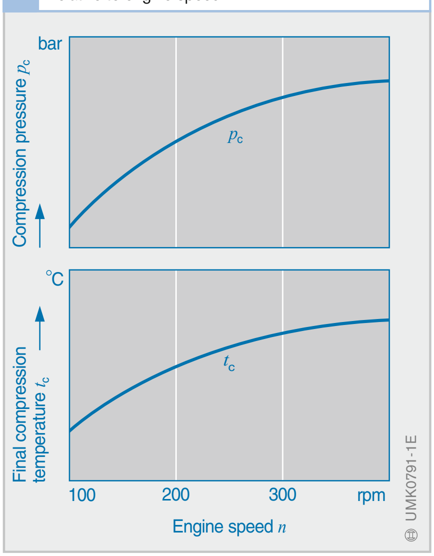
- The lower the engine speed, the lower the ultimate pressure at the end of the compression stroke and, accordingly, the ultimate temperature (Fig. 1). The reasons for this phenomenon are leakage losses through the piston ring gaps between the piston and the cylinder wall and the fact that when the engine is first started, there is no thermal expansion and an oil film has not formed. Due to heat loss during compression, maximum compression temperature is reached a few degrees before TDC (thermodynamic loss angle, Fig. 2).
- When the engine is cold, heat loss occurs across the combustion-chamber surface area during the compression stroke. On indirect-injection (IDI) engines, this heat loss is particularly high due to the larger surface area.
- Internal engine friction is higher at low temperatures than at normal operating temperature because of the higher viscosity of the engine oil. For this reason, and also due to low battery voltage, the starter-motor speed is only relatively low.
- The speed of the starter motor is particularly low when it is cold because the battery voltage drops at low temperatures.

The following measures are taken to raise temperature in the cylinder during the starting phase:

Fuel heating

A filter heater or direct fuel heater (Fig. 3 on next page) can prevent the precipitation of paraffin crystals that generally occurs at low

1 Compression pressure and ultimate temperature relative to engine speed



2 Compression temperature as a factor of crankshaft angle

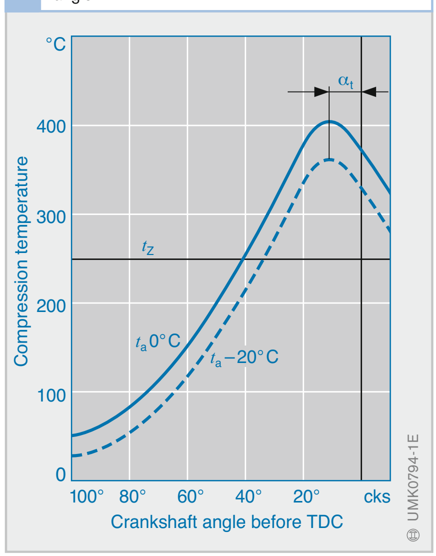


Fig. 2
t_a Outside temperature
t_Z Ignition temperature of diesel fuel
α_T Thermodynamic loss angle

$n \approx 200$ rpm

temperatures (during the starting phase and at low outside temperatures).

Start-assist systems

The air/fuel mixture in the combustion chamber (or in the prechamber or whirl chamber) is normally heated by sheathed-element glow plugs in the starting phase on direct-injection (DI) engines for passenger cars, or indirect-injection engines (IDI). On direct-injection (DI) engines for commercial vehicles, the intake air is preheated. Both the above methods assist fuel vaporization and air/fuel mixing and therefore facilitate reliable combustion of the air/fuel mixture.

Glow plugs of the latest generation require a preheating time of only a few seconds (Fig. 4), thus allowing a rapid start. The lower post-glow temperature also permits longer post-glow times. This reduces not only harmful pollutant emissions but also noise levels during the engine's warm-up period.

Injection adaptation

Another means of assisted starting is to inject an excess amount of fuel for starting to compensate for condensation and leakage losses in the cold engine, and to increase engine torque in the running-up phase.

Advancing the start of injection during the warming-up phase helps to offset longer ignition lag at low temperatures and to ensure reliable ignition at top dead center, i.e. at maximum final compression temperature.

The optimum start of injection must be achieved within tight tolerance limits. As the internal cylinder pressure (compression pressure) is still too low, fuel injected too early has a greater penetration depth and precipitates on the cold cylinder walls. There, only a small proportion of it vaporizes since then the temperature of the air charge is too low.

If the fuel is injected too late, ignition occurs during the downward stroke (expansion phase), and the piston is not fully accelerated, or combustion misses occur.

3 Diesel fuel heater

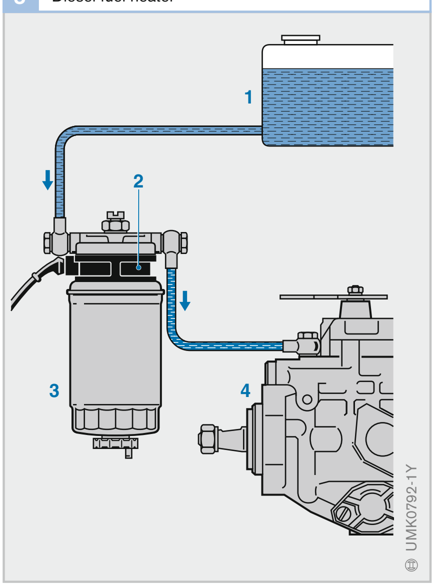


Fig. 3
1 Fuel tank
2 Fuel heater
3 Fuel filter
4 Fuel-injection pump

4 Temperature progression of two glow plugs in still air

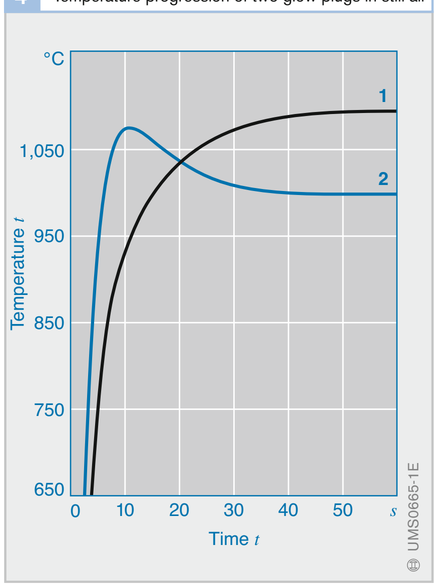


Fig. 4
Filament material:
1 Nickel (conventional glow plug type S-RSK)
2 CoFe alloy (2nd-generation glow plug type GSK2)

No load
No load refers to all engine operating statuses in which the engine is overcoming only its own internal friction. It is not producing any torque output. The accelerator pedal may be in any position. All speed ranges up to and including breakaway speed are possible.

Idle
The engine is said to be idling when it is running at the lowest no-load speed. The accelerator pedal is not depressed. The engine does not produce any torque. It only overcomes its internal friction. Some sources refer to the entire no-load range as idling. The upper no-load speed (breakaway speed) is then called the upper idle speed.

Full load
At full load (or Wide-Open Throttle (WOT)), the accelerator pedal is fully depressed, or the full-load delivery limit is controlled by the engine management dependent on the operating point. The maximum possible fuel volume is injected and the engine generates its maximum possible torque output under steady-state conditions. Under non steady-state conditions (limited by turbocharger/supercharger pressure) the engine develops the maximum possible (lower) full-load torque with the quantity of air available. All engine speeds from idle speed to nominal speed are possible.

Part load
Part load covers the range between no load and full load. The engine is generating an output between zero and the maximum possible torque.

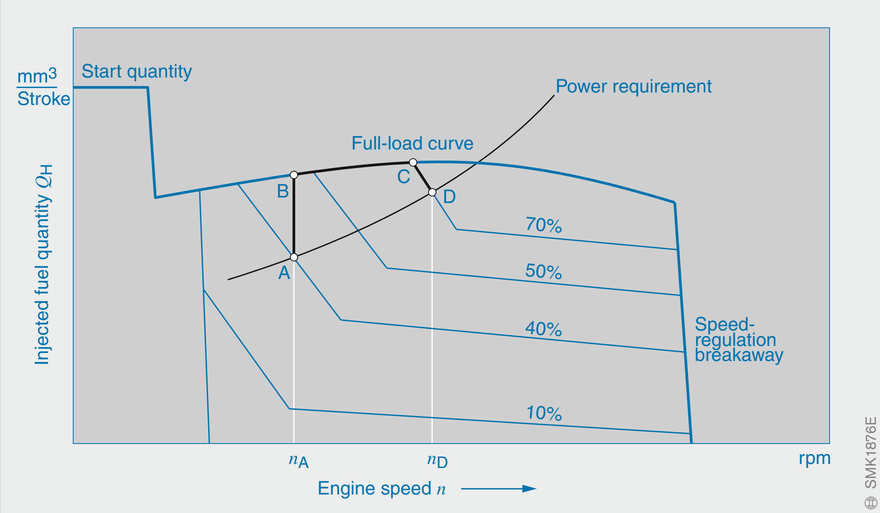
Lower part-load range
This is the operating range in which the diesel engine's fuel consumption is particularly economical in comparison with the gasoline engine. "Diesel knock" that was a problem on earlier diesel engines – particularly when cold – has virtually been eliminated on diesels with pre-injection.

As explained in the "Starting" section, the final compression temperature is lower at lower engine speeds and at lower loads. In comparison with full load, the combustion chamber is relatively cold (even when the engine is running at operating temperature) because the energy input and, therefore, the temperatures, are lower. After a cold start, the combustion chamber heats up very slowly in the lower part-load range. This is particularly true for engines with prechamber or whirl chambers because the larger surface area means that heat loss is particularly high.

At low loads and with pre-injection, only a few mm^3 of fuel are delivered in each injection cycle. In this situation, particularly high demands are placed on the accuracy of the start of injection and injected fuel quantity. As during the starting phase, the required combustion temperature is reached also at idle speed only within a small range of piston travel near TDC. Start of injection is controlled very precisely to coincide with that point.

During the ignition-lag period, only a small amount of fuel may be injected since, at the point of ignition, the quantity of fuel in the combustion chamber determines the sudden increase in pressure in the cylinder.

5 Injected-fuel quantity as a factor of engine speed and accelerator-pedal position (example)



The greater the increase in pressure, the louder the combustion noise. Pre-injection of approx. 1 mm³ (for cars) of fuel virtually cancels out ignition lag at the main injection point, and thus substantially reduces combustion noise.

Overrun
The engine is said to be overrunning when it is driven by an external force acting through the drivetrain (e.g. when descending an incline). No fuel is injected (overrun fuel cut-off).

Steady-state operation
Torque delivered by the engine corresponds to the torque required by the accelerator-pedal position. Engine speed remains constant.

Non-steady-state operation
The engine's torque output does not equal the required torque. The engine speed is not constant.

Transition between operating statuses
If the load, the engine speed, or the accelerator-pedal position change, the engine's operating state changes (e.g. its speed or torque output).

The response characteristics of an engine can be defined by means of characteristic data diagrams or maps. The map in Figure 5 shows an example of how the engine speed changes when the accelerator-pedal position changes from 40% to 70% depressed. Starting from operating point A, the new part-load operating point D is reached via the full-load curve (B-C). There, power demand and engine power output are equal. The engine speed increases from n_A to n_D.

Operating conditions

In a diesel engine, the fuel is injected directly into the highly compressed hot air which causes it to ignite spontaneously. Therefore, and because of the heterogeneous air/fuel mixture, the diesel engine – in contrast with the gasoline engine – is not restricted by ignition limits (i.e. specific air-fuel ratios λ). For this reason, at a constant air volume in the cylinder, only the fuel quantity is controlled.

The fuel-injection system must assume the functions of metering the fuel and distributing it evenly over the entire charge. It must accomplish this at all engine speeds and loads, dependent on the pressure and temperature of the intake air.

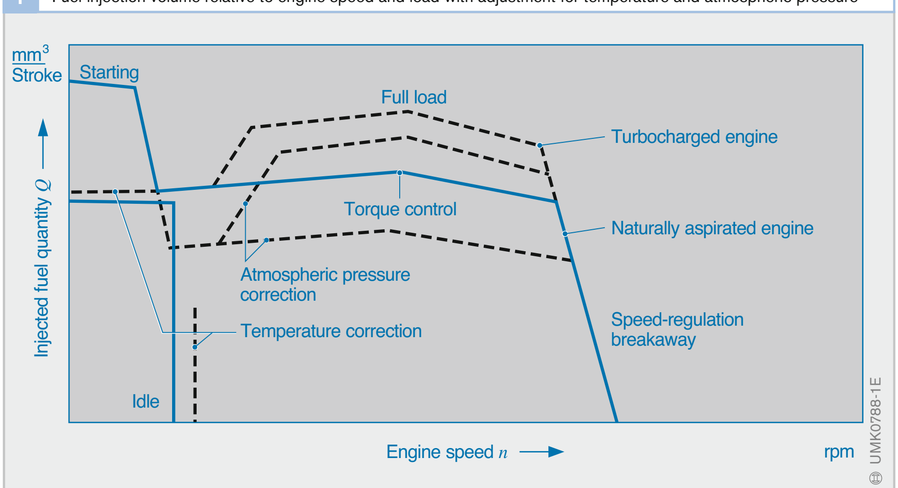
Thus, for any combination of engine operating parameters, the fuel-injection system must deliver:
- The correct amount of fuel
- At the correct time
- At the correct pressure
- With the correct timing pattern and at the correct point in the combustion chamber

In addition to optimum air/fuel mixture considerations, metering the fuel quantity also requires taking account of operating limits such as:
- Emission restrictions (e.g. smoke emission limits)
- Combustion-peak pressure limits
- Exhaust temperature limits
- Engine speed and full-load limits
- Vehicle or engine-specific load limits, and
- Altitude and turbocharger/supercharger pressure limits

Smoke limit
There are statutory limits for particulate emissions and exhaust-gas turbidity. As a large part of the air/fuel mixing process only takes place during combustion, localized over-enrichment occurs, and, in some cases, this leads to an increase in soot-particle emissions, even at moderate levels of excess air. The air-fuel ratio usable at the statutory full-load smoke limit is a measure of the efficiency of air utilization.

Combustion pressure limits
During the ignition process, the partially vaporized fuel mixed with air burns at high compression, at a rapid rate, and at a high

1 Fuel-injection volume relative to engine speed and load with adjustment for temperature and atmospheric pressure



initial thermal-release peak. This is referred to as "hard" combustion. High final compression peak pressures occur during this phenomenon, and the resulting forces exert stresses on engine components and are subject to periodic changes. The dimensioning and durability of the engine and drivetrain components, therefore, limit the permissible combustion pressure and, consequently, the injected fuel quantity. The sudden rise in combustion pressure is mostly counteracted by pre-injection.

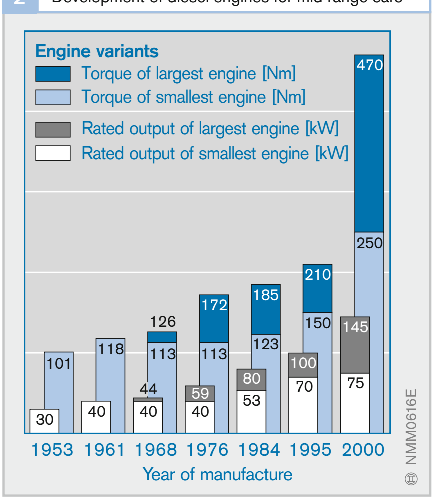
Exhaust-gas temperature limits

The high thermal stresses placed on the engine components surrounding the hot combustion chamber, the heat resistance of the exhaust valves and of the exhaust system and cylinder head determine the maximum exhaust temperature of a diesel engine.

Engine speed limits

Due to the existing excess air in the diesel engine, power at constant engine speed mainly depends on injected fuel quantity. If the amount of fuel supplied to a diesel engine is increased without a corresponding increase in the load that it is working against, then the engine speed will rise. If the fuel supply is not reduced before the engine reaches a critical speed, the engine may exceed its maximum permitted engine speed, i.e. it could self-destruct. Consequently, an engine speed limiter or governor is absolutely essential on a diesel engine.

2 Development of diesel engines for mid-range cars



On diesel engines used to drive road-going vehicles, the engine speed must be infinitely variable by the driver using the accelerator pedal. In addition, when the engine is under load or when the accelerator pedal is released, the engine speed must not be allowed to drop below the idling speed to a standstill. This is why a minimum-maximum-speed governor is fitted. The speed range between these two points is controlled using the accelerator-pedal position. If the diesel engine is used to drive a machine, it is expected to keep to a specific speed constant, or remain within permitted limits, irrespective of load. A variable-speed governor is then fitted to control speed across the entire range.

A program map is definable for the engine operating range. This map (Fig. 1, previous page) shows the fuel quantity in relation to engine speed and load, and the necessary adjustments for temperature and air-pressure variations.

Altitude and turbocharger/supercharger pressure limits

The injected fuel quantity is usually designed for sea level. If the engine is operated at high elevations (height above mean sea level), the fuel quantity must be adjusted in relation to the drop in air pressure in order to comply with smoke limits. A standard value is the barometric elevation formula, i.e. air density decreases by approximately 7% per 1,000 m of elevation.

With turbocharged engines, the cylinder charge in dynamic operation is often lower than in static operation. Since the maximum injected fuel quantity is designed for static operation, it must be reduced in dynamic operation in line with the lower air-flow rate (full-load limited by charge-air pressure).

Fuel-injection system

The low-pressure fuel supply conveys fuel from the fuel tank and delivers it to the fuel-injection system at a specific supply pressure. The fuel-injection pump generates the fuel pressure required for injection. In most systems, fuel runs through high-pressure delivery lines to the injection nozzle and is injected into the combustion chamber at a pressure of 200...2,200 bar on the nozzle side.

Engine power output, combustion noise, and exhaust-gas composition are mainly influenced by the injected fuel mass, the injection point, the rate of discharge, and the combustion process.

Up to the 1980s, fuel injection, i.e. the injected fuel quantity and the start of injection on vehicle engines, was mostly controlled mechanically. The injected fuel quantity is then varied by a piston timing edge or via slide valves, depending on load and engine speed. Start of injection is adjusted by mechanical control using flyweight governors, or hydraulically by pressure control (see section entitled "Overview of diesel fuel-injection systems").

Now electronic control has fully replaced mechanical control – not only in the automotive sector. **E**lectronic **D**iesel **C**ontrol (EDC) manages the fuel-injection process by involving various parameters, such as engine speed, load, temperature, geographic elevation, etc. in the calculation. Start of injection and fuel injection quantity are controlled by solenoid valves, a process that is much more precise than mechanical control.

Size of injection

An engine developing 75 kW (102 HP) and a specific fuel consumption of 200 g/kWh (full load) consumes 15 kg fuel per hour. On a 4-stroke 4-cylinder engine, the fuel is distributed by 288,000 injections at 2,400 revs per minute. This results in a fuel volume of approx. 60 mm³ per injection. By comparison, a raindrop has a volume of approximately 30 mm³.

Even greater precision in metering requires an idle with approx. 5 mm³ fuel per injection and a pre-injection of only 1 mm³. Even the minutest variations have a negative effect on the smooth running of the engine, noise, and pollutant emissions.

The fuel-injection system not only has to deliver precisely the right amount of fuel for each individual, it also has to distribute the fuel evenly to the individual cylinder of an engine. Electronic Diesel Control (EDC) adapts the injected fuel quantity for each cylinder in order to achieve a particularly smooth-running engine.

Combustion chambers

The shape of the combustion chamber is one of the decisive factors in determining the quality of combustion and therefore the performance and exhaust characteristics of a diesel engine. Appropriate design of combustion-chamber geometry combined with the action of the piston can produce whirl, squish, and turbulence effects that are used to improve distribution of the liquid fuel or air/fuel vapor spray inside of the combustion chamber.

The following technologies are used:
- Undivided combustion chamber (**D**irect **I**njection (DI) engines) and
- Divided combustion chamber (**I**ndirect **I**njection (IDI) engines)

The proportion of direct-injection engines is increasing due to their more economical fuel consumption (up to 20% savings). The harsher combustion noise (particularly under acceleration) can be reduced to the level of indirect-injection engines by pre-injection. Engines with divided combustion chambers now hardly figure at all among new developments.

1 Direct injection

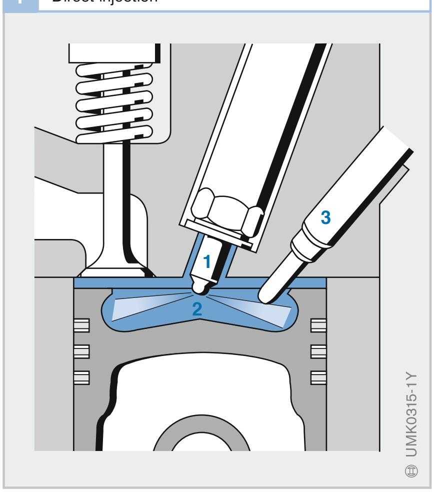


Fig. 1
1 Multihole injector
2 ω piston recess
3 Glow plug

Undivided combustion chamber (direct-injection engines)

Direct-injection engines (Fig. 1) have a higher level of efficiency and operate more economically than indirect-injection engines. Accordingly, they are used in all types of commercial vehicles and most modern diesel cars.

As the name suggests, the direct-injection process involves injecting the fuel directly into the combustion chamber, part of which is formed by the shape of the piston crown (piston crown recess, 2). Fuel atomization, heating, vaporization and mixing with the air must therefore take place in rapid succession. This places exacting demands on fuel and air delivery. During the induction and compression strokes, the special shape of the intake port in the cylinder head creates an air vortex inside of the cylinder. The shape of the combustion chamber also contributes to the air flow pattern at the end of the compression stroke (i.e. at the moment of fuel injection). Of the combustion chamber designs used over the history of the diesel engine, the most widely used at present is the ω piston crown recess.

In addition to creating effective air turbulence, the technology must also ensure that fuel is delivered in such a way that it is evenly distributed throughout the combustion chamber to achieve rapid mixing. A multihole nozzle is used in the direct-injection process and its nozzle-jet position is optimized as a factor of combustion-chamber design. Direct fuel injection requires very high injection pressures (up to 2,200 bar).

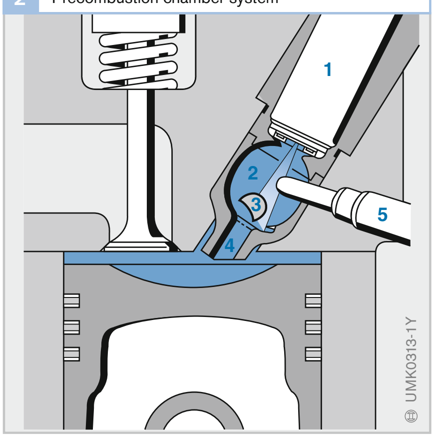
In practice, there are two types of direct fuel injection:
- Systems in which mixture formation is assisted by specifically created air-flow effects and
- Systems which control mixture formation virtually exclusively by means of fuel injection and largely dispense with any air-flow effects

In the latter case, no effort is expended in creating air-turbulence effects and this is evident in smaller gas replacement losses and more effective cylinder charging. At the same time, however, far more demanding requirements are placed on the fuel-injection system with regard to injection-nozzle positioning, the number of nozzle jets, the degree of atomization (dependent on spray-hole diameter), and the intensity of injection pressure in order to obtain the required short injection times and quality of the air/fuel mixture.

Divided combustion chamber (indirect injection)

For a long time diesel engines with divided combustion chambers (indirect-injection engines) held an advantage over direct-injection engines in terms of noise and exhaust-gas emissions. That was the reason why they were used in cars and light commercial vehicles. Now direct-injection engines are more economical than IDI engines, with comparable noise emissions as a result of their high injection pressures, electronic diesel control, and pre-injection. As a result, indirect-injection engines are no longer used in new vehicles.

2 Precombustion chamber system



There are two types of processes with divided combustion chamber:

- The precombustion chamber system and
- The whirl-chamber system

Precombustion chamber system

In the prechamber (or precombustion chamber) system, fuel is injected into a hot prechamber recessed into the cylinder head (Fig. 2, 2). The fuel is injected through a pintle nozzle (1) at a relatively low pressure (up to 450 bar). A specially shaped baffle (3) in the center of the chamber diffuses the jet of fuel that strikes it and mixes it thoroughly with the air.

Combustion starting in the prechamber drives the partly combusted air/fuel mixture through the connecting channel (4) into the main combustion chamber. Here and further down the combustion process, the injected fuel is mixed intensively with the existing air. The ratio of precombustion chamber volume to main combustion chamber volume is approx. 1 : 2.

The short ignition lag[1]) and the gradual release of energy produce a soft combustion effect with low levels of noise and engine load.

[1]) Time from start of injection to start of ignition

A differently shaped prechamber with an evaporation recess and a different shape and position of the baffle (spherical pin) apply a specific degree of whirl to the air that passes from the cylinder into the prechamber during the compression stroke. The fuel is injected at an angle of 5 degrees in relation to the prechamber axis.

So as not to disrupt the progression of combustion, the glow plug (5) is positioned on the “lee side” of the air flow. A controlled post-glow period of up to 1 minute after a cold start (dependent on coolant temperature) helps to improve exhaust-gas characteristics and reduce engine noise during the warm-up period.

Fig. 2
1 Nozzle
2 Precombustion chamber
3 Baffle surface
4 Connecting channel
5 Glow plug

Swirl-chamber system

With this process, combustion is also initiated in a separate chamber (swirl chamber) that has approx. 60% of the compression volume. The spherical and disk-shaped swirl chamber is linked by a connecting channel that discharges at a tangent into the cylinder chamber (Fig. 3, 2).

During the compression cycle, air entering via the connecting channel is set into a swirling motion. The fuel is injected so that the swirl penetrates perpendicular to its axis and meets a hot section of the chamber wall on the opposite side of the chamber.

As soon as combustion starts, the air/fuel mixture is forced under pressure through the connecting channel into the cylinder chamber where it is turbulently mixed with the remaining air. With the swirl-chamber system, the losses due to gas flow between the main combustion chamber and the swirl chamber are less than with the precombustion chamber system because the connecting channel has a larger cross-section. This results in smaller throttle-effect losses and consequent benefits for internal efficiency and fuel consumption. However, combustion noise is louder than with the precombustion chamber system.

It is important that mixture formation takes place as completely as possible inside the swirl chamber. The shape of the swirl chamber, the alignment and shape of the fuel jet and the position of the glow plug must be carefully matched to the engine in order to obtain optimum mixture formation at all engine speeds and under all operating conditions.

Another demand is for rapid heating of the swirl chamber after a cold start. This reduces ignition lag and combustion noise as well as preventing unburned hydrocarbons (blue smoke) during the warm-up period.

3 Swirl-chamber system

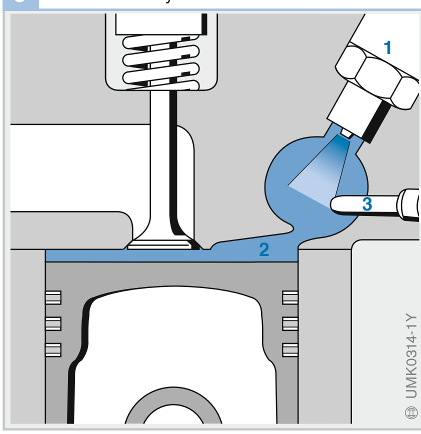
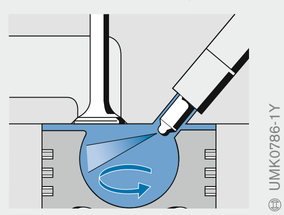


Fig. 3
1 Fuel injector
2 Tangential connecting channel
3 Glow plug

M System

In the direct-injection system with recess-wall deposition (M system) for commercial-vehicle and fixed-installation diesel engines and multi-fuel engines, a single-jet nozzle sprays the fuel at a low injection pressure against the wall of the piston crown recess. There, it vaporizes and is absorbed by the air. This system thus uses the heat of the piston recess wall to vaporize the fuel. If the air flow inside of the combustion chamber is properly adapted, an extremely homogeneous air/fuel mixture with a long combustion period, low pressure increase and, therefore, quiet combustion can be achieved. Due to its consumption disadvantages compared with the air-distributing direct injection process, the M system is no longer used in modern applications.



Fuel consumption in everyday practice

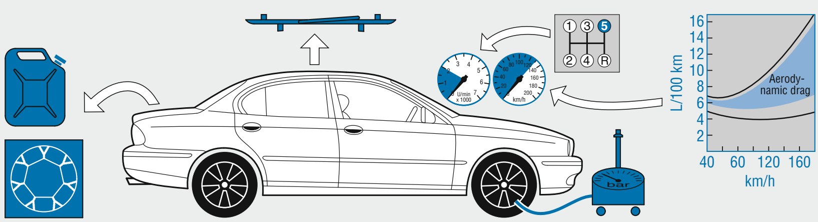
Automotive manufacturers are obliged by law to specify the fuel consumption of their vehicles. This figure is determined from the exhaust-gas emissions during the exhaust-gas test when the vehicle travels a specific route profile (test cycle). The fuel consumption figures are therefore comparable for all vehicles.

Every driver makes a significant contribution to reducing fuel consumption by his or her driving style. Reducing the fuel consumption that the driver can achieve with a vehicle depends on several factors.

Applying the measures listed below, an "economical" driver can reduce fuel consumption in everyday traffic by 20 to 30% compared to an average driver. The reduction in fuel consumption achievable by applying the individual measures depends on a number of factors, mainly the route profile (city streets, overland roads), and on traffic conditions. For this reason, it is not always practical to specify figures for fuel-consumption savings.

Positive influences on fuel consumption

- Tire pressure: Remember to increase tire pressure when the vehicle is carrying a full payload (saving: approx. 5%).
- When accelerating at high load and low engine speed, shift up at 2,000 rpm.
- Drive in the highest possible gear. You can even drive at full-load at engine speeds below 2,000 rpm.
- Avoid braking and re-accelerating by adopting a forward-looking style of driving.
- Use overrun fuel cutoff to the full.
- Switch off the engine when the vehicle is stopped for an extended period of time, e.g. at traffic lights with a long red phase, or at closed railroad crossings (3 minutes at idle consumes as much fuel as driving 1 km).
- Use high-lubricity engine oils (saving: approx. 2% according to manufacturer specifications).



Negative influences on fuel consumption

- Greater vehicle weight due to ballast, e.g. in the trunk (additional approx. 0.3 *l*/100 km).
- High-speed driving.
- Greater aerodynamic drag from carrying objects on the roof.
- Additional electrical equipment, e.g. rear-window heating, foglamps (approx. 1 *l*/1 kW).
- Dirty air filter.

Basic principles of diesel fuel injection

The combustion processes in the diesel engine, also linked to engine performance, fuel consumption, exhaust-gas composition, and combustion noise, depend to a great extent on how the air/fuel mixture is prepared.

The fuel-injection parameters that are decisive on the quality of the mixture formation are primarily:
- start of injection
- rate-of-discharge curve and injection duration
- injection pressure
- number of injection events

On the diesel engine, exhaust-gas and noise emissions are largely reduced by measures inside of the engine, i.e. combustion-process control.

Until the 1980s injected fuel quantity and start of injection were controlled on vehicle engines by mechanical means only. However, compliance with prevailing emission limits requires the high-precision adjustment of injection parameters, e.g. pre-injection, main injection, injected fuel quantity, injection pressure, and start of injection, adapted to the engine operating state. This is only achievable using an electronic control unit that calculates injection parameters as a factor of temperature, engine speed, load, altitude (elevation), etc. Electronic Diesel Control (EDC) has generally become widespread on diesel engines.

As exhaust-gas emission standards become more severe in future, further measures for minimizing pollutants will have to be introduced. Emissions, as well as combustion noise, can continue to be reduced by means of very high injection pressures, as achieved by the Unit Injector System, and by a rate-of-discharge curve that is adjustable independent of pressure buildup, as implemented by the common-rail system.

Mixture distribution

Excess-air factor λ

The excess-air factor λ (lambda) was introduced to indicate the degree by which the actual air/fuel mixture actually deviates from the stoichiometric[1] mass ratio. It indicates the ratio of intake air mass to required air mass for stoichiometric combustion, thus:

$$\lambda = \frac{\textit{Air mass}}{\textit{Fuel mass} \cdot \textit{Stoichiometric ratio}}$$

$\lambda = 1$: The intake air mass is equal to the air mass theoretically required to burn all of the fuel injected.

$\lambda < 1$: The intake air mass is less than the amount required and therefore the mixture is rich.

$\lambda > 1$: The intake air mass is greater than the amount required and therefore the mixture is lean.

[1]) The stoichiometric ratio indicates the air mass in kg required to completely burn 1 kg of fuel (m_L/m_K). For diesel fuel, this is approx. 14.5.

1 Progress of combustion in a direct-injection test engine with a multihole nozzle

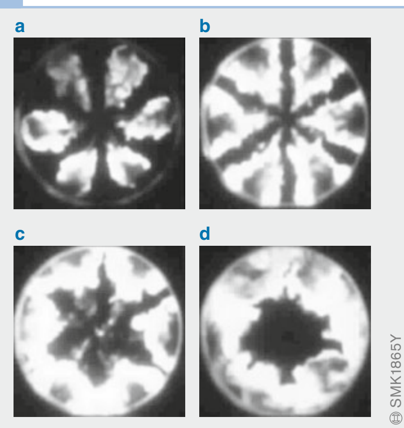


Fig. 1
Special engines with glass inserts and mirrors allow observation of the fuel injection and combustion processes.

The times are measured from the start of spontaneous combustion.
a 200 µs
b 400 µs
c 522 µs
d 1,200 µs

K. Reif (Ed.), *Fundamentals of Automotive and Engine Technology*,
DOI 10.1007/978-3-658-03972-1_5, © Springer Fachmedien Wiesbaden 2014

Lambda levels in diesel engines

Rich areas of mixture are responsible for sooty combustion. In order to prevent the formation of too many rich areas of mixture, diesel engines – in contrast to gasoline engines – have to be run with an overall excess of air.

The lambda levels for turbocharged diesel engines at full load are between $\lambda = 1.15$ and $\lambda = 2.0$. When idling and under no-load conditions, those figures rise to $\lambda > 10$.

These excess-air factors represent the ratio of total masses of fuel and air in the cylinder. However, the lambda factor, which is subject to strong spatial fluctuation, is primarily responsible for auto-ignition and the production of pollutants.

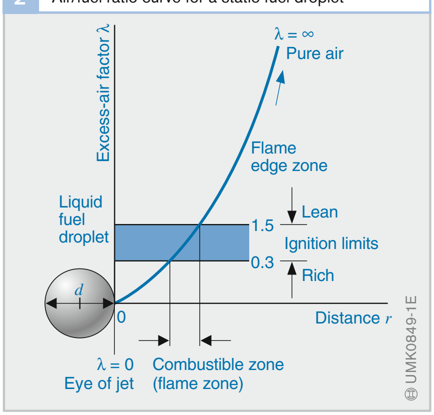
Diesel engines operate with heterogeneous mixture formation and auto-ignition. It is not possible to achieve completely homogeneous mixing of the injected fuel with the air charge prior to or during combustion. Within the heterogeneous mixture encountered in a diesel engine, the localized excess-air factors can cover the entire range from $\lambda = 0$ (pure fuel) in the eye of the jet close to the injector to $\lambda = \infty$ (pure air) at the outer extremities of the spray jet. Around the outer zone of a single liquid droplet (vapor envelope), there are localized lambda levels of 0.3 to 1.5 (Figs. 2 and 3). From this, it can be deduced that optimized atomization (large numbers of very small droplets), high levels of excess air, and "metered" motion of the air charge produce large numbers of localized zones with lean, combustible lambda levels. This results in less soot occurring during combustion. EGR compatibility then rises, and NO_x emissions are reduced.

Optimized fuel atomization is achieved by high injection pressures that range up to max. 2,200 bar for UIS. Common-rail systems (CRS) operate at an injection pressure of max. 1,800 bar. This results is a high relative velocity between the jet of fuel and the air in the cylinder which has the effect of scattering the fuel jet.

With a view to reducing engine weight and cost, the aim is to obtain as much power as possible from a given engine capacity. To achieve this, the engine must run on the lowest possible excess air at high loads. On the other hand, a deficiency in excess air increases the amount of soot emissions. Therefore, soot has to be limited by precisely metering the injected fuel quantity to match the available air mass as a factor of engine speed.

Low atmospheric pressures (e.g. at high altitudes) also require the fuel volume to be adjusted to the smaller amount of available air.

2 Air/fuel ratio curve for a static fuel droplet



3 Air/fuel ratio curve for a moving fuel droplet

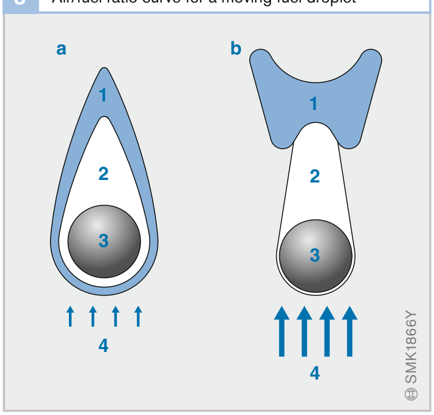


Fig. 2
d Droplet diameter (approx. 2...20 µm)

Fig. 3
a Low relative velocity
b High relative velocity

1 Flame zone
2 Vapor envelope
3 Fuel droplet
4 Air flow

Fuel-injection parameters

Start of injection and delivery

Start of injection

The point at which injection of fuel into the combustion chamber starts has a decisive effect on the point at which combustion of the air/fuel mixture starts, and therefore on emission levels, fuel consumption and combustion noise. For this reason, start of injection plays a major role in optimizing engine performance characteristics.

Start of injection specifies the position stated in degrees of crankshaft rotation relative to crankshaft Top Dead Center (TDC) at which the injection nozzle opens, and fuel is injected into the engine combustion chamber.

The position of the piston relative to top dead center at that moment influences the flow of air inside of the combustion chamber, as well as air density and temperature. Accordingly, the degree of mixing of air and fuel is also dependent on start of injection. Thus, start of injection affects emissions such as soot, nitrogen oxides (NO_x), unburned hydrocarbons (HC), and carbon monoxide (CO).

The start-of-injection setpoints vary according to engine load, speed, and temperature. Optimized values are determined for each engine, taking into consideration the impacts on fuel consumption, pollutant emission, and noise. These values are then stored in a start-of-injection program map (Fig. 4). Load-dependent start-of-injection variability is controlled across the program map.

Compared with cam-controlled systems, common-rail systems offer more freedom in selecting the quantity and timing of injection events and injection pressure. As a consequence, fuel pressure is built up by a separate high-pressure pump, optimized to every operating point by the engine management system, and fuel injection is controlled by a solenoid valve or piezoelectric element.

4 Start of injection versus engine speed and load for a car engine started from cold and at normal operating temperature (example)

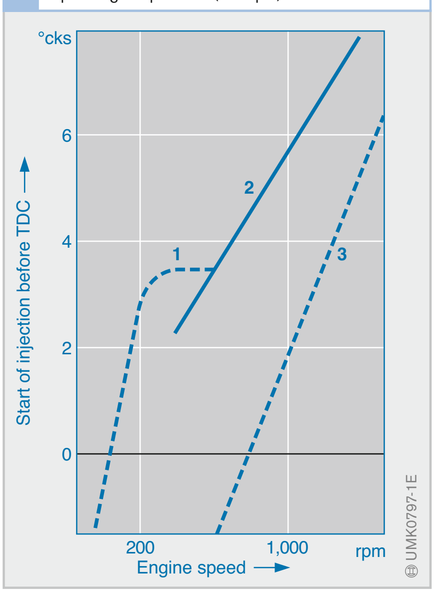


Fig. 4
1 Cold start (< 0°C)
2 Full load
3 Medium load

5 Distribution patterns for NO_X and HC emissions plotted against start of injection for a commercial vehicle without exhaust-gas recirculation

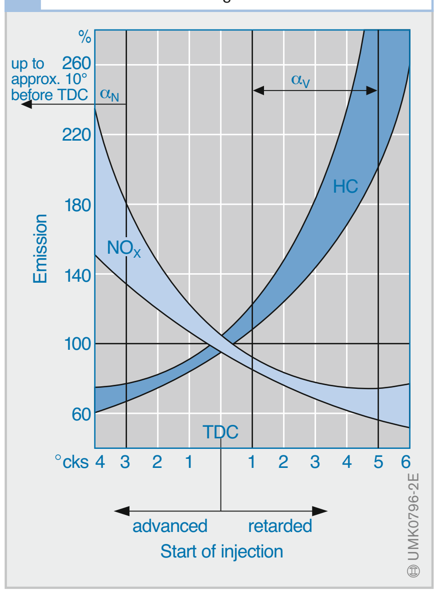


Fig. 5
Example of an application:
α_N Optimum start of injection at no-load: low HC emissions while NO_x emissions at no load are low anyway.
α_V Optimum start of injection at full load: low NO_x emissions while HC emissions are low at full load anyway.

Standard values for start of injection
On a diesel-engine data map, the optimum points of combustion start for low fuel consumption are in the range of 0...8° crankshaft angle before TDC. As a result, and based on statutory exhaust-gas emission limits, the start of injection points are as follows:

Direct-injection car engines:
- No load: 2° crankshaft angle before TDC to 4° crankshaft angle after TDC
- Part load: 6° crankshaft angle before TDC to 4° crankshaft angle after TDC
- Full load: 6 to 15° crankshaft angle before TDC

Direct-injection commercial-vehicle engines (without exhaust-gas recirculation):
- No load: 4 to 12° crankshaft angle before TDC
- Full load: 3 to 6° crankshaft angle before TDC to 2° crankshaft angle after TDC

When the engine is cold, the start of injection for car and commercial-vehicle engines is 3 to 10° earlier. Combustion time at full load is 40 to 60° crankshaft angle.

Advanced start of injection
The highest compression temperature (final compression temperature) occurs shortly before piston **T**op **D**ead **C**enter (TDC). If combustion starts a long way before TDC, combustion pressure rises steeply, and acts as a retarding force against the piston stroke. Heat lost in the process diminishes engine efficiency and, therefore, increases fuel consumption. The steep rise in compression pressure also makes combustion much noisier.

An advanced start of injection increases temperature in the combustion chamber. As a result, NO_x emission levels rise, but HC emissions are lower (Fig. 5).

Minimizing blue and white smoke levels requires advanced start of injection and/or pre-injection when the engine is cold.

Retarded start of injection
A retarded start of injection at low-load conditions can result in incomplete combustion and, therefore, in the emission of unburned hydrocarbons (HC) and carbon monoxide (CO) since the temperature in the combustion chamber is already dropping (Fig. 5).

The partially conflicting tradeoffs of specific fuel consumption and hydrocarbon emissions on the one hand, and soot (black smoke) and NO_x emissions on the other, demand compromises and very tight tolerances when modifying the start of injection to suit a particular engine.

Start of delivery
In addition to start of injection, start of delivery is another aspect that is often considered. It relates to the point at which the fuel-injection pump starts to deliver fuel to the injector.

On older fuel-injection systems, start of delivery plays an important role since the in-line or distributor injection pump must be allocated to the engine. The relative timing between pump and engine is fixed at start of delivery, since this is easier to define than the actual start of injection. This is made possible because there is a definite relationship between start of delivery and start of injection (injection lag[1])).

Injection lag results from the time it takes the pressure wave to travel from the high-pressure pump through to the injection nozzle. Therefore, it depends on the length of the line. At different engine speeds, there is a different injection lag measured as a crankshaft angle (degrees of crankshaft rotation). At higher engine speeds, the engine has a greater ignition lag[2]) related to the crankshaft position (in degrees of crankshaft angle). Both of these effects must be compensated for – which is why a fuel-injection system must be able to adjust the start of delivery/start of injection in response to engine speed, load, and temperature.

1) Time or crankshaft angle swept from start of delivery through start of injection

2) Time or crankshaft angle swept from start of injection through start of ignition

Injected-fuel quantity

The required fuel mass, m_e, for an engine cylinder per power stroke is calculated using the following equation:

$$m_e = \frac{P \cdot b_e \cdot 33.33}{n \cdot z} \text{ [mg/stroke]}$$

where:
P engine power in kilowatts
b_e engine specific fuel consumption in g/kWh
n engine speed in rpm
z number of engine cylinders

The corresponding fuel volume (injected fuel quantity), Q_H, in mm³/stroke or mm³/injection cycle is then:

$$Q_H = \frac{P \cdot b_e \cdot 1{,}000}{30 \cdot n \cdot z \cdot \rho} \text{ [mm}^3\text{/stroke]}$$

Fuel density, ρ, in g/cm³ is temperature-dependent.

Engine power output at an assumed constant level of efficiency ($\eta \sim 1/b_e$) is directly proportional to the injected fuel quantity.

The fuel mass injected by the fuel-injection system depends on the following variables:
- The fuel-metering cross-section of the injection nozzle
- The injection duration
- The variation over time of the difference between the injection pressure and the pressure in the combustion chamber
- The density of the fuel

Diesel fuel is compressible, i.e it is compressed at high pressures. This increases the injected fuel quantity. The deviation between the setpoint quantity in the program map and the actual quantity impacts on performance and pollutant emissions. In high-precision fuel-injection systems controlled by electronic diesel control, the required injected fuel quantity can be metered with a high degree of accuracy.

Injection duration

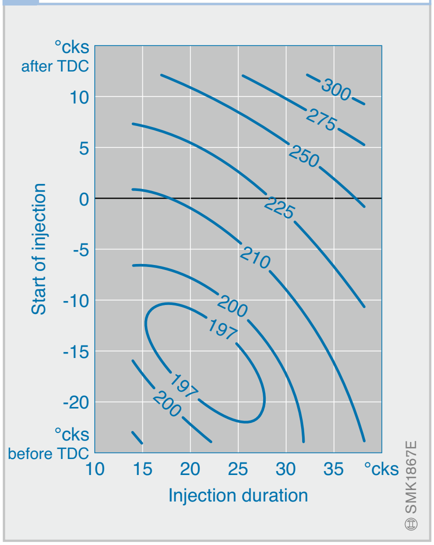
One of the main parameters of the rate-of-discharge curve is injection duration. During this period, the injection nozzle is open, and fuel flows into the combustion chamber. This parameter is specified in degrees of crankshaft or camshaft angle, or in milliseconds. Different diesel combustion processes require different injection durations, as illustrated by the following examples (approximate figures at rated output):
- Passenger-car direct-injection (DI) engine approx. 32...38° crankshaft angle
- Indirect-injection car engines: 35...40° crankshaft angle
- Direct-injection commercial-vehicle engines: 25...36° crankshaft angle

A crankshaft angle of 30° during injection duration is equivalent to a camshaft angle of 15°. This results in an injection pump speed [1]) of 2,000 rpm, equivalent to an injection duration of 1.25 ms.

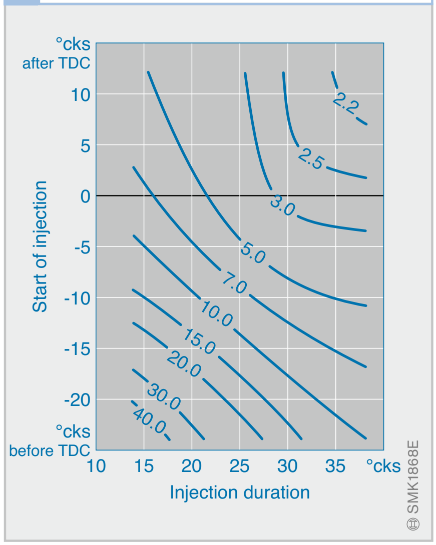
In order to minimize fuel consumption and emissions, the injection duration must be defined as a factor of the operating point and start of injection (Figs. 6 through 9).

[1]) Equivalent to half the engine speed on four-stroke engines

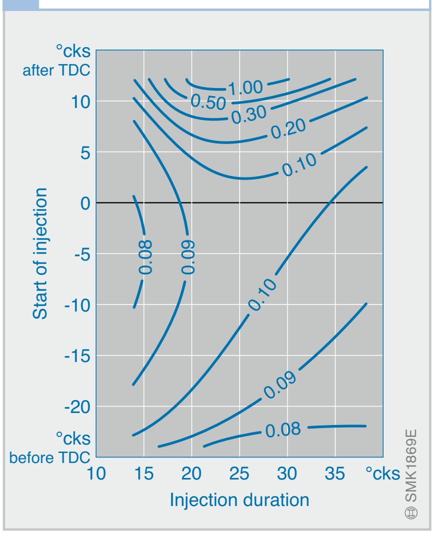
6 Specific fuel consumption b_e in g/kWh versus start of injection and injection duration



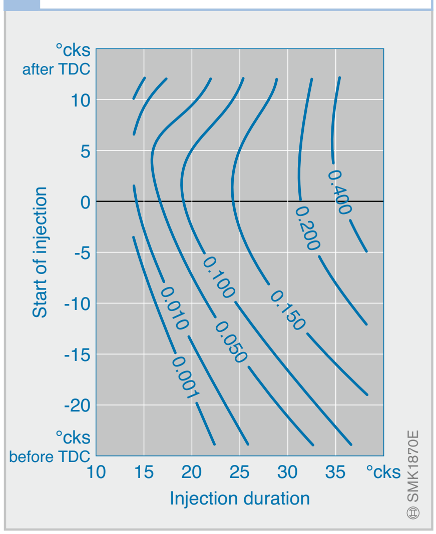
7 Specific nitrogen oxide (NO_x) emission in g/kWh versus start of injection and injection duration



8 Specific emission of unburned hydrocarbons (HC) in g/kWh versus start of injection and injection duration



9 Specific soot emission in g/kWh versus start of injection and injection duration



Figs. 6 to 9
Engine:
Six-cylinder commercial-vehicle diesel engine with common-rail fuel injection
Operating conditions:
n = 1,400 rpm,
50% full load.

The injection duration is varied in this example by changing the injection pressure to such an extent that a constant injected fuel quantity results for each injection event.

Rate-of-discharge curve

The rate-of-discharge curve describes the fuel-mass flow plotted against time when injected into the combustion chamber during the injection duration.

Rate-of-discharge curve on cam-controlled fuel-injection systems

On cam-controlled fuel-injection systems, pressure is built up continuously throughout the injection process by the fuel-injection pump. Thus, the speed of the pump has a direct impact on fuel delivery rate and, consequently, on injection pressure.

Port-controlled distributor and in-line fuel-injection pumps do not permit any pre-injection. With two-spring nozzle-and-holder assemblies, however, the injection rate can be reduced at the start of injection to improve combustion noise.

Pre-injection is also possible with solenoid-valve controlled distributor injection pumps. Unit Injector Systems (UIS) for passenger cars are equipped with hydro-mechanical pre-injection, but its control is only limited in time.

Pressure generation and delivery of the injected fuel quantity are interlinked by the cam and the injection pump in cam-controlled systems. This has the following impacts on injection characteristics:

- Injection pressure rises as engine speed and injected fuel quantity increase, and until maximum pressure is reached (Fig. 10).
- Injection pressure rises at the start of injection, but drops back to nozzle-closing pressure before the end of injection (starting at end of delivery).

The consequences of this are as follows:

- Small injected fuel quantities are injected at lower pressure.
- The rate-of-discharge curve is approximately triangular in shape.

This triangular curve promotes combustion in part-load and at low engine speeds since it achieves a shallower rise, and thus quieter combustion; however, this curve is less beneficial at full-load as a square curve achieves better air efficiency.

On indirect-injection engines (engines with prechamber or whirl chambers), throttling-pintle nozzles are used to produce a single jet of fuel and define the rate-of-discharge curve. This type of injection nozzle controls the outlet cross-section as a function of needle lift. It produces a gradual increase in pressure and, consequently, "quiet combustion".

10 Injection-pressure curve for conventional fuel injection

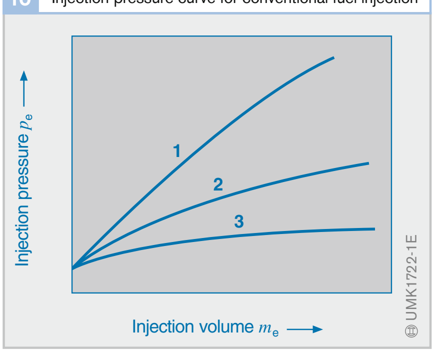


Fig. 10
1 High engine speeds
2 Medium engine speeds
3 Low engine speeds

11 Injection pattern of common-rail injection system

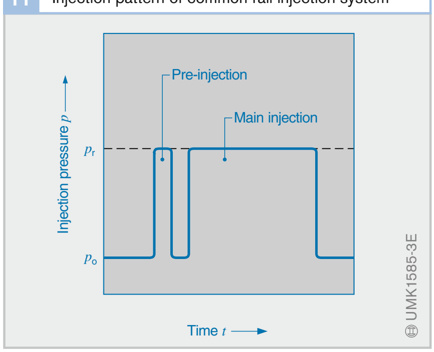


Fig. 11
p_r Fuel-rail pressure
p_o Nozzle-opening pressure

Rate-of-discharge curve in the common-rail system

A high-pressure pump generates the fuel-rail pressure independently of the injection cycle. Injection pressure during the injection process is virtually constant (Fig. 11). At a given system pressure, the injected fuel quantity is proportional to the length of time the injector is open, and it is independent of engine or pump speed (time-based injection).

This results in an almost square rate-of-discharge curve which intensifies with short injection durations and the almost constant, high spray velocities at full-load, thus permitting higher specific power outputs.

However, this is not beneficial to combustion noise since a large quantity of fuel is injected during ignition lag because of the high injection rate at the start of injection. This leads to a high pressure rise during premixed combustion. As it is possible to exclude up to two pre-injection events, the combustion chamber can be preconditioned. This shortens ignition lag and achieves the lowest possible noise emissions.

Since the electronic control unit triggers the injectors, start of injection, injection duration, and injection pressure are freely definable for the various engine operating points in an engine application. They are controlled by Electronic Diesel Control (EDC). EDC balances out injected-fuel-quantity spread in individual injectors by means of injector delivery compensation (IMA).

Modern piezoelectric common-rail fuel-injection systems permit several pre-injection and secondary injection events. In fact, up to five injection events are possible during a power cycle.

12 Injection patterns

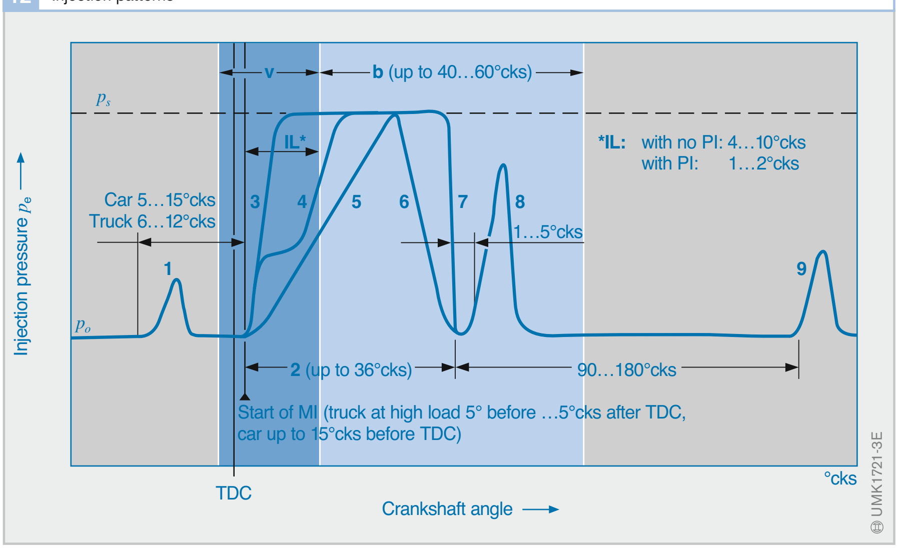


Fig. 12
Adjustments aimed at low NO_x levels require starts of injection close to TDC.
The fuel delivery point is significantly in advance of the start: injection lag is dependent on the fuel-injection system

1 Pre-injection
2 Main injection
3 Steep pressure gradient (common-rail system)
4 "Boot-shaped" pressure rise (UPS with 2-stage opening solenoid-valve needle (CCRS). Dual-spring nozzle holders can achieve a boot-shaped curve of the needle lift (not pressure curve!).
5 Gradual pressure gradient (conventional fuel injection)
6 Flat pressure drop (in-line and distributor injection pumps)
7 Steep pressure drop (UIS, UPS, slightly less steep with common rail)
8 Advanced secondary injection
9 Retarded post-injection

p_s Peak pressure
p_o Nozzle-opening pressure
b Duration of combustion for main injection phase
v Duration of combustion for pre-injection phase
IL Ignition lag of main injection

Injection functions

Depending on the application for which the engine is intended, the following injection functions are required (Fig. 12):

- *Pre-injection* (1) reduces combustion noise and NO_x emissions, in particular on DI engines.
- *Positive-pressure gradient* during the main injection event (3) reduces NO_x emissions on engines without exhaust-gas recirculation.
- *Two-stage pressure gradient (4)* during the main injection event reduces NO_x and soot emissions on engines without exhaust-gas recirculation.
- *Constant high pressure* during the main injection event (3, 7) reduces soot emissions when operating the engine with exhaust-gas recirculation.
- *Advanced secondary injection* (8) reduces soot emissions.
- *Retarded secondary injection* (9).

Pre-injection

The pressure and temperature levels in the cylinder at the point of main injection rise if a small fuel quantity (approx. 1 mg) is burned during the compression phase. This shortens the ignition lag of the main injection event and has a positive impact on combustion noise, since the proportion of fuel in the premixed combustion process decreases. At the same time the quantity of diffuse fuel combusted increases. This increases soot and NO_x emissions, also due to the higher temperature prevailing in the cylinder.

On the other hand, the higher combustion-chamber temperatures are favorable mainly at cold start and in the low load range in order to stabilize combustion and reduce HC and CO emissions.

A good compromise between combustion noise and NO_x emissions is obtainable by adapting the time interval between pre-injection and main injection dependent on the operating point, and metering the pre-injected fuel quantity.

13 Effect of pre-injection on combustion-pressure pattern

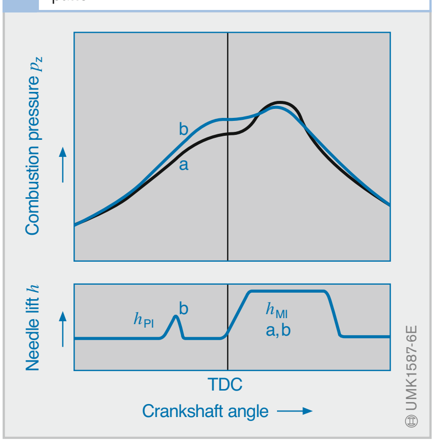


Fig. 13
a Without pre-injection
b With pre-injection

h_{PI} Needle lift during pre-injection
h_{MI} Needle lift during main injection

Retarded secondary injection

With retarded secondary injection, fuel is not combusted, but is evaporated by residual heat in the exhaust gas. The secondary-injection phase follows the main-injection phase during the expansion or exhaust stroke at a point up to 200° crankshaft angle after TDC. It injects a precisely metered quantity of fuel into the exhaust gas. The resulting mixture of fuel and exhaust gas is expelled through the exhaust ports into the exhaust-gas system during the exhaust stroke.

Retarded secondary injection is mainly used to supply hydrocarbons which also cause an increase in exhaust-gas temperature by oxidation in an oxidation-type catalytic converter. This measure is used to regenerate downstream exhaust-gas treatment systems, such as particulate filters or NO_x accumulator-type catalytic converters.

Since retarded secondary injection may cause thinning of the engine oil by the diesel fuel, it needs clarification with the engine manufacturer.

Advanced secondary injection

On the common-rail system, secondary injection can occur directly after main injection while combustion is still taking place. In this way, soot particles are reburned, and soot emissions can be reduced by 20 to 70%.

Timing characteristics of fuel-injection systems

Figure 14 presents an example of a radial-piston distributor pump (VP44). The cam on the cam ring starts delivery, and fuel then exits from the nozzle. It shows that pressure and injection patterns vary greatly between the pump and the nozzle, and are determined by the characteristics of the components that control injection (cam, pump, high-pressure valve, fuel line, and nozzle). For this reason, the fuel-injection system must be precisely matched to the engine.

The characteristics are similar for all fuel-injection systems in which pressure is generated by a pump plunger (in-line injection pumps, unit injectors, and unit pumps).

Detrimental volume in conventional injection systems

The term “detrimental volume” refers to the volume of fuel on the high-pressure side of the fuel-injection system. This is made up of the high-pressure side of the fuel-injection pump, the high-pressure fuel lines, and the volume of the nozzle-and-holder assembly. Every time fuel is injected, the detrimental volume is pressurized and depressurized. As a result, compression losses occur, thus retarding injection lag. The fuel volume inside of the pipes is compressed by the dynamic processes generated by the pressure wave.

The greater the detrimental volume, the poorer the hydraulic efficiency of the fuel-injection system. A major consideration when developing a fuel-injection system is, therefore, to minimize detrimental volume as much as possible. The unit injector system has the smallest detrimental volume.

In order to guarantee uniform control of the engine, the detrimental volume must be equal for all cylinders.

14 Chain of interaction from cam pitch to injection pattern plotted against camshaft angle

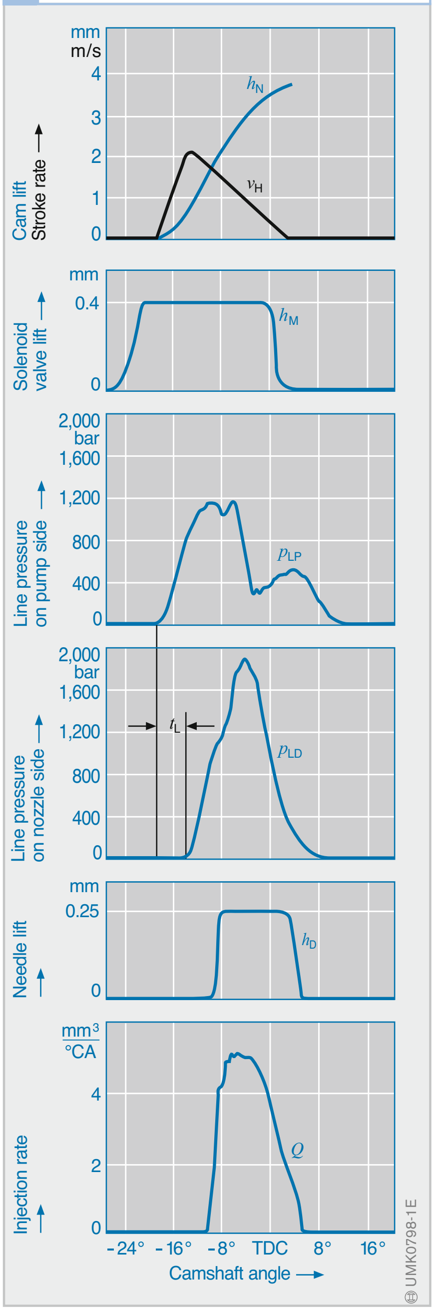


Fig. 14
Example of radial-piston distributor injection pump (VP 44) at full load without pre-injection

t_L Time for fuel to pass through line

Injection pressure

The process of fuel injection uses pressure in the fuel system to induce the flow of fuel through the injector jets. A high fuel-system pressure results in a high rate of fuel outflow at the injection nozzle. Fuel atomization is caused by the collision of the turbulent jet of fuel with the air inside of the combustion chamber. Therefore, the higher the relative velocity between fuel and air, and the higher the density of the air, the more finely the fuel is atomized. The injection pressure at the nozzle may be higher than in the fuel-injection pump because of the length of the high-pressure fuel line, whose length is matched to the reflected pressure wave.

Direct-injection (DI) engines

On diesel engines with direct injection, the velocity of the air inside of the combustion chamber is relatively slow since it only moves as a result of its mass moment of inertia (i.e. the air "attempts" to maintain the velocity at which it enters the cylinder; this causes whirl). The piston stroke intensifies whirl in the cylinder since the restricted flow forces the air into the piston recess, and thus into a smaller diameter. In general, however, air motion is less and in indirect-injection engines.

15 Influence of injection pressure and start of injection on fuel consumption, soot and nitrogen-oxide emissions

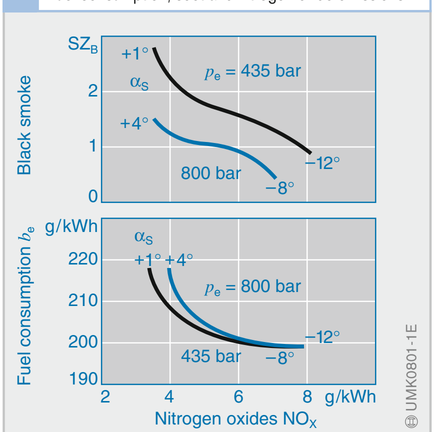


Fig. 15
Direct-injection engine, engine speed 1,200 rpm, mean pressure 16.2 bar

p_e Injection pressure
α_S Start of injection after TDC
SZ_B Black smoke number

The fuel must be injected at high pressure due to low air flow. Modern direct-injection systems now generate full-load peak pressures of 1,000...2,050 bar for car engines, and 1,000...2,200 bar for commercial vehicles. However, peak pressure is available only at higher engine speeds – except on the common-rail system.

A decisive factor to obtain an ideal torque curve with low-smoke operation (i.e. with low particulate emission) is a relatively high injection pressure adapted to the combustion process at low full-load engine speeds. Since the air density in the cylinder is relatively low at low engine speeds, injection pressure must be limited to avoid depositing fuel on the cylinder wall. Above about 2,000 rpm, the maximum charge-air pressure becomes available, and injection pressure can rise to maximum.

To obtain ideal engine efficiency, fuel must be injected within a specific, engine-speed-dependent angle window on either side of TDC. At high engine speeds (rated output), therefore, high injection pressures are required to shorten the injection duration.

Engines with indirect injection (IDI)

On diesel engines with divided combustion chambers, rising combustion pressure expels the charge out of the prechamber or whirl chamber into the main combustion chamber. This process runs at high air velocities in the whirl chamber, in the connecting passage between the whirl chamber, and the main combustion chamber.

Nozzle and nozzle holder designs

Secondary injection

Unintended secondary injection has a particularly undesirable effect on exhaust-gas quality. Secondary injection occurs when the injection nozzle shortly re-opens after closing and allows poorly conditioned fuel to be injected into the cylinder at a late stage in the combustion process. This fuel is not completely burned, or may not be burned at all, with the result that it is released into the exhaust gas as unburned hydrocarbons. This undesirable effect can be prevented by rapidly closing nozzle-and-holder assemblies, at sufficiently high closing pressure and low static pressure in the supply line.

Dead volume

Dead volume in the injection nozzle on the cylinder side of the needle-seal seats has a similar effect to secondary injection. The fuel accumulated in such a volume runs into the combustion chamber on completion of combustion, and partly escapes into the exhaust pipe. This fuel component similarly increases the level of unburned hydrocarbons in the exhaust gas (Fig. 1). Sac-less (vco) nozzles, in which the injection orifices are drilled into the needle-seal seats, have the smallest dead volume.

Injection direction

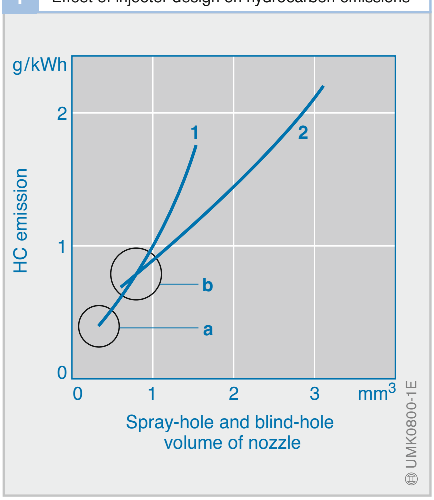
Direct-injection (DI) engines

Diesel engines with direct injection generally have hole-type nozzles with between 4 and 10 injection orifices (most commonly 6 to 8 injection orifices) arranged as centrally as possible. The injection direction is very precisely matched to the combustion chamber. Divergences of the order of only 2 degrees from the optimum injection direction lead to a detectable increase in soot emission and fuel consumption.

Engines with indirect injection (IDI)

Indirect-injection engines use pintle nozzles with only a single injection jet. The nozzle injects the fuel into the precombustion or whirl chamber in such a way that the glow plug is just within the injection jet. The injection direction is matched precisely to the combustion chamber. Any deviations in injection direction result in poorer utilization of combustion air and, therefore, to an increase in soot and hydrocarbon emissions.

1 Effect of injector design on hydrocarbon emissions



2 Nozzle cones

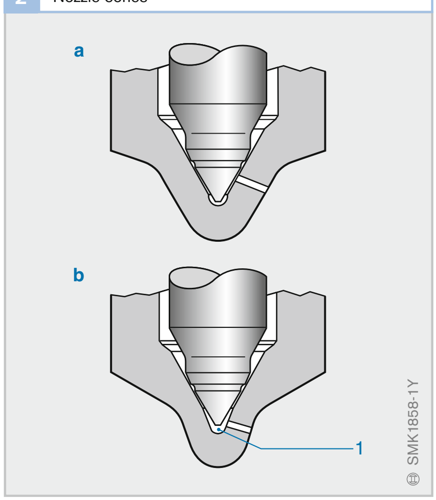


Fig. 1
a Sac-less (vco) nozzle
b Injector with micro-blind hole

1 Engine with 1 l/cylinder
2 Engine with 2 l/cylinder

Fig. 2
a Sac-less (vco) nozzle
b Injector with micro-blind hole

1 Dead volume

Basics of the gasoline (SI) engine

The gasoline or spark-ignition (SI) internal-combustion engine uses the Otto cycle [1]) and externally supplied ignition. It burns an air/fuel mixture and in the process converts the chemical energy in the fuel into kinetic energy.

For many years, the carburetor was responsible for providing an air/fuel mixture in the intake manifold which was then drawn into the cylinder by the downgoing piston.

The breakthrough of gasoline fuel injection, which permits extremely precise metering of the fuel, was the result of the legislation governing exhaust-gas emission limits. Similar to the carburetor process, with manifold fuel injection the air/fuel mixture is formed in the intake manifold.

Even more advantages resulted from the development of gasoline direct injection, in particular with regard to fuel economy and increases in power output. Direct injection injects the fuel directly into the engine cylinder at exactly the right instant in time.

[1]) Named after Nikolaus Otto (1832–1891) who presented the first gas engine with compression using the 4-stroke principle at the Paris World Fair in 1878.

Method of operation

The combustion of the air/fuel mixture causes the piston (Fig. 1, Pos. 8) to perform a reciprocating movement in the cylinder (9). The name reciprocating-piston engine, or better still reciprocating engine, stems from this principle of functioning.

The conrod (10) converts the piston's reciprocating movement into a crankshaft (11) rotational movement which is maintained by a flywheel at the end of the crankshaft. Crankshaft speed is also referred to as engine speed or engine rpm.

Four-stroke principle

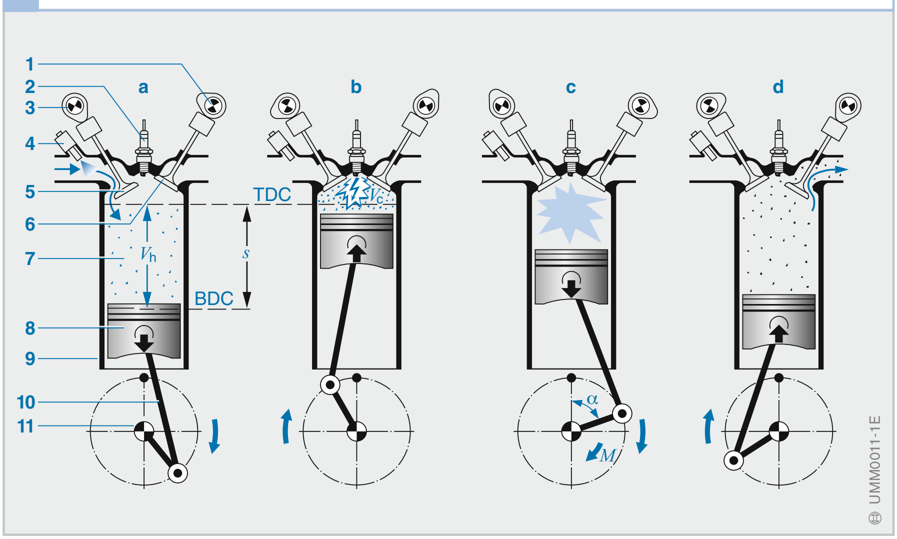
Today, the majority of the internal-combustion engines used as vehicle power plants are of the four-stroke type. The four-stroke principle employs gas-exchange valves (5 and 6) to control the exhaust-and-refill cycle. These valves open and close the cylinder's intake and exhaust passages, and in the process control the supply of fresh air/fuel mixture and the forcing out of the burnt exhaust gases.

Fig. 1
- a Induction stroke
- b Compression stroke
- c Power (combustion) stroke
- d Exhaust stroke

1. Exhaust camshaft
2. Spark plug
3. Intake camshaft
4. Injector
5. Intake valve
6. Exhaust valve
7. Combustion chamber
8. Piston
9. Cylinder
10. Conrod
11. Crankshaft

- M Torque
- α Crankshaft angle
- s Piston stroke
- V_h Piston displacement
- V_c Compression volume

1 Complete working cycle of the 4-stroke spark-ignition (SI) gasoline engine (example shows a manifold-injection engine with separate intake and exhaust camshafts)



K. Reif (Ed.), *Fundamentals of Automotive and Engine Technology*,
DOI 10.1007/978-3-658-03972-1_6, © Springer Fachmedien Wiesbaden 2014

1st stroke: Induction

Referred to Top Dead Center (TDC), the piston is moving downwards and increases the volume of the combustion chamber (7) so that fresh air (gasoline direct injection) or fresh air/fuel mixture (manifold injection) is drawn into the combustion chamber past the opened intake valve (5).

The combustion chamber reaches maximum volume (V_h+V_c) at Bottom Dead Center (BDC).

2nd stroke: Compression

The gas-exchange valves are closed, and the piston is moving upwards in the cylinder. In doing so it reduces the combustion-chamber volume and compresses the air/fuel mixture. On manifold-injection engines the air/fuel mixture has already entered the combustion chamber at the end of the induction stroke. With a direct-injection engine on the other hand, depending upon the operating mode, the fuel is first injected towards the end of the compression stroke.

At Top Dead Center (TDC) the combustion-chamber volume is at minimum (compression volume V_c).

3rd stroke: Power (or combustion)

Before the piston reaches Top Dead Center (TDC), the spark plug (2) initiates the combustion of the air/fuel mixture at a given ignition point (ignition angle). This form of ignition is known as externally supplied ignition. The piston has already passed its TDC point before the mixture has combusted completely.

The gas-exchange valves remain closed and the combustion heat increases the pressure in the cylinder to such an extent that the piston is forced downward.

4th stroke: Exhaust

The exhaust valve (6) opens shortly before Bottom Dead Center (BDC). The hot (exhaust) gases are under high pressure and leave the cylinder through the exhaust valve. The remaining exhaust gas is forced out by the upwards-moving piston.

A new operating cycle starts again with the induction stroke after every two revolutions of the crankshaft.

Valve timing

The gas-exchange valves are opened and closed by the cams on the intake and exhaust camshafts (3 and 1 respectively).
On engines with only 1 camshaft, a lever mechanism transfers the cam lift to the gas-exchange valves.

The valve timing defines the opening and closing times of the gas-exchange valves. Since it is referred to the crankshaft position, timing is given in "degrees crankshaft". Gas flow and gas-column vibration effects are applied to improve the filling of the combustion chamber with air/fuel mixture and to remove the exhaust gases. This is the reason for the valve opening and closing times overlapping in a given crankshaft angular-position range.

The camshaft is driven from the crankshaft through a toothed belt (or a chain or gear pair). On 4-stroke engines, a complete working cycle takes two rotations of the crankshaft. In other words, the camshaft only turns at half crankshaft speed, so that the step-down ratio between crankshaft and camshaft is 2:1.

2 Valve timing diagram for a four-stroke gasoline-engine

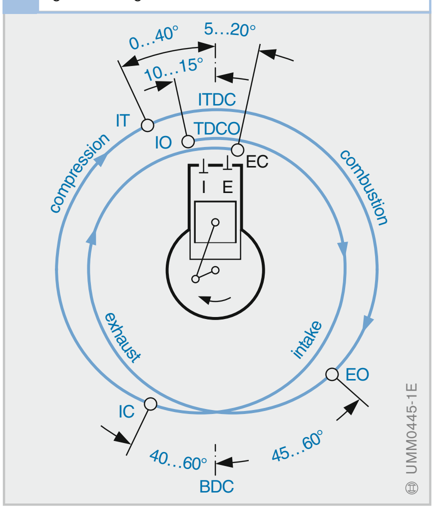


Fig. 2
- I Intake valve
- IO Intake valve opens
- IC Intake valve closes
- E Exhaust valve
- EO Exhaust valve opens
- EC Exhaust valve closes
- TDC Top Dead Center
- TDCO Overlap at TDC
- ITDC Ignition at TDC
- BDC Bottom Dead Center
- IT Ignition point

Compression

The difference between the maximum piston displacement V_h and the compression volume V_c is the compression ratio

$$\varepsilon = (V_h + V_c)/V_c.$$

The engine's compression ratio is a vital factor in determining

- Torque generation
- Power generation
- Fuel economy and
- Emissions of harmful pollutants

The gasoline-engine's compression ratio ε varies according to design configuration and the selected form of fuel injection (manifold or direct injection ε = 7...13). Extreme compression ratios of the kind employed in diesel powerplants (ε = 14...24) are not suitable for use in gasoline engines. Because the knock resistance of the fuel is limited, the extreme compression pressures and the high combustion-chamber temperatures resulting from such compression ratios must be avoided in order to prevent spontaneous and uncontrolled detonation of the air/fuel mixture. The resulting knock can damage the engine.

Air/fuel ratio

Complete combustion of the air/fuel mixture relies on a stoichiometric mixture ratio. A stoichiometric ratio is defined as 14.7 kg of air for 1 kg of fuel, that is, a 14.7 to 1 mixture ratio.

The air/fuel ratio λ (lambda) indicates the extent to which the instantaneous monitored air/fuel ratio deviates from the theoretical ideal:

$$\lambda = \frac{\text{induction air mass}}{\text{theoretical air requirement}}$$

The lambda factor for a stoichiometric ratio is λ 1.0. λ is also referred to as the excess-air factor.

Richer fuel mixtures result in λ figures of less than 1. Leaning out the fuel produces mixtures with excess air: λ then exceeds 1. Beyond a certain point the mixture encounters the lean-burn limit, beyond which ignition is no longer possible. The excess-air factor has a decisive effect on the specific fuel consumption (Fig. 3) and untreated pollutant emissions (Fig. 4).

Induction-mixture distribution in the combustion chamber

Homogeneous distribution

The induction systems on engines with manifold injection distribute a homogeneous air/fuel mixture throughout the combustion chamber. The entire induction charge has a single excess-air factor λ (Fig. 5a). Lean-burn engines, which operate on excess air under

3 Influence of the excess-air factor λ on the power P and on the specific fuel consumption b_e under conditions of homogeneous air/fuel-mixture distribution

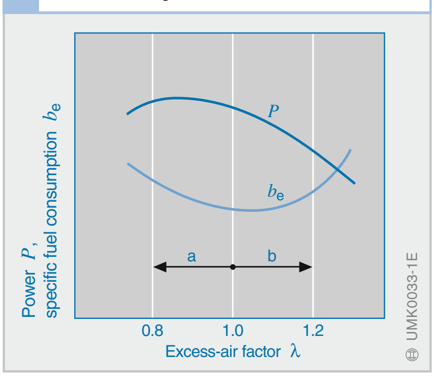
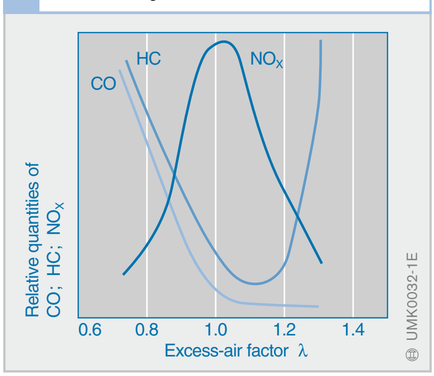


Fig. 3
a Rich air/fuel mixture (air deficiency)
b Lean air/fuel mixture (excess air)

4 Effect of the excess-air factor λ on the pollutant composition of untreated exhaust gas under conditions of homogeneous air/fuel-mixture distribution



specific operating conditions, also rely on homogeneous mixture distribution.

Stratified-charge concept

A combustible mixture cloud with $\lambda \approx 1$ surrounds the tip of the spark plug at the instant ignition is triggered. At this point the remainder of the combustion chamber contains either non-combustible gas with no fuel, or an extremely lean air/fuel charge. The corresponding strategy, in which the ignitable mixture cloud is present only in one portion of the combustion chamber, is the stratified-charge concept (Fig. 5b). With this concept, the overall mixture – meaning the average mixture ratio within the entire combustion chamber – is extremely lean (up to $\lambda \approx 10$). This type of lean operation fosters extremely high levels of fuel economy.

5 Induction-mixture distribution in the combustion chamber

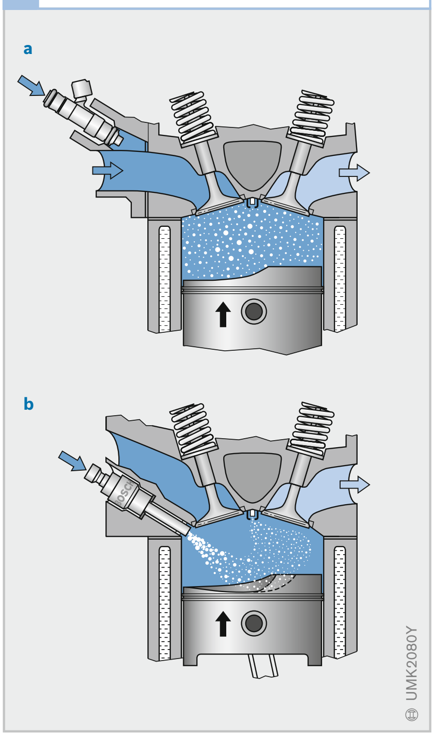


Fig. 5
a Homogeneous mixture distribution
b Stratified charge

Efficient implementation of the stratified-charge concept is impossible without direct fuel injection, as the entire induction strategy depends on the ability to inject fuel directly into the combustion chamber just before ignition.

Ignition and flame propagation

The spark plug ignites the air/fuel mixture by discharging a spark across a gap. The extent to which ignition will result in reliable flame propagation and secure combustion depends in large part on the air/fuel mixture λ, which should be in a range extending from λ = 0.75...1.3. Suitable flow patterns in the area immediately adjacent to the spark-plug electrodes can be employed to ignite mixtures as lean as $\lambda \leq 1.7$.

The initial ignition event is followed by formation of a flame-front. The flame front's propagation rate rises as a function of combustion pressure before dropping off again toward the end of the combustion process. The mean flame front propagation rate is on the order of 15...25 m/s.

The flame front's propagation rate is the combination of mixture transport and combustion rates, and one of its defining factors is the air/fuel ratio λ. The combustion rate peaks at slightly rich mixtures on the order of λ = 0.8...0.9. In this range it is possible to approach the conditions coinciding with an ideal constant-volume combustion process (refer to section on "Engine efficiency"). Rapid combustion rates provide highly satisfactory full-throttle, full-load performance at high engine speeds.

Good thermodynamic efficiency is produced by the high combustion temperatures achieved with air/fuel mixtures of λ = 1.05...1.1. However, high combustion temperatures and lean mixtures also promote generation of nitrous oxides (NO_X), which are subject to strict limitations under official emissions standards.

Cylinder charge

An air/fuel mixture is required for the combustion process in the cylinder. The engine draws in air through the intake manifolds (Fig. 1, Pos. 14), the throttle valve (13) ensuring that the air quantity is metered. The fuel is metered through fuel injectors. Furthermore, usually part of the burnt mixture (exhaust gas) from the last combustion is retained as residual gas (9) in the cylinder or exhaust gas is returned specifically to increase the residual-gas content in the cylinder (4).

Components of the cylinder charge

The gas mixture trapped in the combustion chamber when the intake valve closes is referred to as the cylinder charge. This is comprised of the fresh gas and the residual gas.

The term "relative air charge *rac*" has been introduced in order to have a quantity which is independent of the engine's displacement. It describes the air content in the cylinder and is defined as the ratio of the current air quantity in the cylinder to the air quantity that would be contained in the engine displacement under standard conditions (p_0 = 1013 hPa, T_0 = 273 K). Accordingly, there is a relative fuel quantity *rfq*; this is defined in such a way that identical values for *rac* and *rfq* result in $\lambda = 1$, i.e., $\lambda = rac/rfq$, or with specified λ : $rfq = rac/\lambda$.

1 Cylinder charge in a gasoline engine

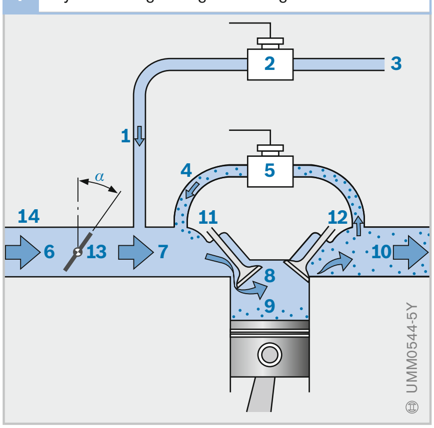


Fig. 1
1 Air and fuel vapor (from evaporative-emissions control system)
2 Canister-purge valve with variable valve-opening cross-section
3 Connection to evaporative-emissions control system
4 Returned exhaust gas
5 Exhaust-Gas Recirculation valve (EGR valve) with variable valve-opening cross-section
6 Air-mass flow (ambient pressure p_a)
7 Air-mass flow (manifold pressure p_m)
8 Fresh-gas charge (combustion-chamber pressure p_c)
9 Residual-gas charge (combustion-chamber pressure p_c)
10 Exhaust gas (exhaust-gas back pressure p_e)
11 Intake valve
12 Exhaust valve
13 Throttle valve
14 Intake manifold

α Throttle-valve angle

Fresh gas

The freshly introduced gas mixture in the cylinder is comprised of the fresh air drawn in and the fuel entrained with it. In a manifold-injection engine, all the fuel has already been mixed with the fresh air upstream of the intake valve. On direct-injection systems, on the other hand, the fuel is injected directly into the combustion chamber.

The majority of the fresh air enters the cylinder with the air-mass flow (Fig. 1, Pos. 6, 7) via the throttle valve (13). Additional fresh gas, comprising fresh air and fuel vapor, is directed to the cylinder via the evaporative-emissions control system (3, 2).

For homogeneous operation at $\lambda \leq 1$, the air in the cylinder directed via the throttle valve after the intake valve (11) has closed is the decisive quantity for the work at the piston during the combustion stroke and therefore for the engine's output torque. In this case, the air charge corresponds to the torque and the engine load. Here, changing the throttle-valve angle only indirectly leads to a change in the air charge. First of all, the pressure in the intake manifold must rise so that a greater air mass flows into the cylinder via the intake valves. Fuel can, on the other hand, be injected more contemporaneously with the combustion process and metered precisely to the individual cylinder. Therefore the injected fuel quantity is dependent on the current air quantity, and the gasoline engine is an air-directed system in "conventional" homogeneous mode at $\lambda \leq 1$.

During lean-burn operation (stratified charge), however, the torque (engine load) – on account of the excess air – is a direct product of the injected fuel mass. The air mass can thus differ for the same torque. The gasoline engine is therefore fuel-directed during lean-burn operation.

Almost always, measures aimed at increasing the engine's maximum torque and maximum output power necessitate an increase in the maximum possible fresh-gas charge. This can be achieved by increasing the engine displacement but also by supercharging (see section entitled "Supercharging").

Residual gas

The residual-gas share of the cylinder charge comprises that portion of the cylinder charge which has already taken part in the combustion process. In principle, one differentiates between internal and external residual gas. Internal residual gas is the exhaust gas which remains in the upper clearance volume of the cylinder after combustion or which, while the intake and exhaust valves are simultaneously open (valve overlap, see section entitled "Gas exchange"), is drawn from the exhaust port back into the intake manifold (internal exhaust-gas recirculation).

External residual gas is exhaust gas which is introduced via an exhaust-gas recirculation valve (Fig. 1, Pos. 4, 5) into the intake manifold (external exhaust-gas recirculation).

The residual gas is made up of inert gas [1]) and – in the event of excess air, i.e., during lean-burn operation – of unburnt air. The amount of inert gas in the residual gas is particularly important. This no longer contains any oxygen and therefore does not participate in combustion during the following power cycle. However, it does delay ignition and slows down the course of combustion, which results in slightly lower efficiency but also in lower peak pressures and temperatures. In this way, a specifically used amount of residual gas can reduce the emission of nitrogen oxides (NO_X). This then is the benefit of inert gas in lean-burn operation in that the three-way catalytic converter is unable to reduce the nitrogen oxides in the event of excess air.

In homogeneous engine mode, the fresh-gas charge displaced by the residual gas (consisting in this case of inert gas only) is compensated by means of a greater opening of the throttle valve. With a constant fresh-gas charge, this increases the intake-manifold pressure, therefore reduces the throttling losses (see section entitled "Gas exchange"), and in all results in reduced fuel consumption.

Gas exchange

The process of replacing the consumed cylinder charge (exhaust gas, also referred to in the above as residual gas) with fresh gas is known as gas exchange or the charge cycle. It is controlled by the opening and closing of the intake and exhaust valves in combination with the piston stroke. The shape and position of the camshaft cams determine the progression of the valve lift and thereby influence the cylinder charge.

The opening and closing times of the valves are called valve timing and the maximum distance a valve is lifted from its seat is known as the valve lift or valve stroke. The characteristic variables are Exhaust Opens (EO), Exhaust Closes (EC), Intake Opens (IO), Intake Closes (IC) and the valve lift. There are engines with fixed and others with variable timing and valve lifts (see chapter entitled "Cylinder-charge control systems").

The amount of residual gas for the following power cycle can be significantly influenced by a valve overlap. During the valve overlap, intake and exhaust valves are simultaneously open for a certain amount of time, i.e., the intake valve opens before the exhaust valve closes. If in the overlap phase the pressure in the intake manifold is lower than that in the exhaust train, the residual gas flows back into the intake manifold; because the residual gas drawn back in this way is drawn in again after Exhaust Closes, this results in an increase in the residual-gas content.

[1]) Components in the combustion chamber which behave inertly, that is, do not participate in the combustion process.

In the case of supercharging, the pressure before the intake valve can also be higher during the overlap phase; in this event, the residual gas flows in the direction of the exhaust train such that it is properly cleared away ("scavenging") and it is also possible for the air to flow through into the exhaust train.

When the residual gas is successfully scavenged, its volume is then available for an increased fresh-gas charge. The scavenging effect is therefore used to increase torque in the lower speed range (up to approx. 2000 rpm), either in combination with dynamic supercharging in naturally aspirated engines or with turbocharging.

Volumetric efficiency and air consumption

The success of the gas-exchange process is measured in the variables volumetric efficiency, air consumption and retention rate. The volumetric efficiency is the ratio of the fresh-gas charge actually remaining in the cylinder to the theoretically maximum possible charge. It differs from the relative air charge in that the volumetric efficiency is referred to the external conditions at the time of measurement and not to standard conditions.

The air consumption describes the total air-mass throughput during the gas-exchange process, likewise referred to the theoretically maximum possible charge. The air consumption can also include the air mass which is transferred directly into the exhaust train during the valve overlap. The retention rate, the ratio of volumetric efficiency to air consumption, specifies the proportion of the air-mass throughput which remains in the cylinder at the end of the gas-exchange process.

The maximum volumetric efficiency for naturally aspirated engines is 0.6...0.9. It depends on the combustion-chamber shape, the opened cross-sections of the gas-exchange valves, and the valve timing.

Pumping losses

Work is expended in the form of pumping losses or gas-exchange losses in order to replace the exhaust gas with fresh gas in the gas-exchange process. These losses use up part of the mechanical work generated and therefore reduce the effective efficiency of the engine. In the intake phase, i.e., during the downward stroke of the piston, the intake-manifold pressure in throttled mode is less than the ambient pressure and in particular the pressure in the piston return chamber. The piston must work against this pressure differential (throttling losses).

A dynamic pressure occurs in the combustion chamber during the upward stroke of the piston when the burnt gas is emitted, particularly at high engine speeds and loads; the piston must expend energy in order to overcome this pressure (push-out losses).

If with gasoline direct injection stratified-charge operation is used with the throttle valve fully opened or high exhaust-gas recirculation is used in homogeneous operation ($\lambda \leq 1$), this increases the intake-manifold pressure and reduces the pressure differential above the piston. In this way, the engine's throttling losses can be reduced, which in turn improves the effective efficiency.

Supercharging

The torque which can be achieved during homogenous operation at $\lambda \leq 1$ is proportional to the fresh-gas charge. This means that maximum torque can be increased by compressing the air before it enters the cylinder (supercharging). This leads to an increase in volumetric efficiency to values above 1.

Dynamic supercharging

Supercharging can be achieved simply by taking advantage of the dynamic effects inside the intake manifold. The supercharging level depends on the intake manifold's design and on its operating point (for the most part, on engine speed, but also on cylinder charge). The possibility of changing the intake-manifold geometry while the engine is running (variable intake-manifold geometry) means that dynamic supercharging can be applied across a wide operating range to increase the maximum cylinder charge.

Mechanical supercharging
The intake-air density can be further increased by compressors which are driven mechanically from the engine's crankshaft. The compressed air is forced through the intake manifold and into the engine's cylinders.

Exhaust-gas turbocharging
In contrast mechanical supercharging, the compressor of the exhaust-gas turbocharger is driven by an exhaust-gas turbine located in the exhaust-gas flow, and not by the engine's crankshaft. This enables recovery of some of the energy in the exhaust gas.

Charge recording
In a gasoline engine with homogeneous $\lambda = 1$ operation, the injected fuel quantity is dependent on the air quantity. This is necessary because after a change to the throttle-valve angle the air charge changes only gradually while the fuel quantity can be varied from injection to injection.

For this reason, the current available air charge must be determined for each combustion in the engine-management system (charge recording). There are essentially three systems which can be used to record the charge:

- A hot-film air-mass meter (HFM) measures the air-mass flow into the intake manifold.
- A model is used to calculate the air-mass flow from the temperature before the throttle valve, the pressure before and after the throttle valve, and the throttle-valve angle (throttle-valve model, α/n system [1])).
- A model is used to calculate the charge drawn in by the cylinder from the engine speed *(n)*, the pressure *(p)* in the intake manifold (i.e., before the intake valve), the temperature in the intake passage and further additional information (e.g., camshaft/valve-lift adjustment, intake-manifold changeover, position of the swirl control valve) (p/n system). Sophisticated models may be necessary, depending on the complexity of the engine, particularly with regard to the variabilities of the valve gear.

Because only the mass flow passing into the intake manifold can be determined with a hot-film air-mass meter or a throttle-valve model, both these systems only provide a cylinder-charge value during stationary engine operation. Stationary means at constant intake-manifold pressure; because then the mass flows flowing into the intake manifold and off into the engine are identical.

In the event of a sudden load change (change in the throttle-valve angle), the inflowing mass flow changes spontaneously, while the off-flowing mass flow and with it the cylinder charge only change if the intake-manifold pressure has increased or reduced. The accumulator behavior of the intake manifold must therefore also be imitated *(intake-manifold model)*.

[1]) The designation α/n system is historically conditioned since originally the pressure after the throttle valve was not taken into account and the mass flow was stored in a program map covering throttle-valve angle and engine speed. This simplified approach is sometimes still used today.

Torque and power

Torques at the drivetrain

The power P delivered by a gasoline engine is defined by the available clutch torque M and the engine speed n. The clutch torque is the torque developed by the combustion process less friction torque (friction losses in the engine), pumping losses, and the torque needed to drive the auxiliary equipment (Fig. 1). The drive torque is derived from the clutch torque plus the losses arising at the clutch and transmission.

The combustion torque is generated in the power cycle and is determined in engines with manifold injection by the following variables:

- The air mass which is available for combustion when the intake valves close
- The fuel mass which is available at the same moment, and
- The moment in time when the ignition spark initiates the combustion of the air/fuel mixture

Direct-injection gasoline engines function at certain operating points with excess air (lean-burn operation). The cylinder thus contains air, which has no effect on the generated torque. Here, it is the fuel mass which has the most effect.

Generation of torque

The physical quantity torque M is the product of force F times lever arm s:

$$M = F \cdot s$$

The connecting rod utilizes the throw of the crankshaft to convert the piston's linear travel into rotary motion. The force with which the expanding air/fuel mixture drives the piston down the cylinder is converted into torque by the lever arm generated by the throw.

The lever arm l which is effective for the torque is the lever component vertical to the force (Fig. 2). The force and the leverage angle are parallel at Top Dead Center (TDC).

1 Torques at the drivetrain

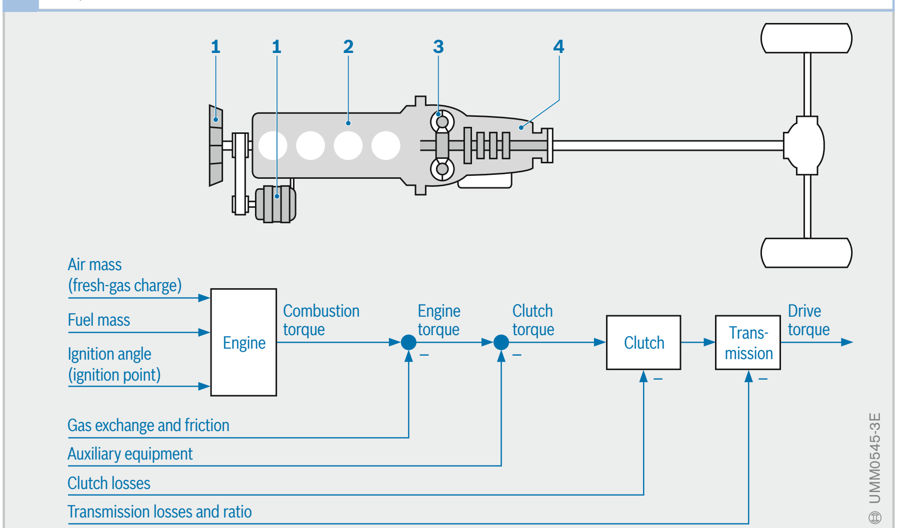


Fig. 1
1 Auxiliary equipment (A/C compressor, alternator, etc.)
2 Engine
3 Clutch
4 Transmission

This results in an effective lever arm of zero. The ignition angle must be selected in such a way as to trigger mixture ignition while the crankshaft is rotating through a phase of increasing lever arm (0...90 °crankshaft). This enables the engine to generate the maximum possible torque. The engine's design (for instance, piston displacement, combustion-chamber geometry, volumetric efficiency, charge) determines the maximum possible torque M that it can generate.

Essentially, the torque is adapted to the requirements of actual driving by adjusting the quality and quantity of the air/fuel mixture and the ignition angle. Fig. 3 shows the typical torque and power curves, plotted against engine speed, for a manifold-injection gasoline engine. As engine speed increases, full-load torque initially increases to its maximum M_{max}. At higher engine speeds, torque falls off again as the shorter opening times of the intake valves limits the cylinder charge.

Engine designers focus on attempting to obtain maximum torque at low engine speeds of around 2000 rpm. This rpm range coincides with optimal fuel economy. Engines with exhaust-gas turbochargers are able to meet these requirements.

Relationship between torque and power

The engine's power output P climbs along with increasing torque M and engine speed n. The following applies:

$$P = 2 \cdot \pi \cdot M \cdot n$$

Engine power increases until it reaches its peak value at rated speed n_{rat} with rated power P_{rat}. Owing to the substantial decrease in torque, power generation drops again at extremely high engine speeds.

A transmission to vary conversion ratios is needed to adapt the gasoline engine's torque and power curves to meet the requirements of vehicle operation.

2 Lever arm for crankshaft drive

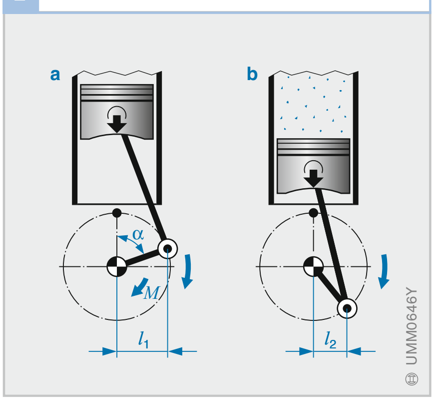


Fig. 2
Changing the effective lever arm during the power cycle
a Increasing lever arm l_1
b Decreasing lever arm l_2

3 Torque and power curves

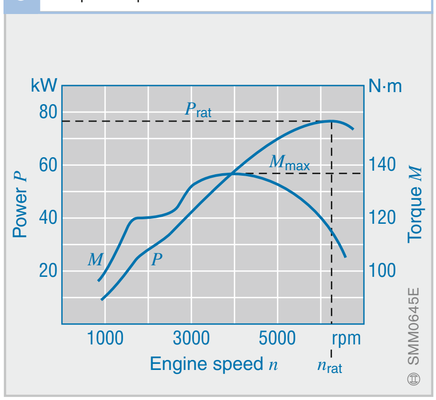


Fig. 3
Typical curves for a manifold-injection gasoline engine

Engine efficiency

Thermal efficiency

The internal-combustion engine does not convert all the energy which is chemically available in the fuel into mechanical work, and some of the added energy is lost. This means that an engine's efficiency is less than 100% (Fig. 1). Thermal efficiency is one of the important links in the engine's efficiency chain.

Pressure-volume diagram (*p-V* diagram)

The *p-V* diagram is used to display the pressure and volume conditions during a complete working cycle of the 4-stroke IC engine.

The ideal cycle
Figure 2 (curve A) shows the compression and power strokes of an ideal process as defined by the laws of Boyle/Mariotte and Gay-Lussac. The piston travels from BDC to TDC (point 1 to point 2), and the air/fuel mixture is compressed without the addition of heat (Boyle/Mariotte). Subsequently, the mixture burns accompanied by a pressure rise (point 2 to point 3) while volume remains constant (Gay-Lussac).

From TDC (point 3), the piston travels towards BDC (point 4), and the combustion-chamber volume increases. The pressure of the burnt gases drops whereby no heat is released (Boyle/Mariotte). Finally, the burnt mixture cools off again with the volume remaining constant (Gay-Lussac) until the initial status (point 1) is reached again.

The area inside the points 1 – 2 – 3 – 4 shows the work gained during a complete working cycle. The exhaust valve opens at point 4 and the gas, which is still under pressure, escapes from the cylinder. If it were possible for the gas to expand completely by the time point 5 is reached, the area described by 1 – 4 – 5 would represent usable energy. On an exhaust-gas-turbocharged engine, the part above the atmospheric line (1 bar) can to some extent be utilized (1 – 4 – 5′).

Real p-V diagram
Since it is impossible during normal engine operation to maintain the basic conditions for the ideal cycle, the actual *p-V* diagram (Fig. 2, curve B) differs from the ideal *p-V* diagram.

Measures for increasing thermal efficiency

The thermal efficiency rises along with increasing air/fuel-mixture compression. The higher the compression, the higher the pressure in the cylinder at the end of the compression phase, and the larger is the enclosed area in the *p-V* diagram. This area is an indication of the energy generated during the combustion process. When selecting the compression ratio, the fuel's antiknock qualities must be taken into account.

Manifold-injection engines inject the fuel into the intake manifold onto the closed intake valve, where it is stored until drawn into the cylinder. During the formation of the air/fuel mixture, the fine fuel droplets vaporize. The energy needed for this process is in the form of heat and is taken from the air and the intake-manifold walls. On direct-injection engines the fuel is injected into the combustion chamber, and the energy needed for fuel-droplet vaporization is taken from the air trapped in the cylinder which cools off as a result. This means that the compressed air/fuel mixture is at a lower temperature than is the case with a manifold-injection engine, so that a higher compression ratio can be chosen.

Thermal losses

The heat generated during combustion heats up the cylinder walls. Part of this thermal energy is radiated and lost. In the case of gasoline direct injection, the stratified-charge air/fuel mixture cloud is surrounded by a jacket of gases which do not participate in the combustion process. This gas jacket hinders the transfer of heat to the cylinder walls and therefore reduces the thermal losses.

Further losses stem from the incomplete combustion of the fuel which has condensed onto the cylinder walls. Thanks to the insulating effects of the gas jacket, these losses are reduced in stratified-charge operation. Further thermal losses result from the residual heat of the exhaust gases.

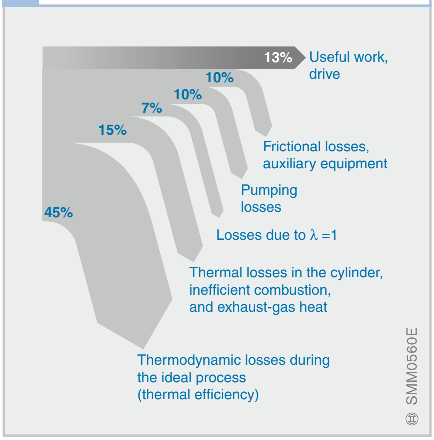
Losses at $\lambda = 1$
The efficiency of the constant-volume cycle climbs along with increasing excess-air factor (λ). Due to the reduced flame-propagation velocity common to lean air/fuel mixtures, at $\lambda > 1.1$ combustion is increasingly sluggish, a fact which has a negative effect upon the SI engine's efficiency curve. In the final analysis, efficiency is the highest in the range λ = 1.1...1.3. Efficiency is therefore less for a homogeneous air/fuel-mixture formation with λ = 1 than it is for an air/fuel mixture featuring excess air. When a 3-way catalytic converter is used for emissions control, an air/fuel mixture with λ = 1 is ab-
solutely imperative for efficient operation.

Pumping losses
During the exhaust and refill cycle, the engine draws in fresh gas during the 1st (induction) stroke. The desired quantity of gas is controlled by the throttle-valve opening.
A vacuum is generated in the intake manifold which opposes engine operation (throttling losses). Since with a gasoline direct-injection engine the throttle valve is wide open at idle and part load, and the torque is determined by the injected fuel mass, the pumping losses (throttling losses) are lower.

In the 4th stroke, work is also involved in forcing the remaining exhaust gases out of the cylinder.

Frictional losses
The frictional losses are the total of all the friction between moving parts in the engine itself and in its auxiliary equipment. For instance, due to the piston-ring friction at the cylinder walls, the bearing friction, and the friction of the alternator drive.

1 Efficiency chain of an SI engine at $\lambda = 1$



2 Sequence of the motive working process in the p-V diagram

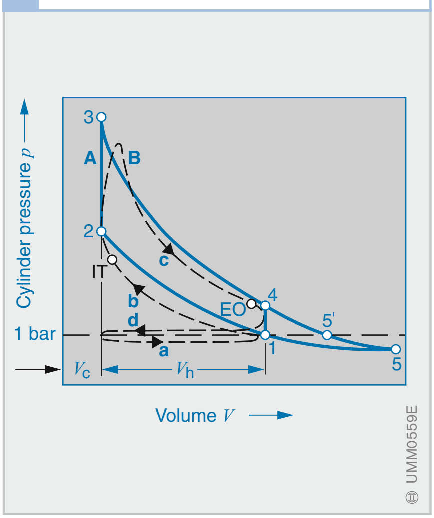


Fig. 2
A Ideal constant-volume cycle
B Real p-V diagram

a Induction
b Compression
c Work (combustion)
d Exhaust

IT Ignition point
EO Exhaust valve opens

Specific fuel consumption

Specific fuel consumption b_e is defined as the mass of the fuel (in grams) that the internal-combustion engine requires to perform a specified amount of work (kW·h, kilowatt hours). This parameter thus provides a more accurate measure of the energy extracted from each unit of fuel than the terms liters per hour, litres per 100 kilometers or miles per gallon.

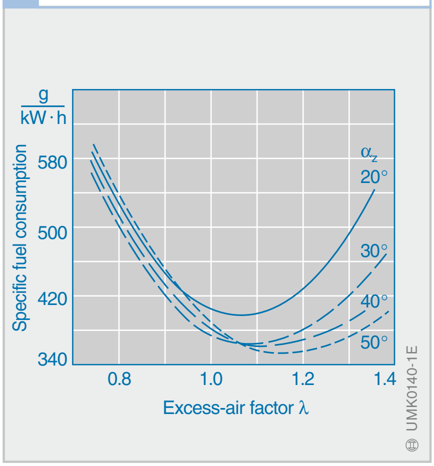
Effects of excess-air factor

Homogeneous mixture distribution

When engines operate on homogeneous induction mixtures, specific fuel consumption initially responds to increases in excess-air factor λ by falling (Fig. 1). The progressive reductions in the range extending to $\lambda = 1.0$ are explained by the incomplete combustion that results when a rich air/fuel mixture burns with inadequate air.

The throttle plate must be opened to wider apertures to obtain a given torque during operation in the lean range ($\lambda > 1$). The resulting reduction in throttling losses combines with enhanced thermodynamic efficiency to furnish lower rates of specific fuel consumption.

1 Effects of excess-air factor λ and ignition timing α_z on fuel consumption during operation with homogeneous mixture distribution



As the excess-air factor is increased, the flame front's propagation rate falls in the resulting, progressively leaner mixtures.
The ignition timing must be further advanced to compensate for the resulting lag in ignition of the combustion mixture.

As the excess-air factor continues to rise, the engine approaches the lean-burn limit, where incomplete combustion takes place (combustion miss). This results in a radical increase in fuel consumption. The excess-air factor that coincides with the lean-burn limit varies according to engine design.

Stratified-charge concept

Engines featuring direct gasoline injection can operate with high excess-air factors in their stratified-charge mode. The only fuel in the combustion chamber is found in the stratification layer immediately adjacent to the tip of the spark plug. The excess-air factor within this layer is approximately $\lambda = 1$.

The remainder of the combustion chamber is filled with air and inert gases (exhaust-gas recirculation). The large throttle-plate apertures available in this mode lead to a reduction in pumping losses. This combines with the thermodynamic benefits to provide a substantial reduction in specific fuel consumption.

Effects of ignition timing

Homogeneous mixture distribution

Each point in the cycle corresponds to an optimal phase in the combustion process with its own defined ignition timing (Fig. 1). Any deviation from this ignition timing will have negative effects on specific fuel consumption.

Stratified-charge concept

The range of possibilities for varying the ignition angle is limited on direct-injection gasoline engines operating in the stratified-charge mode. Because the ignition spark must be triggered as soon as the mixture cloud reaches the spark plug, the ideal ignition point is largely determined by injection timing.

Achieving ideal fuel consumption

During operation on homogeneous induction mixtures, gasoline engines must operate on a stoichiometric air/fuel ratio of $\lambda = 1$ to create an optimal operating environment for the 3-way catalytic converter. Under these conditions using the excess-air factor to manipulate specific fuel consumption is not an option. Instead, the only available recourse is to vary the ignition timing. Defining ignition timing always equates with finding the best compromise between maximum fuel economy and minimal levels of raw exhaust emissions. Because the catalytic converter's treatment of toxic emissions is very effective once it is hot, the aspects related to fuel economy are the primary considerations once the engine has warmed to normal operating temperature.

Fuel-consumption map

Testing on an engine dynamometer can be used to determine specific fuel consumption in its relation to brake mean effective pressure and to engine speed. The monitored data are then entered in the fuel consumption map (Fig. 2). The points representing levels of specific fuel consumption are joined to form curves. Because the resulting graphic portrayal resembles a sea shell, the lines are also known as shell or conchoid curves.

As the diagram indicates, the point of minimum specific fuel consumption co - incides with a high level of brake mean effective pressure p_{me} at an engine speed of roughly 2600 rpm.

Because the brake mean effective pressure also serves as an index of torque generation M, curves representing power output P can also be entered in the chart. Each curve assumes the form of a hyperbola. Although the chart indicates identical power at different engine speeds and torques (operating points A and B), the specific fuel consumption rates at these operating points are not the same. At Point B the engine speed is lower and the torque is higher than at Point A. Engine operation can be shifted toward Point A by using the transmission to select a gear with a higher conversion ratio.

2 Fuel-consumption map for gasoline engine with homogeneous induction mixture

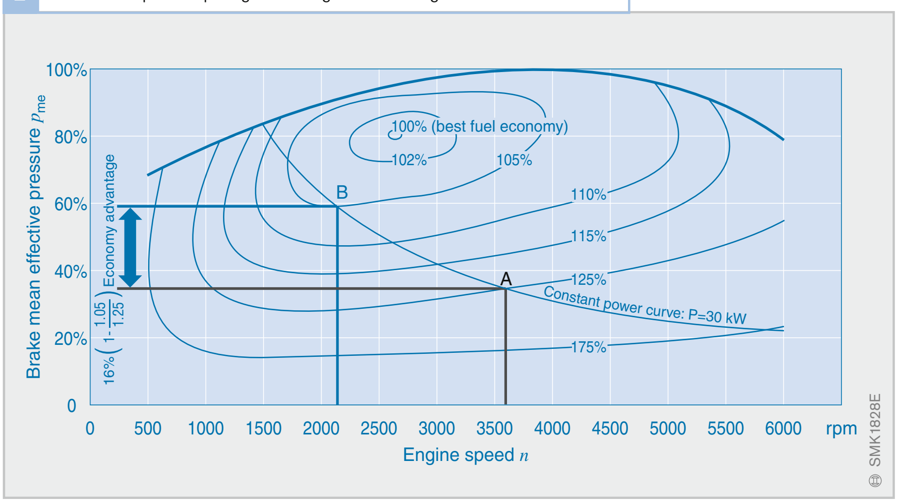


Fig. 2
Engine data:
4-cylinder gasoline engine
Displacement:
$V_H = 2.3$ litres
Power:
$P = 110$ kW at 5400 rpm
Torque peak:
$M = 220$ N·m at 3700...4500 rpm
Brake mean effective pressure:
$p_{me} = 12$ bar (100 %)

Calculating torque M and power P with numerical value equations:
$M = V_H \cdot p_{me}/0.12566$
$P = M \cdot n/9549$

M in N·m
V_H in dm³
p_{me} in bar
n in rpm
P in kW

Combustion knock

Among the factors imposing limits on the latitude for enhancing an engine's thermodynamic efficiency and increasing power-plant performance are spontaneous pre-ignition and detonation. This highly undesirable phenomenon is frequently accompanied by an audible "pinging" noise, which is why the generally applicable term for this condition is "knock". Knock occurs when portions of the mixture ignite spontaneously before being reached by the flame front. The intense heat and immense pressure peaks produced by combustion knock subject pistons, bearings, cylinder head and head gasket to enormous mechanical and thermal loads. Extended periods of knock can produce blown head gaskets, holed piston crowns and engine seizure, and leads to destruction of the engine.

The sources of combustion knock

The spark plug ignites the air/fuel mixture toward the end of the compression stroke, just before the piston reaches Top Dead Centre (TDC). Because several milliseconds can elapse until the entire air/fuel mixture can ignite (the precise ignition lag varies according to engine speed), the actual combustion peak occurs after TDC.

The flame front extends outward from the spark plug. After being compressed during the compression stroke, the induction mixture is heated and pressurized as it burns within the combustion chamber. This further compresses any unburned air/fuel mixture within the chamber. As a result, some portions of the compressed air/fuel mixture can attain temperatures high enough to induce spontaneous auto-ignition (Fig. 1). Sudden detonation and uncontrolled combustion are the results.

When this type of detonation occurs it produces a flame front with a propagation rate 10 to 100 times that associated with the normal combustion triggered by the spark plug (approximately 20 m/s). This uncontrolled combustion generates pressure pulses which spread out in circular patterns from the core of the process. It is when these pulsations impact against the walls of the cylinder that they generate the metallic pinging sound typically associated with combustion knock.

Other flame fronts can be initiated at hot spots within the combustion chamber. Among the potential sources of this hot-spot ignition are spark plugs which during operation heat up excessively due to their heat range being too low. This type of pre-ignition produces engine knock by initiating combustion before the ignition spark is triggered.

Engine knock can occur throughout the engine's speed range. However, it is not possible to hear it at extremely high rpm, when its sound is obscured by the noise from general engine operation.

Factors affecting tendency to knock

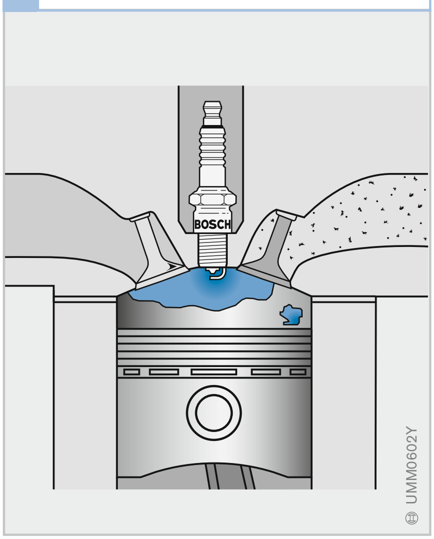
Substantial ignition advance: Advancing the timing to ignite the mixture earlier produces progressively higher combustion-chamber temperatures and correspondingly extreme pressure rises.
High cylinder-charge density: The charge density must increase as torque demand rises (engine load factor). This leads to high temperatures during compression.
Fuel grade: Because fuels with low octane ratings furnish only limited resistance to knock, compliance with manufacturer's specifications for fuel grade(s) is vital.
Excessively high compression ratio: One potential source of excessively high compression would be a cylinder head gasket of less than the specified thickness. This leads to higher pressures and temperatures in the air/fuel mixture during compression. Deposits and residue in the combustion chamber (from ageing, etc.) can also produce a slight increase in the effective compression ratio.
Cooling: Ineffective heat dissipation within the engine can lead to high mixture temperatures within the combustion chamber.
Geometry: The engine's knock tendency can be aggravated by unfavorable combustion-chamber geometry. Poor turbulence and swirl characteristics caused by unsatisfactory intake-manifold tract configurations are yet another potential problem source.

Engine knock with direct gasoline injection

With regard to engine knock, when operating with homogeneous air/fuel mixtures direct-injection gasoline engines behave the same as manifold-injected power plants. One major difference is the cooling effect exerted by the evaporating fuel during direct injection, which reduces the temperature of the air within the cylinder to levels lower than those encountered with manifold injection.

During operation in the stratified-charge mode it is only in the area immediately adjacent to the spark plug tip that an ignitable mixture is present. When the remainder of the combustion chamber is filled with air or inert gases, there is no danger of spontaneous ignition and engine knock. Nor is there any danger of detonation when an extremely lean air/fuel mixture is present within these outlying sections of the combustion chamber. The ignition energy required to generate a flame in this kind of lean mixture would be substantially higher than that needed to spark a stoichiometric combustion mixture. This is why stratified-charge operation effectively banishes the danger of engine knock.

1 The sources of combustion knock



Avoiding consistent engine knock

To effectively avoid pre-ignition and detonation, ignition systems not equipped with knock detection rely on ignition timing with a safety margin of 5...8 degrees (crankshaft) relative to the knock limit.

Ignition systems featuring knock detection employ one or several knock sensors to monitor acoustic waves in the engine. The engine-management ECU detects knock in individual combustion cycles by analysing the electrical signals relayed by these sensors. The ECU then responds by retarding the ignition timing for the affected cylinder to prevent continuous knock. The system then gradually advances the ignition timing back toward its original position. This progressive advance process continues until the ignition timing is either back at the initial reference point programmed into the engine's software map, or until the system starts to detect knock again. The engine management regulates the timing advance for each cylinder individually.

The limited number of combustion events with mild knock of the kind that also occur with knock control are not injurious to the health of the engine. On the contrary: They help dissolve deposits formed by oil and fuel additives within the combustion chamber (on intake and exhaust valves, etc.), allowing them to be combusted and/or discharged with the exhaust gases.

Advantages of knock control

Thanks to reliable knock recognition, engines with knock control can use higher compression ratios. Co-ordinated control of the ignition's timing advance also makes it possible to do without the safety margin between the timing point and the knock threshold; the ignition timing can be selected for the "best case" instead of the "worst case" scenario. This provides benefits in terms of thermodynamic efficiency. Knock control

- reduces fuel consumption,
- enhances torque and power, and
- allows engine operation on different fuels within an extended range of octane ratings (both premium and regular unleaded, etc.).

Inductive ignition system

Ignition of the air/fuel mixture in the gasoline engine is electric; it is produced by generating a flashover between the electrodes on a spark plug. The ignition-coil energy converted in the spark ignites the compressed mixture immediately adjacent to the spark plug, creating a flame front which then spreads to ignite the mixture in the entire combustion chamber. The inductive ignition system generates in each power stroke the high voltage required for flashover and the spark duration required for ignition. The electrical energy drawn from the vehicle electrical system battery is temporarily stored in the ignition coil for this purpose.

The most significant application for the inductive ignition system is in passenger cars with gasoline engines. The most commonly used are four-stroke engines with four cylinders.

Design

Figure 1 shows the basic design of the ignition circuit of an inductive ignition system using the example of a system with distributorless (stationary) voltage distribution – as is used in all current applications – and single-spark ignition coils. The ignition circuit comprises the following components:

- Ignition driver stage (5), which is integrated in the Motronic ECU or in the ignition coil
- Ignition coils (3), designed as pencil coils or as a compact coil to generate one spark (as illustrated) or two sparks
- Spark plugs (4), and
- Connecting devices and interference suppressors

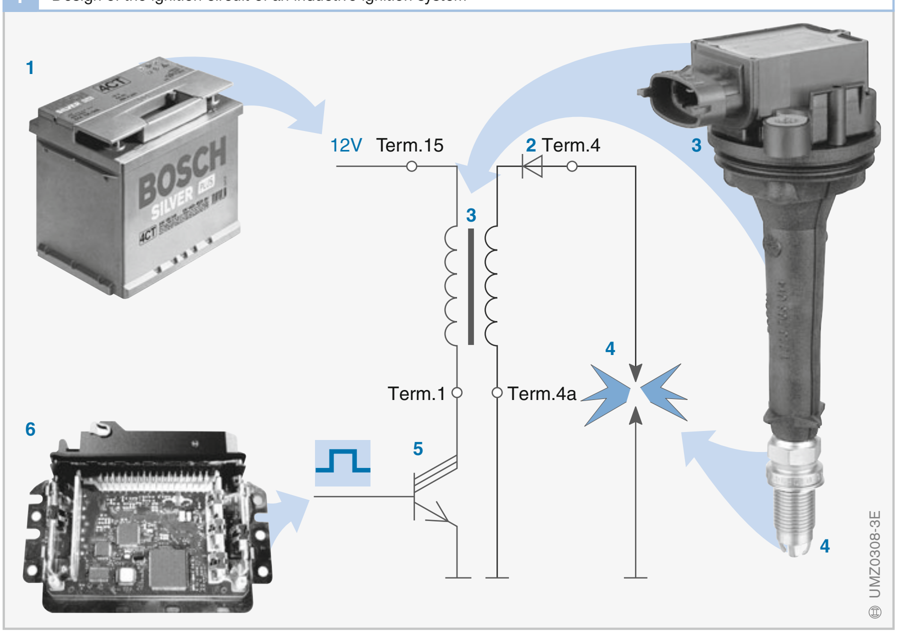
Older ignition systems with rotating high-voltage distribution require an additional high-voltage distributor. This ensures that the ignition energy generated in the ignition coil is directed to the correct spark plug.

Fig. 1
Illustration of a cylinder of an inductive ignition system with distributorless voltage distribution and single-spark ignition coils

1 Battery
2 AAS diode (Activation Arc Suppression), integrated in ignition coil
3 Ignition coil
4 Spark plug
5 Ignition driver stage (integrated in engine ECU or in ignition coil)
6 Engine ECU Motronic

Term. 15, Term. 1, Term. 4, Term. 4a
Terminal designation
⎍ Actuation signal for ignition driver stage

1 Design of the ignition circuit of an inductive ignition system



K. Reif (Ed.), *Fundamentals of Automotive and Engine Technology*,
DOI 10.1007/978-3-658-03972-1_7, © Springer Fachmedien Wiesbaden 2014

Function and method of operation

It is the function of the ignition to ignite the compressed air/fuel mixture and thus initiate its combustion. Safe combustion of the mixture must be guaranteed in the process. To this end, sufficient energy must be stored in the ignition coil prior to the moment of ignition and the ignition spark must be generated at the correct moment of ignition.

All the components of the ignition system are adapted in terms of their designs and performance data to the demands of the overall system.

Generating the ignition spark

A magnetic field is built up in the ignition coil when a current flows in the primary circuit. The ignition energy required for ignition is stored in this magnetic field.

Interrupting the coil current at the moment of ignition causes the magnetic field to collapse. This rapid magnetic-field change induces a high voltage (Fig. 2) on the secondary side of the ignition coil as a result of the large number of turns (turns ratio approx. 1:100). When the ignition voltage is reached, flashover occurs at the spark plug and the compressed air/fuel mixture is ignited.

The current in the primary winding only gradually attains its setpoint value because of the induced countervoltage. Because the energy stored in the ignition coil is dependent on the current ($E = {}^1/_2LI^2$), a certain amount of time (dwell period time) is required in order to store the energy necessary for ignition. This dwell period is dependent on, among others, the vehicle system voltage. The ECU program calculates from the dwell period and the moment of ignition the cut-in point, and cuts the ignition coil in via the ignition driver stage and out again at the moment of ignition.

Flame-front propagation

After the flashover, the voltage at the spark plug drops to the spark voltage (Fig. 2). The spark voltage is dependent on the length of the spark plasma (electrode gap and deflection due to flow) and ranges between a few hundred volts and well over 1 kV. The ignition-coil energy is converted in the ignition spark during the ignition-spark period; this spark duration lasts between 100 µs to over 2 ms. Following the breakaway of the spark, the attenuated voltage decays.

The electrical spark between the spark-plug electrodes generates a high-temperature plasma. When the mixture at the spark plug is ignitable and sufficient energy is supplied by the ignition system, the flame core that is created develops into an automatically propagating flame front.

Moment of ignition

The instant at which the spark ignites the air/fuel mixture within the combustion chamber must be selected with extreme precision. It is usually specified as an ignition angle in °cks (crankshaft) referred to Top Dead Center (TDC). This variable has a crucial influence on engine operation and determines

2 Voltage curve at the electrodes

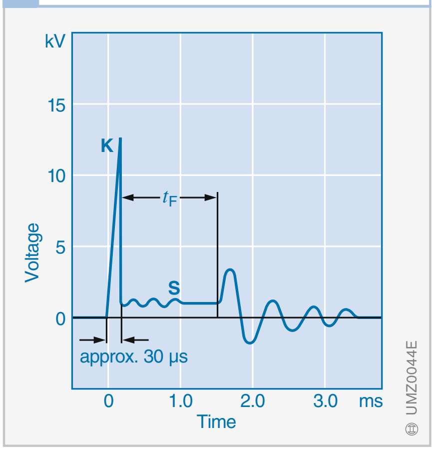


Fig. 2
K Spark head
S Spark tail
t_F Spark duration

- The delivered torque
- The exhaust-gas emissions, and
- The fuel consumption

The moment of ignition is specified in such a way that all requirements are met as effectively as possible. However, continuous engine knocking must not develop during operation.

The influencing variables that determine the moment of ignition are engines speed and engine load, or torque. Additional variables, such as, for example, engine temperature, are also used to determine the optimal moment of ignition. These variables are recorded by sensors and then relayed to the engine ECU (Motronic). The moment of ignition is calculated from program maps and characteristic curves, and the actuation signal for the ignition driver stage is generated.

Knock control

Knock is a phenomenon which occurs when ignition takes place too early (Fig. 3). Here, once regular combustion has started, the rapid pressure increase in the combustion chamber leads to auto-ignition of the unburnt residual mixture which has not been reached by the flame front. The resulting abrupt combustion of the residual mixture leads to a considerable local pressure increase. The pressure wave which is generated propagates, strikes the cylinder walls, and can be heard as combustion knock.

If knock continues over a longer period of time, the engine can incur mechanical damage caused by the pressure waves and the excessive thermal loading. To prevent knock on today's high-compression engines, no matter whether of the manifold-injection or direct-injection type, knock control is now a standard feature of the engine-management system. With knock control, knock sensors (structure-borne-noise sensors) detect the start of knock and the ignition timing is retarded at the cylinder concerned (Fig. 4). The pressure increase after the mixture has ignited therefore occurs later, which reduces the tendency to knock. When the knocking stops, the ignition-timing adjustment is reversed in stages. To obtain the best-possible engine efficiency, therefore, the basic adaptation of the ignition angle (ignition map) can be located directly at the knock limit.

Fig. 3
Pressure curves at different moments of ignition
1 Ignition Z_a at correct moment
2 Ignition Z_b too advanced (combustion knock)
3 Ignition Z_c too retarded

3 Pressure curve in the combustion chamber

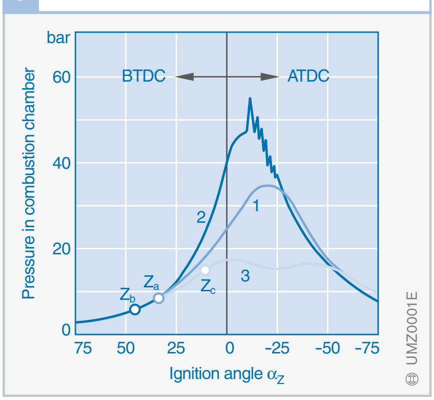
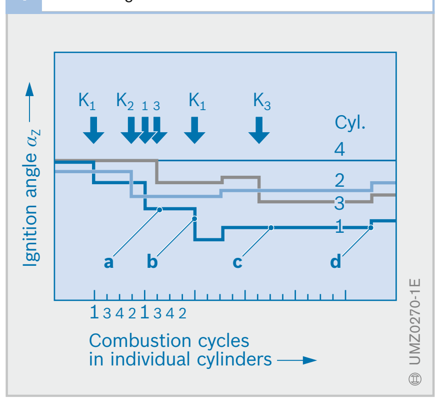


Fig. 4
K1…3 Occurrence of knock at cylinders 1…3, no knock at cylinder 4
a Dwell time before timing retardation
b Drop depth
c Dwell time before reverse adjustment
d Timing advance

4 Control algorithm of knock control



Ignition parameters

Moment of ignition

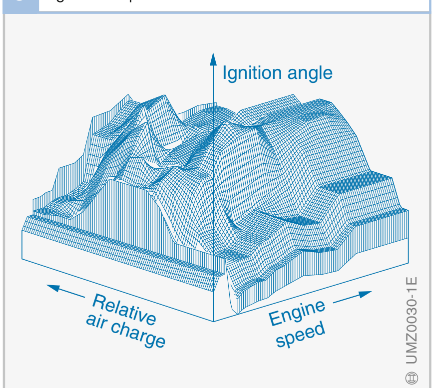
Engine-speed and load dependence

Once ignition has been initiated by the ignition spark, it takes a few milliseconds for the air/fuel mixture to burn completely. This period of time remains roughly constant as long as the mixture composition remains unchanged. The moment of ignition point must be selected so that main combustion, and the accompanying pressure peak in the cylinder, takes place shortly after TDC. As engine speed increases, the ignition angle must therefore be advanced.

The cylinder charge also has an effect on the combustion curve. The flame front propagates at a slower rate when the cylinder charge is low. For this reason, with a low cylinder charge, the ignition angle must also be advanced.

In the case of gasoline direct injection, the range for variation of the moment of ignition in stratified-charge mode is limited by the end of injection and the time needed for mixture preparation during the compression stroke.

5 Ignition map



Basic adaptation of ignition angle

In electronically controlled ignition systems, the ignition map (Fig. 5) takes into account the influence of engine speed and cylinder charge on the ignition angle. This map is stored in the engine-management system's data memory, and forms the basic adaptation of the ignition angle.

The map's x and y axes represent the engine speed and the relative air charge. A specific number of values, typically 16, forms the data points of the map. One ignition angle is stored for each pair of values. The map therefore contains 256 adjustable ignition-angle values. By applying linear interpolation between two data points, the number of ignition-angle values is increased to 4096.

Using the ignition-map principle for electronic control of the ignition angle means that, for every engine operating point, it is possible to select the best-possible ignition angle. These maps are ascertained on the engine test stand, or dynamic power analyzer, where demands pertaining to, for example, noise, comfort and component protection are also taken into account.

Additive ignition-angle corrections

Different impacting factors on the moment of ignition are taken into account through additive corrections of the basic ignition angle, such as, for instance, knock control or warming-up after the starting phase. The engine temperature has a further influence on the selection of the ignition angle (e.g., shifting of the knock limit when the engine is hot).

Temperature-dependent ignition-angle corrections are therefore also necessary. Such corrections are stored in the data memory in the form of fixed values or characteristic curves (e.g., temperature-dependent correction). They shift the basic ignition angle by the specified amount. The ignition-angle correction can be either an advance or a retardation.

Ignition angles for specific operating conditions

Specific operating states, e.g., starting or stratified-charge mode with gasoline direct injection, require ignition angles that deviate from the ignition map. In such cases, access is obtained to special ignition angles stored in the data memory.

Dwell period

The energy stored in the ignition coil is dependent on the level of the primary current at the moment of ignition (cut-out current) and the inductance of the primary winding. The level of the cut-out current is essentially dependent on the cut-in period (dwell period) and the vehicle system voltage. The dwell periods for obtaining the desired cut-out current are stored in voltage-dependent curves or program maps. Changing the dwell period by way of the temperature can also be compensated for.

In order not to thermally overload the ignition coil, it is essential to adhere rigidly to the time required to generate the required ignition energy in the coil.

Ignition voltage

The ignition voltage at the point where flashover between the spark-plug electrodes occurs is the ignition-voltage demand. It is dependent, among other things, on

- The density of the air/fuel mixture in the combustion chamber, and thus on the moment of ignition
- The composition of the air/fuel mixture (excess-air factor, lambda value)
- The flow velocity and turbulence
- The electrode geometry
- The electrode material, and
- The electrode gap

It is vital that the ignition voltage supplied by the ignition system always exceed the ignition-voltage demand under all conditions.

Ignition energy

The cut-out current and the ignition-coil parameters determine the amount of energy that the coil stores for application as ignition energy in the spark. The level of ignition energy has a decisive influence on flame-front propagation. Good flame-front propagation is essential to delivering high-performance engine operation coupled with low levels of toxic emissions. This places considerable demands on the ignition system.

Energy balance of an ignition

The energy stored in the ignition coil is released as soon as the ignition spark is initiated. This energy is divided into two separate components.

Spark head

In order that an ignition spark can be generated at the spark plug, first the secondary-side capacitance C of the ignition circuit must be charged, and this is released again on flashover. The energy required for this increases quadratically with the ignition voltage U ($E = {}^1/_2\, CU^2$). Figure 6 shows the component of this energy contained in the spark head.

Spark tail

The energy still remaining in the ignition coil after flashover (inductive component) is then released in the course of the spark duration. This energy represents the difference between the total energy stored in the ignition coil and the energy released during capacitive discharge. In other words: The higher the ignition-voltage demand, the greater the component of total energy contained in the spark head, and the less energy is converted in the spark duration, i.e., the shorter the spark duration. When the ignition-voltage demand is high, due for instance to badly worn spark plugs, the energy stored in the spark tail may no longer be enough to completely burn an already ignited mixture or to re-ignite a spark that has broken away.

Further increases in the ignition-voltage demand lead to the ignition-miss limit being reached. Here, the available energy is no longer enough to generate a flashover, and instead it decays in a damped oscillation (ignition miss).

Energy losses

Figure 6 shows a simplified representation of the existing conditions. Ohmic resistance in the ignition coil and the ignition cables combined with the suppression resistors cause losses, which are then unavailable as ignition energy.

Additional losses are produced by shunt resistors. While these losses can result from contamination on the high-voltage connections, the primary cause is soot and deposits on the spark plugs within the combustion chamber.

The level of shunt losses is also dependent on the ignition-voltage demand. The higher the voltage applied at the spark plug, the greater the currents discharging through the shunt resistors.

Mixture ignition

Under ideal (e.g., laboratory) conditions, the energy required to ignite an air/fuel mixture with an electrical spark for each individual injection is approximately 0.2 mJ, provided the mixture in question is static, homogeneous and stoichiometric. Under such conditions, rich and lean mixtures require in excess of 3 mJ.

The energy that is actually required to ignite the mixture is only a fraction of the total energy in the ignition spark, the ignition energy. With conventional ignition systems, energy levels in excess of 15 mJ are needed to generate a high-voltage flashover at the moment of ignition at high breakdown voltages. This additional energy is required to charge the capacitance on the secondary side. Further energy is required to maintain a specific spark duration and to compensate for losses, due for instance to contamination shunts at the spark plugs. These requirements amount to ignition energies of at least 30...50 mJ, a figure which corresponds to an energy level of 60...120 mJ stored in the ignition coil.

6 Ignition energy balance without shunt, resistance and Zener losses

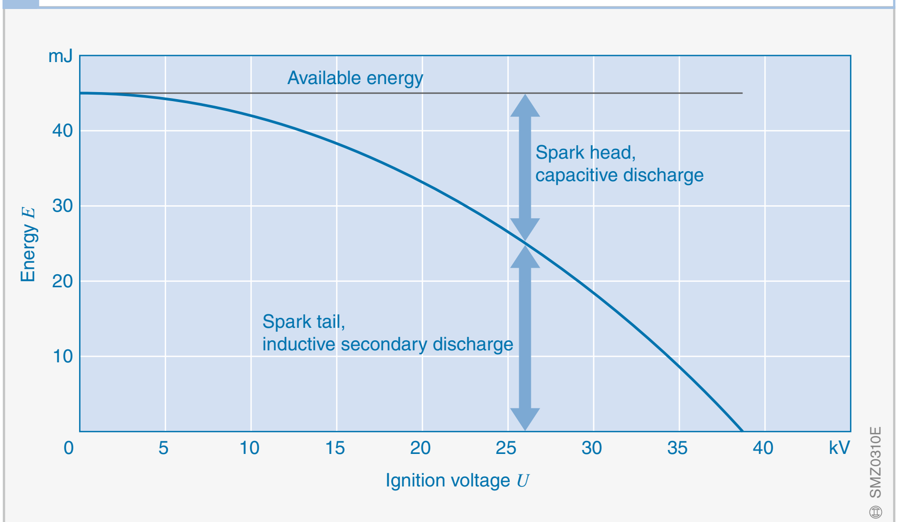


Fig. 6
The energy figures are for a sample ignition system with a coil capacitance of 35 pF, an external load of 25 pF (total capacitance C = 60 pF) and secondary inductance of 15 H.

Turbulence within the mixture of the kind encountered when engines with gasoline direct injection are operated in stratified-charge mode can deflect the ignition spark to such an extent that it breaks away (Fig. 7). A number of follow-up sparks is then needed to ignite the mixture, and this energy must also be provided by the ignition coil.

The ignition tendency decreases in the case of lean mixtures. A particularly high level of energy is therefore required to be able to cover the increased ignition-voltage demand and at the same time to ensure an effectively long spark duration.

If inadequate ignition energy is available, the mixture will fail to ignite. No flame front is established, and combustion miss occurs. This is why the system must furnish adequate reserves of ignition energy: To ensure reliable detonation of the air/fuel mixture, even under unfavorable external conditions. It may be enough to ignite just a small portion of the mixture directly with the spark plug. The mixture igniting at the spark plug then ignites the remaining mixture in the cylinder and thereby initiates the combustion process.

Factors affecting ignition performance

Efficient preparation of the mixture with unobstructed access to the spark plug improves ignition performance, as do extended spark durations and large spark lengths or large electrode gaps. Mixture turbulence can also be an advantage, provided enough energy is available for follow-up ignition sparks should these be needed. Turbulence supports rapid flame-front distribution in the combustion chamber, and with it the complete combustion of the mixture in the entire combustion chamber.

Spark-plug contamination is also a significant factor. If the spark plugs are very dirty, energy is discharged from the ignition coil through the spark-plug shunt (deposits) during the period in which the high voltage is being built up. This reduces the high voltage whilst simultaneously shortening spark duration. This affects exhaust emissions, and can even lead to ignition misses under extreme conditions, as when the spark plugs are severely contaminated or wet.

Ignition misses lead to combustion misses, which increase both fuel consumption and pollutant emissions, and can also damage the catalytic converter.

7 Ignition spark in an engine with direct injection

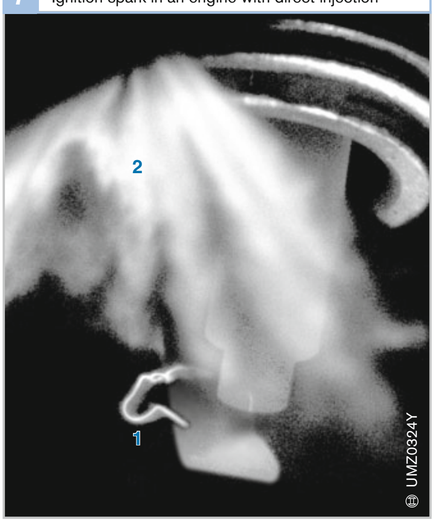


Fig. 7
Photograph of an ignition spark: taken in a transparent engine using a high-speed camera

1 Ignition spark
2 Fuel spray

Danger of accident

All electrical ignition systems are high-voltage systems. To avoid potential dangers, always switch off the ignition or disconnect the power source before working on any ignition system. These precautions apply to, e.g.,

- Replacing components, such as ignition coils, spark plugs, ignition cables, etc.
- Connecting engine testers, such as timing stroboscope, dwell-angle/speed tester, ignition oscilloscope, etc.

When checking the ignition system, remember that dangerously high levels of voltage are present within the system whenever the ignition is on. All tests and inspections should therefore only be carried out by qualified professional personnel.

Voltage distribution

Rotating high-voltage distribution

The high voltage generated in the ignition coil (Fig. 8a, Pos. 2) must be applied at the correct spark plug at the moment of ignition. In the case of rotating high-voltage distribution, the high voltage generated by this single ignition coil is mechanically distributed to the individual spark plugs (5) by an ignition distributor (3). The rotation speed and the position of the distributor rotor, which establishes the electrical connection between the ignition coil and the spark plug, are coupled to the camshaft.

This form of distribution is no longer of any significance to new, modern-day engine-management systems.

Distributorless (stationary) voltage distribution

The mechanical components have been dispensed with in the distributorless, or stationary, voltage-distribution system (Fig. 8b). Voltage is distributed on the primary side of the ignition coils, which are connected directly to the spark plugs. This permits wear-free and loss-free voltage distribution. There are two versions of this type of voltage distribution.

System with single-spark ignition coils

Each cylinder is allocated an ignition driver stage and an ignition coil. The engine ECU actuates the ignition driver stages in specified firing order.

Since there are no distributor losses, these ignition coils can be very small in design. They are preferably mounted directly over the spark plug.

Distributorless voltage distribution with single-spark ignition coils can be used with any number of cylinders. There are no limitations on the ignition-timing adjustment range. In this case, the spark plug of the cylinder which is at firing TDC is the one that fires. However, the system does also have to be synchronized by means of a camshaft sensor with the camshaft.

System with dual-spark ignition coils

One ignition driver stage and one ignition coil are allocated to every two cylinders. The ends of the secondary winding are each connected to a spark plug in different cylinders. The cylinders have been chosen so that when one cylinder is in the compression stroke, the other is in the exhaust stroke (applies only to engines with an even number of cylinders). Flashover occurs at both spark plugs at the moment of ignition. Because it is important to prevent residual exhaust gas or fresh induction gas from being ignited by the spark during the exhaust stroke additional spark, the latitude for varying ignition timing is limited with this system. However, it does not need to be synchronized with the camshaft. Because of these limitations, dual-spark ignition coils cannot be recommended.

8 Voltage-distribution concept

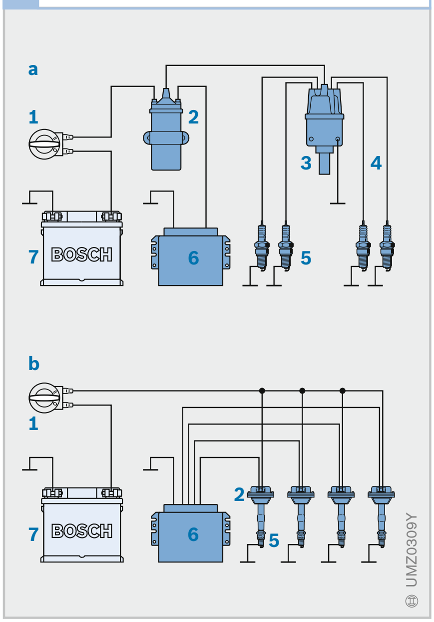


Fig. 8
a Rotating distribution
b Distributorless (stationary) distribution with single-spark ignition coils

1 Ignition lock
2 Ignition coil
3 Ignition distributor
4 Ignition cable
5 Spark plug
6 ECU
7 Battery

Ignition driver stage

Function and method of operation

The function of the ignition driver stage is to control the flow of primary current in the ignition coil. It is usually designed as a three-stage power transistor with BIP technology (Bosch Integrated Power, bipolar technology). The functions of primary-voltage limitation and primary-current limitation are integrated as monolithic components on the ignition driver stage, and protect the ignition components against overload.

During operation, the ignition driver stage and the ignition coil both heat up. In order not to exceed the permissible operating temperatures, it is necessary that appropriate measures be taken to ensure that the heat losses are reliably dissipated to the surroundings even when outside temperatures are high. In order to avoid high power loss in the ignition driver stage, the function of primary-current limitation is only to limit the current in the event of a fault (e.g., short circuit).

9 Block diagrams of ignition driver stages

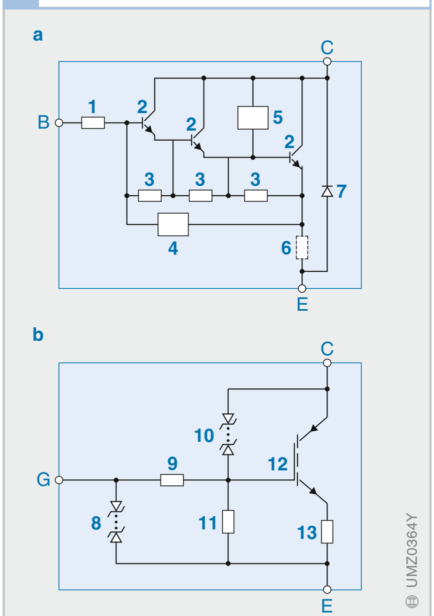


Fig. 9
- a BIP ignition driver stage (monolithic integrated)
- a IGBT ignition driver stage (monolithic integrated)

1. Base resistor
2. Triple Darlington transistor
3. Basic emitter resistors
4. Emitter current regulator
5. Collector-voltage limitation
6. Current-recording resistor
7. Inverse diode
8. Polysilicon protective-diode chain
9. Gate resistor
10. Polysilicon clamp-diode chain for collector-voltage limitation
11. Gate emitter resistor
12. IGBT transistor
13. Resistor (omitted from standard IGBT)

- B Base
- E Emitter
- C Collector
- G Gate

In the future, the three-stage circuit-breakers will be superseded by the new IGBTs (Insulated Gate Bipolar Transistors, hybrid form on field-effect and bipolar transistors), also for ignition applications. The IGBT has some advantages over BIP:

- Virtually power-free actuation (voltage instead of current)
- Low saturation voltage
- Higher load current
- Lower switching times
- Higher clamp voltage
- Higher holding temperature
- Protected against polarity reversal in the 12 V vehicle electrical system

Design variations

Ignition driver stages are categorized into internal and external driver stages. The former are integrated on the engine ECU's printed-circuit board, and the latter are located in their own housing outside the engine ECU. Due to the costs involved, external driver stages are no longer used on new developments.

Furthermore, it is becoming increasingly common for driver stages to be incorporated in the ignition coil. This solution avoids cables in the wiring harness which carry high currents and are subjected to high voltages. In addition, the power loss incurred in the Motronic ECU is accordingly lower. Stricter demands with regard to actuation, diagnostic capability and temperature load are made of the driver stages integrated in the ignition coil. These demands are derived from the installation circumstances directly on the engine with higher ambient temperatures, ground offsets between ECU and ignition coil, and the additional expenditure involved in transmitting diagnostic information from the ignition coil to the ECU either via an additional cable or through the intelligent use of the control line to include the return transmission of diagnostic information.

Connecting devices and interference suppressors

Ignition cables

The high voltage generated in the ignition coil must be delivered to the spark plug. For this purpose, plastic-insulated, high-voltage-proof cables with special connectors at their ends for contacting the high-voltage components are used with ignition coils which are not mounted directly on the spark plug (e.g., dual-spark ignition coils).

Since, for the ignition system, each high-voltage cable represents a capacitive load which reduces the available secondary voltage, the ignition cables must be kept as short as possible.

Interference-suppression resistors, screening

Each flashover is a source of interference due to its pulse-shaped discharge. Interference-suppression resistors in the high-voltage circuit limit the peak current during discharge. In order to minimize the interference radiation from the high-voltage circuit, the suppression resistors should be installed as close as possible to the source of interference.

Normally, the suppression resistors are integrated in the spark-plug connectors and cable connectors. Spark plugs are also available which feature an integral suppression resistor. However, increasing resistance on the secondary side leads to additional energy losses in the ignition circuit, with lower ignition energy at the spark plug as the ultimate result.

Interference radiation can be even further reduced by partially or completely screening the ignition system. This screening includes the ignition cables. This effort is justified only in special cases (official government and military vehicles, radio equipment with high transmitting power).

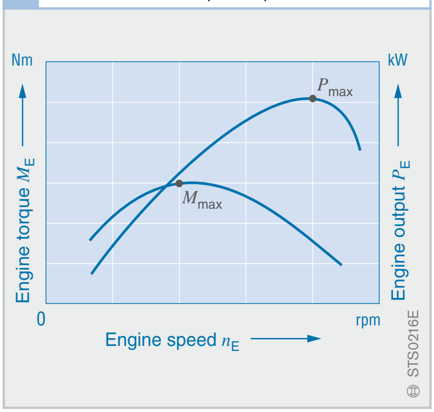
10 Ignition cables

Fig. 10
a Cable set with straight connectors and unscreened spark-plug connectors
b Cable set with elbow connectors and partially screened spark-plug connectors

Transmissions for Motor Vehicles

Every motor vehicle engine operates within a specific speed range which is limited by the idle speed and the maximum speed. Power and torque are not offered uniformly and the maximum values are only available in partial ranges.
Transmissions therefore convert the engine torque and the engine speed in accordance with vehicle traction requirements in such a way that the power remains roughly constant. They also allow the different directions of rotation for forward and reverse travel.

1 Internal-combustion engine, characteristics for torque and power



Transmission in the Drivetrain

Internal-combustion engines do not have a constant torque and power characteristic over the speed range available to them (idle to high idle speed). The optimum “elastic” speed range lies between maximum torque and maximum power (Figure 1). For this reason, a vehicle cannot start off from a state of rest where the engine is stopped. To do this, it requires a power take-up element (e.g. clutch).

Furthermore, the available engine torque is not sufficient for gradients and powerful acceleration. For this purpose, a suitable gear ratio for adapting traction and torque and for optimizing fuel consumption must be made available.

Engines only have one direction of rotation as well, with the result that they require a changeover facility for forward and reverse travel.

As Figure 2 shows, the transmission is situated in a central position on the drivetrain and thus substantially influences the drivetrain’s effectiveness.

In addition, an analysis of the losses that arise in the drivetrain show that, after the engine, it is the transmission which offers the most possibilities for optimization based on NEFZ driving cycle (Figure 3).

2 Drivetrain (overview)

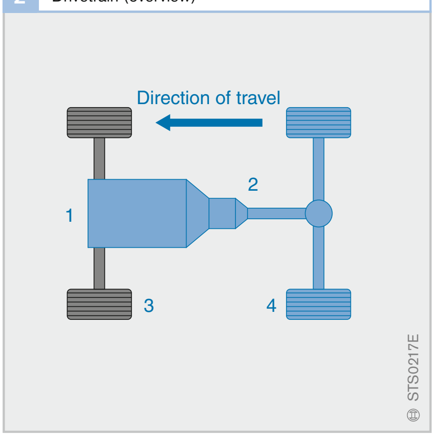
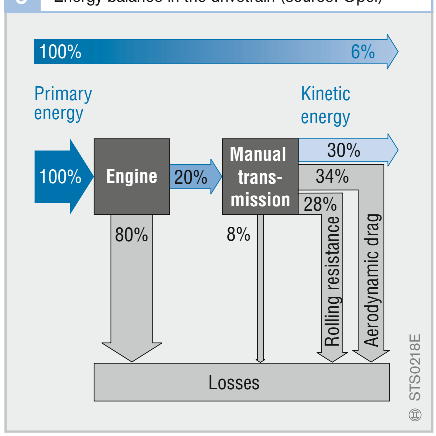


Fig. 2
1 Engine
2 Transmission
3 Front axle
4 Rear axle with differential (output)

3 Energy balance in the drivetrain (source: Opel)



K. Reif (Ed.), *Fundamentals of Automotive and Engine Technology*,
DOI 10.1007/978-3-658-03972-1_8, © Springer Fachmedien Wiesbaden 2014

Transmission History 1

Benz patent motor carriage from 1886 with belt and chain drives

When Daimler, Maybach, and Benz launched their first road vehicles, pioneers of motive power engineering had already developed the machine parts required for power transmission to a considerable extent. Names such as Leonardo da Vinci, Dürer, Galileo, Hooke, Bernoulli, Euler, Grashof, and Bach had played a significant role in these developments.

Power transmission in an automobile must guarantee the functions of starting and engine-speed and torque conversion for forward and reverse travel. These functions call for actuators and shifting elements which intervene in the power-flow and perform engine-speed and torque conversion.

The first operational Benz patent motor carriage appeared in 1886. It was the *first three-wheel vehicle* to be conceived in its entirety specifically for motorized road traffic. It may well have had just one gear, but it did not have a start-up clutch. In order to get the carriage moving at all, it was necessary to push it or crank it with the flywheel.

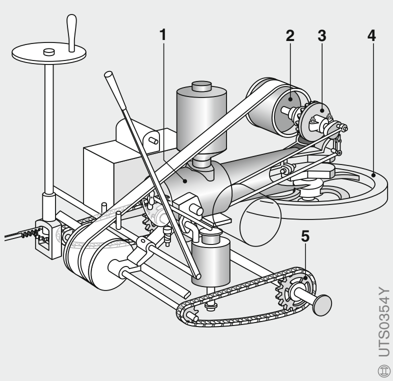
A single-cylinder four-stroke engine with a displacement of 984 cc and a power output of 0.88 HP (0.65 kW) served as the drive unit for this Benz three-wheeler.

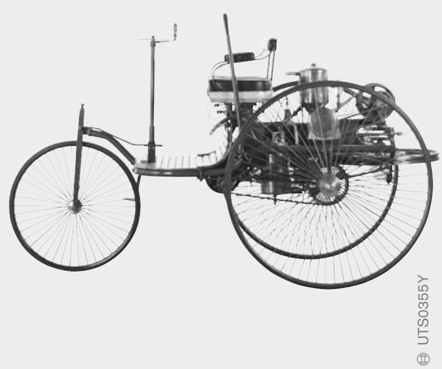
Benz utilized the following machine parts to transfer the motive force of his engine to the road:

The end of the engine's crankshaft held the flywheel, which ensured that the engine ran more smoothly and which could also be used to crank the engine. Since the engine was built over the rear axle, a bevel gear arranged at right angles transmitted power in a small space to a belt drive, which reduced the rotational speed slightly to an intermediate shaft. Finally, a chain drive reduced the speed further to the powered axle.

The belt and chain drives dating from the origins of the automobile were gradually replaced by a gear train. But, today, they are experiencing a renaissance in the form of the continuously variable transmission (CVT). A CVT transmission consists of a variator with two V-pulleys and a flexible steel push-belt. As soon as the pressure of the transmission oil displaces the moving V-pulley halves, this changes the position of the steel push-belt between the two pulleys and with it the gear ratio. This technology allows continuous adjustment of the gear ratio without interrupting the power transmission and operation of the engine in its most favorable power range.

Benz patent motor carriage from 1886 with its machine parts (source: DaimlerChrysler Classic)





1 Engine
2 Belt drive to intermediate shaft
3 Bevel gear
4 Crankshaft with flywheel
5 Chain drive to powered axle

Transmission Requirements

Every motor vehicle places quite specific demands on its transmission. Each of the transmission types differ in terms of design and associated features. The objectives or main points of emphasis in transmission development can be divided into the following categories:

- comfort and convenience,
- fuel consumption,
- driveability,
- installation space, and
- production costs.

Comfort and Convenience

Essential requirements in terms of comfort and convenience are, in addition to a smooth gear change without engine-speed jumps, comfortable gearshifts irrespective of engine load and operating conditions, and a low level of noise. Nor should there be any loss of convenience over the entire lifetime of the transmission.

Fuel Consumption

The following transmission characteristics are essential in keeping fuel consumption as low as possible:

- large transmission-ratio range,
- high mechanical efficiency,
- "intelligent" shifting strategy,
- low power for control,
- low weight, and
- stand-by control,
  torque converter lockup clutch,
  low churning losses (resistance of the transmission oil passing through the gears), etc.

Driveability

The following transmission functions ensure good driveability:

- shifting points adapted to the relevant driving situation,
- recognition of the type of driver,
- high accelerating performance,
- engine braking action during downhill driving,
- suppression of gear change during cornering at high speed, and
- recognition of winter road conditions.

Installation Space

Depending on the type of drive, there are different stipulations for the space available:

Thus, the transmission for a rear-wheel drive should be as small as possible in terms of diameter, while the transmission for a front-wheel drive should be as low as possible in overall length. There are also precisely defined specifications for satisfying requirements in a crash test.

Production Costs

The preconditions for the lowest production costs possible are:

- production in large-scale numbers,
- simple control-system layout and automated assembly.

1 Manually shifted transmission (section, source: DaimlerChrysler)

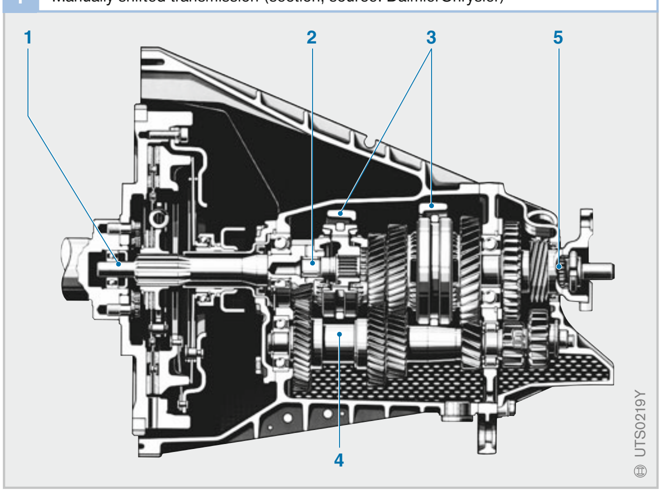


Fig. 1
1 Input shaft
2 Main shaft
3 Shifting elements
4 Countershaft
5 Output shaft

Manual Transmission

Application

Manually shifted transmissions are the simplest and most inexpensive transmissions for car drivers (final users). For this reason, they still dictate the market in Europe.

Due to increasing engine performance and higher vehicle weight together with decreasing c_d values, 5-speed manual transmissions have been superseding the previously dominant 4-speed manual transmissions since the beginning of the 1980s. And now the 6-speed transmission is virtually standard.

This development provided, on the one hand, safe starting and good acceleration and, on the other hand, lower engine speeds at higher road speeds, and thus reduced fuel consumption.

Design

The design of a manually shifted transmission (Figures 1 and 2) incorporates

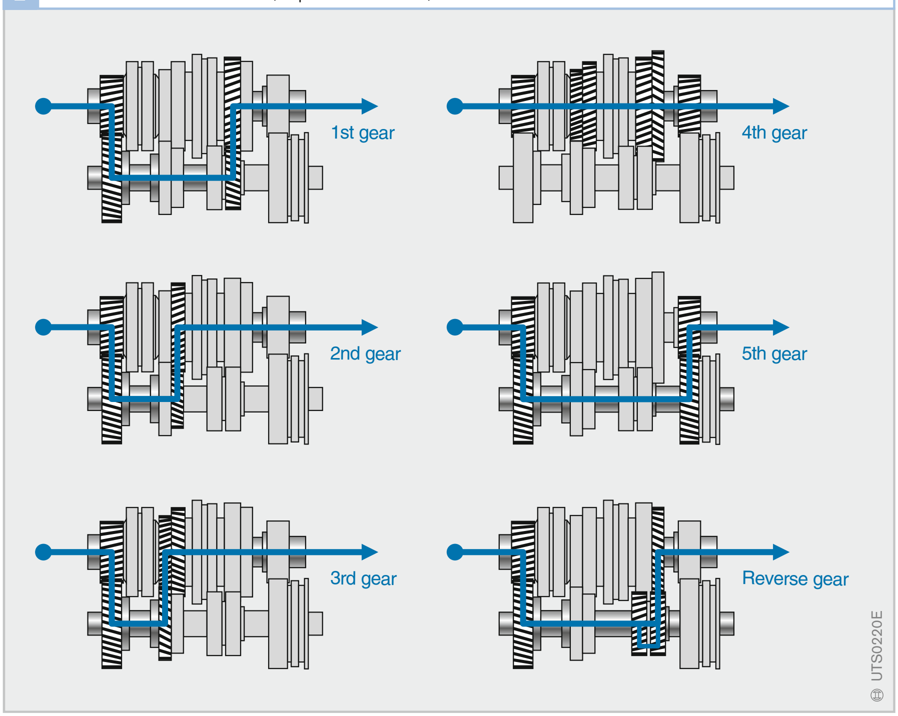
- a single-plate dry clutch as the power take-up element and for interrupting the power-flow during gear changes,
- gears mounted on two shafts,
- positive clutches as shifting elements, actuated via a synchronizer assembly.

Features

The main features of the manual transmission are:

- high efficiency,
- compact, light design,
- economical construction,
- absence of comfortable operation (clutch pedal, manual gear changing),
- driver-dependent shifting strategy,
- interruption of tractive force during gearshifting.

2 Power-flow in a standard drive (5-speed transmission)



Automated Shift Transmission (AST)

Application

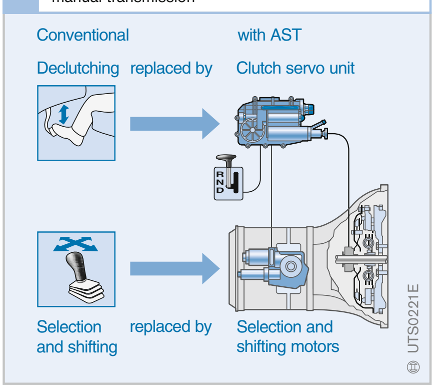
Automated shift transmissions (AST), or **a**utomated **m**anual **t**ransmissions (AMT), help to simplify transmission operation and increase economic operation. They are an add-on solution to conventional manual transmissions. The previously manual gearshifts are now performed by pneumatic, hydraulic, or electrical means. Bosch favors the electrical solution described below (Figure 1).

Design and Operating Concept

The AST is made possible by **e**lectronic **c**lutch **m**anagement (ECM), supplemented by two servomotors (selection and shifting motors) for selection and shifting. Depending on the system used, the required electrical control signals can be issued directly from a shift lever actuated by the driver or from an intermediate electronic control system.

Thanks to the electric-motor-driven actuators of the AST concept, it is possible at little expense to achieve automation of the transmission complete with the associated increase in convenience. An important argument for the transmission manufacturers here is the ability to reuse existing production facilities.

1 Automated shift transmission as add-on solution for manual transmission



In the simplest system, the mechanical linkage is merely replaced by a remote switch. The shift lever (tip lever or switch with H gearshift pattern) just outputs electrical signals. Power take-up and clutching are performed as in the manual transmission, partly linked to a gearshift recommendation.

In fully automatic systems, the transmission and power take-up element are automated. A lever or tip switch is the control element for the driver. The driver can skip the automatic facility with a manual setting or with +/– buttons. Automatically controlling a multispeed transmission requires a complex shifting strategy which also takes into account the present total running resistance (determined by load and road profile).

To support the synchronization process in the interruption of tractive force during the gearshift operation, an electronic engine-control facility (depending on the shift type) automatically closes the throttle briefly.

The design of automated shift transmissions is characterized by the following features:

- basic design as for manual transmissions,
- actuation of clutch and gear change by actuators (pneumatic, hydraulic or electric-motor-driven), and
- electronic control.

Features

The main features of the automated shift transmission are:

- compact design,
- high efficiency,
- adaptation to existing transmission possible,
- more competitively priced than automatic or CVT transmissions,
- simple operation,
- suitable shifting strategies for achieving optimum fuel consumption and best consumption figures, and
- interruption of tractive force during gearshifting.

Examples of AST in Series Production

AST Electric-Motor-Driven
Opel Corsa (Easytronic, Figure 2a),
Ford Fiesta (Durashift),

AST with Electromechanical Drum Transmission
Smart.

AST Electrohydraulic
DaimlerChrysler Sprinter (Sequentronic, Figure 2b),
BMW-M with SMG2,
Toyota MR2,
Ford Transit,
VW Lupo,
Ferrari, Alfa,
BMW 325i/330i.

2 Series examples of AST (sources: Opel, DaimlerChrysler)

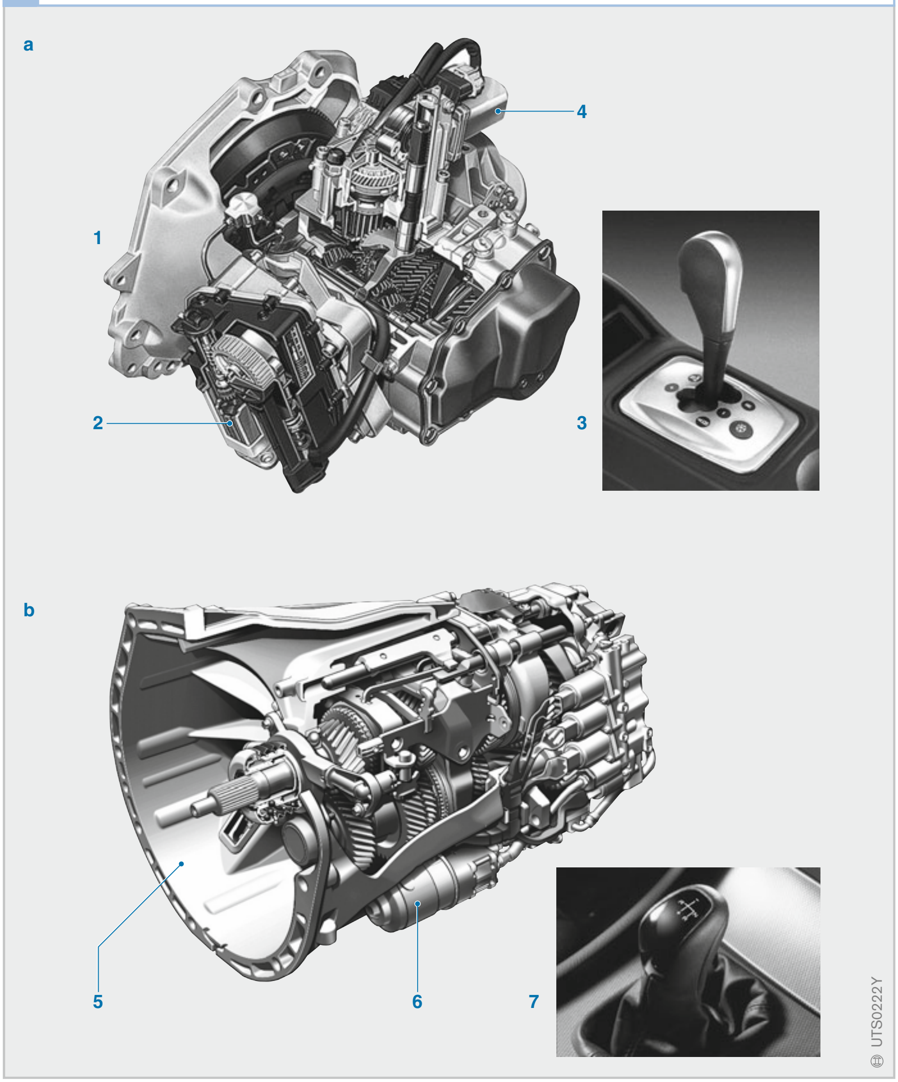


Fig. 2
- a Easytronic (Opel Corsa)
- b Sequentronic (DaimlerChrysler)
- 1 Transverse transmission
- 2 Clutch servo unit with integrated ECU
- 3 Tip lever
- 4 Shifting/ selection motor
- 5 Longitudinal transmission
- 6 Shifting/ selection motor
- 7 Shift lever

AST Components

The components of an AST must be able to withstand high loads in terms of temperature, leak-tightness, lifetime, and vibration. Table 1 lists the most important requirements.

Clutch Servo Unit

The clutch servo unit (Figures 4 and 5) with integrated electronic control unit (ECU) (Figure 3) serves to actuate the clutch. The entire AST function is also incorporated in the electronics. The clutch servo unit comprises

- integrated ECU,
- housing with cooling function,
- DC motor,
- helical gear,
- push rod, and
- return spring.

Table 1

1	Demands placed on AST components
Temperature	105°C permanent 125°C briefly Winding and commutation system
Leak-tightness	Steam jet Splash water Transmission fluid
Operating life	1 million shift cycles
Vibrations	7...20 *g* sine Armature mounting Electrical / electronic components Electronics PCB

3 Integrated ECU (view)

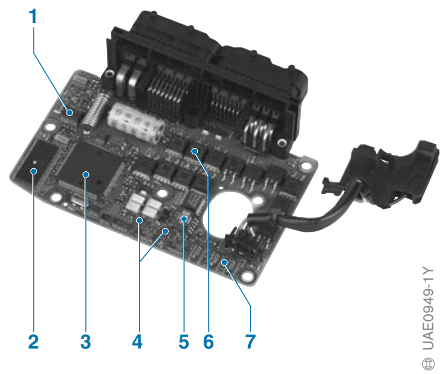


Fig. 3
1 Monitoring computer
2 Flash memory
3 Microcomputer (16-bit)
4 Travel-sensor contacts
5 DC converter
6 Driver stage for electric motors
7 Bridge driver

DC Motors for Gear Selection and Engagement

There are two types of DC motor for AST (Figures 5 and 6):

- The selector motor has a short response time.
- The shift motor has a high rotational force.

The transmission types for the selector motor and the shift motor can be set up symmetrically (left and right), and different mounting bores are also possible. The layout of the 6-pin connector can be chosen as desired. The motors with their aluminum housings

4 Clutch servo unit (section)

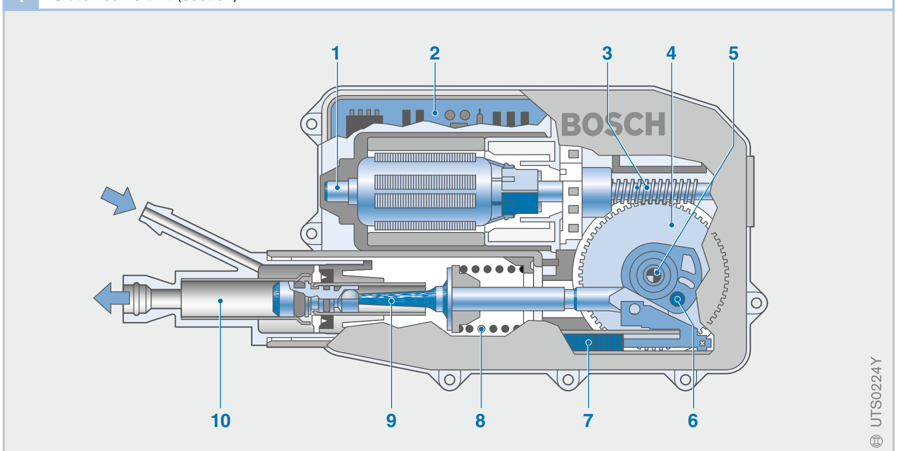


Fig. 4
1 Actuator motor
2 ECU
3 Worm
4 Worm gear
5 Worm-gear shaft
6 Pin
7 Position sensor
8 Compensation spring
9 Push rod
10 Master cylinder

5 Clutch servo unit with integrated ECU and DC motors for gear selection and engagement (view)

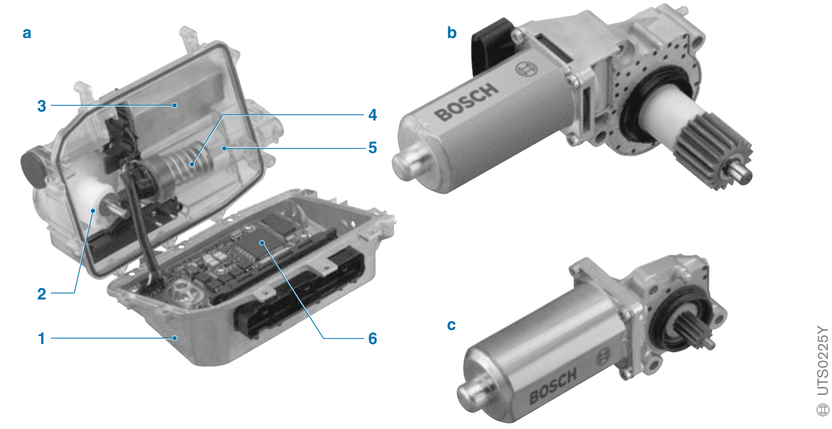


Fig. 5
a Clutch servo unit with integrated ECU
b Shifting motor
c Selection motor

1 Housing with cooling function
2 Helical gear
3 DC motor
4 Return spring
5 Push rod
6 Integrated ECU

are mounted directly on the transmission. They have a brush holder with integrated connector. This also contains an integrated double Hall-effect sensor (IC) with a resolution of 40 increments per engine revolution. A Hall-effect sensor with output channels for the rotor angle (lateral pulse) and the direction (high and low) can detect the position of the output shaft.

A 20-pin magnet on the rotor shaft allows a resolution of 9° per increment. In relation to the transmission step-up ratio, it is possible to obtain a resolution of between 0.59° per increment and 0.20° per increment at the output. Depending on the requirement, the gear has a crank or eccentric drive. The worm-gear pair has 1 to 4 teeth.

EC Motors

EC motors are brushless, permanently excited, electronically commutated DC motors and are used as an alternative to the DC motors. They are equipped with a rotor-position sensor, supplied with direct current via control and power electronics (Figure 7), and characterized by long lifetime and the minimal space they take up.

6 DC motor (section)

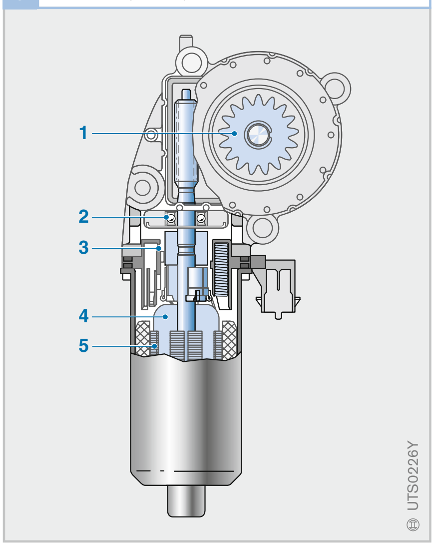


Fig. 6
1 Solid pinion for manual-transmission intervention
2 Armature bearing with pressed-on ball bearing (axial lock with clamp)
3 20-pin ring magnet and double Hall sensor
4 Vibration-resistant winding
5 Narrow armature shape for high dynamics

7 EC motor (schematic)

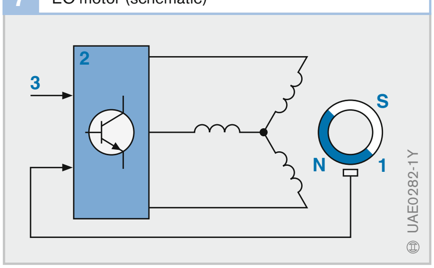


Fig. 7
1 Electrical machine with rotor position sensor
2 Control and power electronics
3 Power supply

Dual-Clutch Transmission (DCT)

Application

Dual-clutch transmissions, DCT (Figure 1), are seen as the further development of the AST. They operate without interruption of tractive force, a major drawback of the AST.

The DCT's main benefit lies in its lower fuel consumption compared with automated shift transmissions.

The dual-clutch transmission was used for the first time in 1992 in motor racing (Porsche). However, owing to the high computation effort required in the control system to ensure a comfortable overlapping gearshift, it failed to make it into mass production.

With the availability of high-power computers, several manufacturers (e.g. VW, Audi) are now working on introducing dual-clutch transmissions for mass production.

Since its requirements profile matches that of an automatic transmission in terms of convenience and functionality, it has found its niche in the superior, luxury vehicle categories.

Dual-clutch transmissions also match the wishes of vehicle manufacturers for modular concepts since both manually shifted and automated shift transmissions can be manufactured on the same production line.

Design

The design of dual-clutch transmissions is characterized by the following features:

- basic design as for manual transmissions,
- gears mounted on three shafts,
- two clutches,
- actuation of clutch and shifting elements via transmission-shift control and actuators.

1 Dual-clutch transmission, DCT (cutaway view, source: VW)

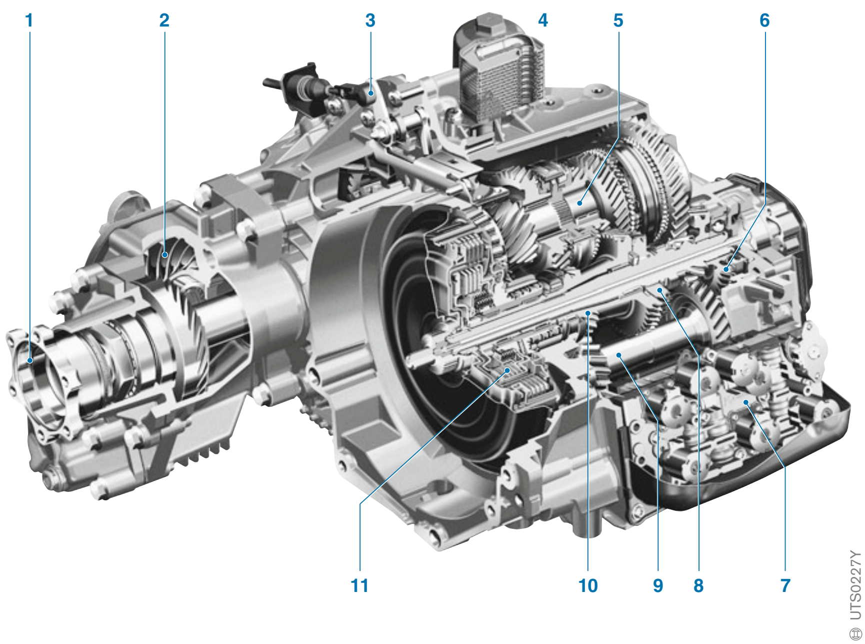


Fig. 1
1 Output for right front wheel
2 Bevel-gear drive for rear axle
3 Parking lock
4 Oil cooler
5 Output shaft 1
6 Input shaft 2
7 Mechatronic module
8 Input shaft for oil pump
9 Return shaft
10 Input shaft 1
11 Dual clutch

Operating Concept

The dual-clutch transmission operates as follows:

The gear wheels assigned to the gear steps are divided into groups of even and uneven gears. Although it is similar in terms of basic design to a conventional manual countershaft transmission, there is one crucial difference: Even the main shaft is split – namely into a solid shaft and a surrounding hollow shaft, both coupled to a gear train.

Each partial shaft is assigned its own clutch at the transmission input. Since now two gears are engaged during the gear change (both the active gear and the neighboring, preselected gear), a faster change between the gears is thus possible. In this way, the gear change can take place between the two partial transmissions, in the same way as in an automatic transmission, without interruption of tractive force (Figure 2).

Features

The main features of the dual-clutch transmission are:

- similar levels of convenience to an automatic transmission,
- high efficiency,
- no interruption of tractive force during gearshifting,
- skipping of a gear possible,
- takes up more space than an AST,
- high bearing forces, solid construction.

2 Dual-clutch transmission, operating principle with power-flow when accelerating in 1st gear (source: VW)

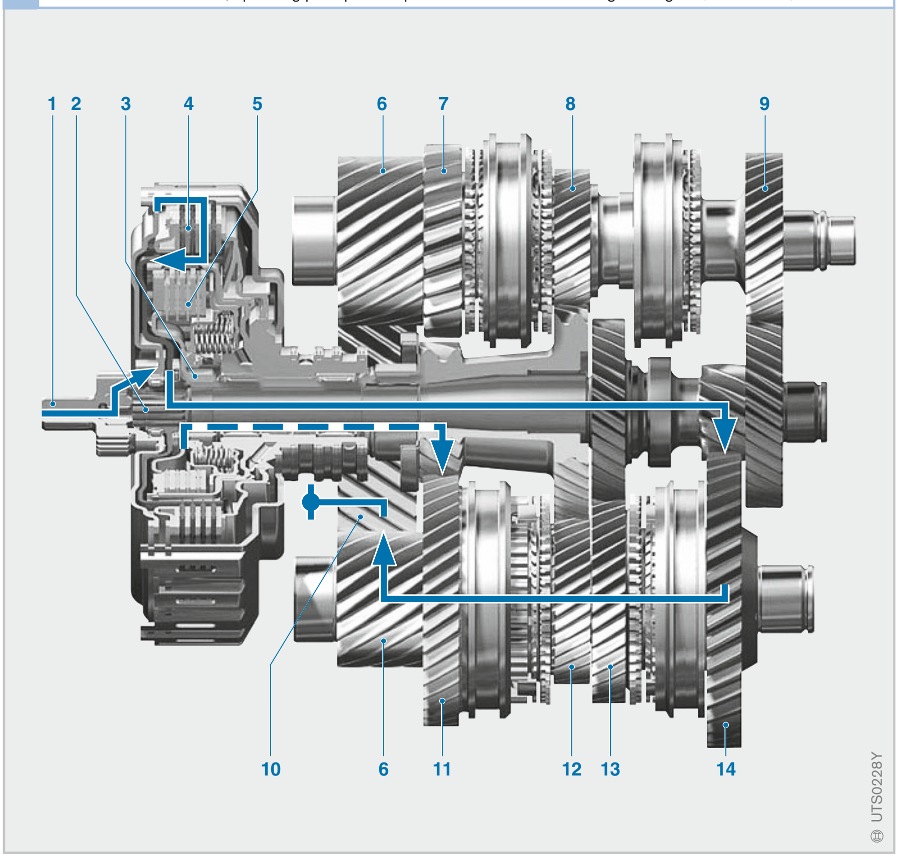


Fig. 2

1 Engine drive
2 Input shaft 1
3 Input shaft 2
4 Clutch 1 (closed)
5 Clutch 2 (open)
6 Output to differential
7 Reverse gear
8 6th gear
9 5th gear
10 Differential
11 2nd gear (preselected)
12 4th gear
13 3rd gear
14 1st gear (active)

Automatic Transmission (AT)

Application

Automatic power-shift transmissions, or simply automatic transmissions (AT) perform power take-up, select the gear ratios, and carry out the gearshifts themselves. A hydrodynamic converter acts as the power take-up element.

Design and Operating Concept

Transmission with Ravigneaux Planetary-Gear Set

The four-shaft planetary-gear set known as the Ravigneaux set is the basis of many automatic 4-speed transmissions. Figure 1 shows the schematic, the shifting logic, and a speed-ladder diagram for this transmission. The transmission schematic clearly shows the layout of the gear wheels and shifting elements.

Sun gears B, C and planetary-gear carrier S can be connected via clutches CB, CC and CS to shaft A, which is guided by the converter turbine into the transmission. Shafts S and C can be connected with the aid of brakes BS and BC to the transmission housing.

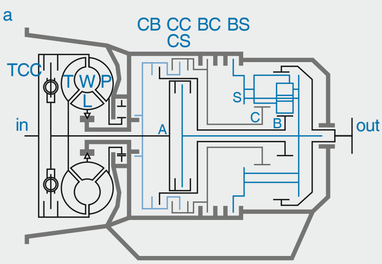
A planetary-gear set of this type has a kinematic degree of freedom of 2. This means that, when two speeds are specified, all the other speeds are established. The individual gears are shifted in such a way that via two shifting elements the speeds of two shafts are defined either as drive speed n_{dr} or as housing speed $n_C = 0$ rpm.

The speed-ladder diagram clearly shows the speed ratios in the transmission. The speeds are entered upwards on the speed ladders for the individual shafts of the overlapping or shift transmission. The gaps of the speed ladders are derived from the gear ratios or numbers of teeth such that the speeds belonging to a particular point of operation can be connected by a straight line.

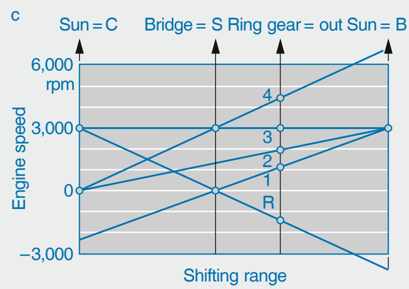
At a specific drive speed, the five reference lines characterize the speed ratios in four forward gears and one reverse gear.
Only the three shafts B, C and S between the input shaft "in" (corresponding to A) and the output shaft "out" are available for the different gearshifts. All three shafts can be con-

1 4-speed automatic transmission based on Ravigneaux planetary-gear set

a



c



b

1	2	3	
1	**2**	3	4
1	2	**3**	4
	2	3	**4**

Gear steps with simple gearshifts

Gear	CC	CS	CB	BS	BC	i_{tot}
R	●			●		–2.550
N						
1			●	●		2.800
2			●		●	1.508
3	●	●	●			1.000
4		●			●	0.718

UTS0229E

Fig. 1
a Transmission schematic
b Shifting logic
c Speed-ladder diagram

nected to input shaft A, but then constructively only two shafts can still be connected to the transmission housing.

The simultaneous shifting of two brakes is not useful for gearshifts as this blocks the transmission. Of equally little use is the simultaneous connection of one shaft to the housing and to the input shaft. The simultaneous shifting of two clutches always results in the direct gear ($i = 1$).

This retains exactly the five gears shown in the shifting logic and in the speed diagram. Beyond the numbers of teeth that are possible within the framework of the installation conditions, the designer still has the option of changing the individual gear ratios, where a direct gear is always specified with $i = 1$.

Finally, these transmissions still make it possible with simple gearshifts to skip gears by cutting in one shifting element and cutting out another shifting element. It is possible to shift from 1st gear into 2nd or 3rd gear, and from 4th gear into 3rd or 2nd gear. From 2nd and 3rd gear, all other gears can be reached with simple gearshifts.

However, it is not possible to shift more than four forward gears with the Ravigneaux set. An automatic transmission with five gears therefore requires either another basic transmission or a front-mounted or rear-mounted range-change unit to expand the Ravigneaux set. But such an expansion stage requires at least two shifting elements.

An example of this is the 5HP19 automatic transmission from ZF, which has three clutches, four brakes, and a one-way clutch to shift only five forward gears.

Obviously, it is also possible to attain more than 5 gears with range-change units but shifting effort then becomes increasingly more pronounced and gearshifts of several shifting elements for one gear change cannot be avoided.

Transmission with Lepelletier Planetary-Gear Set

A more elegant way of shifting five and more gears was devised by the French engineer Lepelletier. He expanded the Ravigneaux set to include a range-change transmission for only two shafts of the Ravigneaux set in order to drive them with means other than the drive speed.

The unusual feature of the Lepelletier planetary-gear set as set out in Figure 2 (following page) lies in the fact that the additional three-shaft planetary-gear set reduces the speed of shaft D in respect to that of shaft A. In the first three gears of this 6-speed automatic transmission, the shifting logic corresponds to the logic of the 4-speed Ravigneaux set. The gear ratios are greater by the orbit ratio of the internal gear to the carrier at the fixed sun gear of the additional planetary-gear set.

In 4th and 5th gears, shaft S is connected via clutch KS to shaft A. It rotates faster than shafts B and C. The transmission ratios are produced from the gearshifts in 4th gear: S = A and B = D and in 5th gear S = A and C = D. Without the additional transmission from A to D, the gear ratios in 3rd, 4th and 5th gears would be identical and all $i = 1$.

The 6th gear of this 6-speed automatic transmission corresponds in terms of the shifting logic again to the 4th gear of the 4-speed automatic transmission. Even the gearshifts of the reverse gears are identical in these 4-speed and 6-speed automatic transmissions.

The 6-speed automatic transmission (Figure 3) also makes possible wide gear steps with simple gearshifts, which can be necessary particularly in the case of rapid downshifts.
The Lepelletier planetary-gear set therefore differs from the Ravigneaux set only in that it has an additional planetary-gear set with a fixed gear ratio. The number of shifting elements remains the same. They are simply used repeatedly for the additional gears. In terms of space, weight, and cost, this transmission is more suitable than a 5-speed automatic transmission. With the numbers of teeth shown in Figure 2, this 6-speed automatic transmission achieves a setting range of $\varphi = 6$ with easily shiftable gear spacings.

The additional planetary-gear set consists of sun gear E, internal gear A, and planetary-gear carrier D and is used in reverse gear and the first 5 gears as a fixed ratio stage. Shaft E is permanently connected as reaction member to the transmission housing. If this connection were removed and replaced by an additional brake BE, the vehicle could be started with this brake instead of the converter.

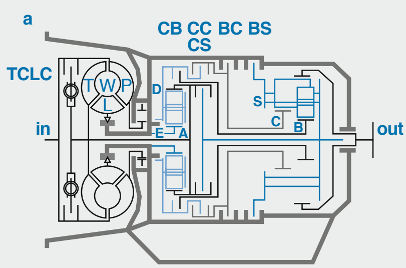
Power Take-Up Elements

In the majority of automatic transmissions which are geared towards convenience, a hydrodynamic converter adopts the power take-up function. A hydrodynamic converter is an ideal power take-up element because of the way it works as a turbo element. In order to minimize converter losses during vehicle operation, it is however (as often as is possible) locked up with the torque converter lockup clutch (TCLC).

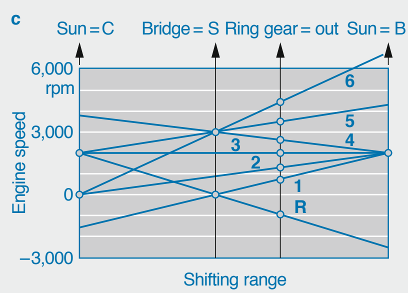
When used with very high-torque turbodiesel engines, the converter can no longer be designed to achieve optimum results for all operating states. A drive of this type requires a relatively soft converter characteristic for safe starting in cold conditions. The maximum pump torque may only have an effect at high speeds so that the drag losses do not stall the weak engine without suffi-

2 6-speed automatic transmission based on Lepelletier planetary-gear set

a



c



b

1	2	3	4		
1	**2**	3	4		
1	2	**3**	4	5	
	2	3	**4**	5	6
		3	4	**5**	6
			4	5	**6**

Gear	CC	CS	CB	BS	BC	i_{tot}
R	■			■		–3.400
N						
1			■	■		4.171
2			■		■	2.340
3	■		■			1.521
4		■	■			1.143
5	■	■				0.867
6		■			■	0.691

UTS0231E

Fig. 2
a Transmission schematic
b Shifting logic
c Speed-ladder diagram

cient charge-air pressure. At normal operating temperatures and at speeds at which sufficient charge-air pressure is available, a hard converter characteristic with a steep rise in pump torque as engine speed increases is advantageous.

Series applications with fast and accurate pressure control now also make it possible to start up comfortably with friction clutches. A good example of this is the Audi A6 with the continuously variable Multitronic transmission.

Pressure control and heat dissipation can be better achieved with a brake than with a clutch. It should therefore also be possible to obtain comfortable starting with the brake. Even during the gear changes, a slipping brake can remove the load from the other shifting elements in the same way as a converter.

Automatic Transmission Fluid (ATF)

Automatic transmissions place exacting demands on the ATF (**a**utomatic **t**ransmission **f**luid):

- increased pressure-absorption capability,
- good viscosity-temperature characteristics,
- high resistance to aging,
- low foaming tendency,
- compatibility with sealing materials,

These requirements must be guaranteed in the oil pan in a temperature range of –30...+150°C. Temperatures of up to 400°C are briefly and locally possible between the clutch plates during a gearshift.

The transmission fluid is specially adapted for fault-free operation of the automatic transmission. A series of chemical substances (additives) is added to the basic oil for this purpose. The main additives are:

- friction modifiers, which influence the frictional behavior of the shifting elements,
- antioxidants for reducing thermooxidative aging at high temperatures,
- dispersants for preventing deposits in the transmission,
- foam inhibitors for preventing the buildup of oil foam,
- corrosion inhibitors for preventing corrosion of transmission components when condensation water is formed, and
- seal-swell agents, which control the swelling of sealing materials (elastomers) under the influence of oil to defined levels.

3 ZF 6-speed 6HP26 automatic transmission (source: ZF Friedrichshafen)

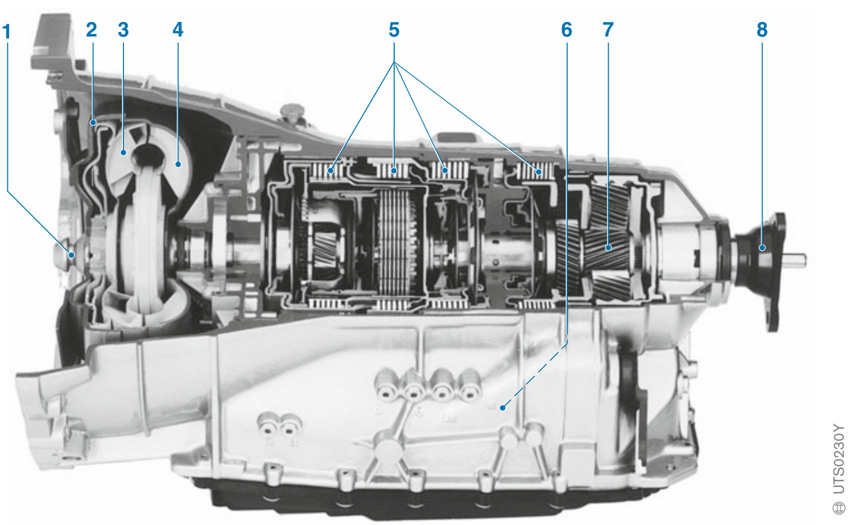


Fig. 3
1 Transmission input from engine
2 Torque converter lockup clutch
3 Turbine
4 Converter
5 Multiplate clutches
6 Module for transmission control
7 Planetary-gear set
8 Transmission output

GM established the first specification for ATF back in 1949. Typical technical data for SAE viscosity classes in accordance with DIN 51 512 are:

Flash point		(> 180°C)
Pour point		(< –45°C)
Viscosity index		(VI > 190)
Kin. viscosity:	37 cSt	(at +40°C)
	7 cSt	(at +100°C)
Dyn. viscosity:	17 000 cP	(at –40°C)
	3 300 cP	(at –30°C)
	1 000 cP	(at –20°C)

In the meantime, automatic transmissions are increasingly being filled with lifetime fluid, thus rendering unnecessary a change of fluid.

Oil Pump

The transmission requires an oil pump (Figure 4) to build up a control pressure. This pump is driven by the engine. At the same time, the oil-pump drive power reduces the transmission efficiency. The following equation applies here:

Pump output = pressure × flow

Figure 5 shows the output characteristics of a gear pump (1) and a radial piston pump (2) in comparison. Possibilities for optimization in the oil-pump range are offered by a variable flow or a controllable pump pressure:

Variable Pump Flow

The particular features of variable pump flow are as follows:

- The design creates a sufficiently high flow to fill the clutch at idle speed.
- An additional displacement at higher speeds causes a loss of output.
- The variable-capacity pump enables the pump output to be adapted as required.

- However, variable pump flow has the drawback of being expensive and susceptible to failure.

Controllable Pump Pressure

The particular features of controllable pump pressure are as follows:

- The pump pressure is adapted to the torque to be transferred in each case.
- Main-pressure control allows effective operation (by means of the actuator) close to the clutch slipping point.

4 Crescent oil pump (section)

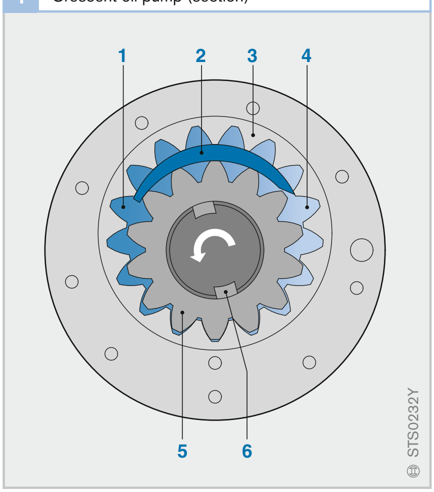


Fig. 4
1 Pressure outlet
2 Crescent
3 Internal gear
4 Suction side
5 External gear, driven by engine
6 Drive lug

5 Oil pumps (pump outputs in comparison)

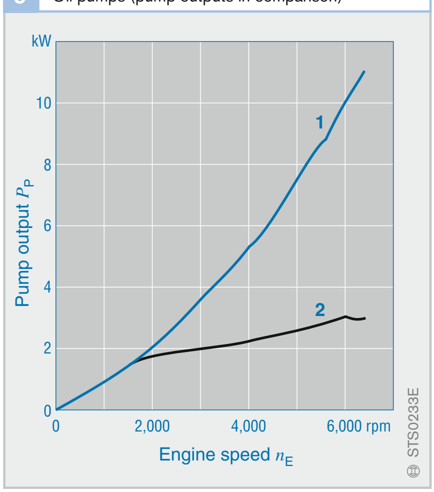


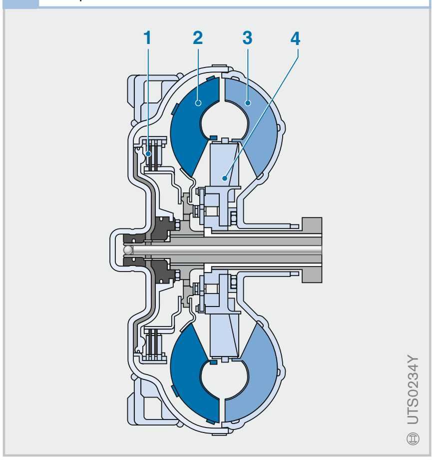
Fig. 5
1 Gear pump
2 Radial piston pump

Torque Converter

The torque converter (Figure 6) is a power take-up aid, which works as an additional gear in the start-up range and also serves to damp vibrations. It was the hydraulic converter with centripetal turbine which first enabled automatic transmissions to be used in passenger cars. The most important elements of a converter are:

- pump (driven by the engine),
- turbine,
- stator on the one-way clutch, and
- fluid (for the transfer of torque).

6 Torque converter (section)



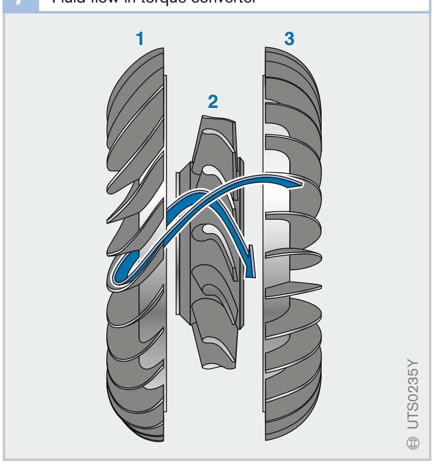
The impeller sets the fluid from the hub in motion in an outward direction. There the fluid hits the turbine, which directs it inwards. The fluid from the turbine in the hub area then hits the stator, which diverts it back to the pump (Figure 7).

In the converter area ($\nu < 85\%$), the turbine torque is increased by the reaction torque at the stator. In the clutch area, the stator one-way clutch is released and there is no further increase in torque. The maximum efficiency is $< 97\%$ (Figure 8).

A transmission of power via the converter can only take place when slip occurs between the impeller and the turbine wheel. This is low in most vehicle operating states, ranging from 2...10%. However, this slip causes a loss in power output and thereby increased vehicle fuel consumption. A torque converter lockup clutch must therefore always cut in when the converter is not required for start-up or torque conversion (see also section "Controlled Torque Converter Lockup Clutch"). This is a multiplate clutch, which connects the impeller to the turbine wheel by frictional locking.

Fig. 6
1 Torque converter lockup clutch
2 Turbine wheel
3 Impeller
4 Stator

7 Fluid flow in torque converter



8 Torque converter (characteristic)

Fig. 7
1 Turbine wheel
2 Stator
3 Impeller

Multiplate Clutches

Multiplate clutches (Figure 9) facilitate shifting without an interruption of tractive force and support the torque in the gear in which they are being actuated.

The lining and steel plates of the clutches and brakes take on the dynamic torque and the shifting energy during the gearshift and the load torque to be transferred after the gearshift. In order to ensure high gear-shift sophistication (convenience), the friction linings must demonstrate coefficients of friction which are as constant as possible and not dependent on temperature and load.

The friction linings in automatic transmissions have a cellulose support structure (paper linings). Added aramide fibers (high-strength plastic) ensure temperature stability. Further constituents are mineral aggregates, graphite, or friction particles for influencing the coefficient of friction. The entire lining is soaked in phenol resin to give it its mechanical strength. The steel plates are made of cold-rolled sheet steel.

The friction process occurs in the fluid layer between the lining and steel plates. The friction lining maintains this fluid layer through its porosity and by supplying cooling fluid.

The following problems can occur in connection with multiplate clutches:

- combustion at high temperatures,
- supply of fluid with rotating clutches, and
- pressure increase caused by rotation speed.

Planetary-Gear Sets

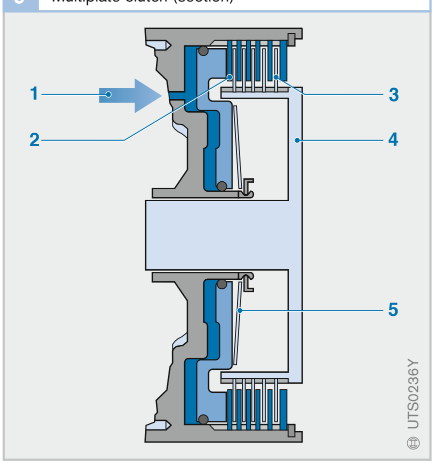
The planetary-gear set (Figure 10) is the heart of the automatic transmission. Its function is to adjust the gear ratios and to ensure constant power transmission. A planetary-gear set is made up of the following components:

- A central gear wheel (sun gear).
- Several (usually three to five) planet gears, which can rotate around their own axes and also around the sun gear. The planet gears are held in place by the planetary-gear carrier, which can rotate around the central axis.
- An internal gear, which surrounds and encloses the planet gears. This internal gear can also rotate around the central axis.

Planetary-gear sets are used in automatic transmissions for the following reasons:

- The power density of planetary-gear sets is very high since the power is transmitted in parallel via several planet gears. Planetary-gear sets are therefore highly compact and low in weight.

9 Multiplate clutch (section)



10 Planetary-gear set (schematic)

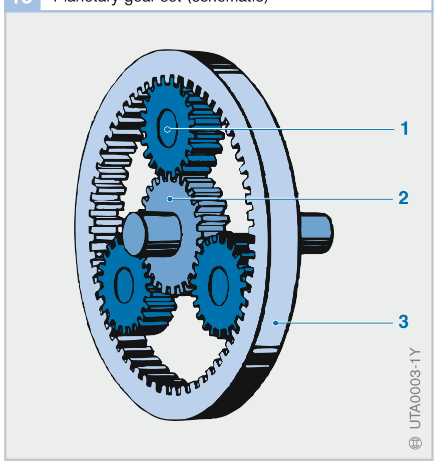


Fig. 9
1 Fluid feed
2 Outer plate
3 Lining plate
4 Plate carrier
5 Return spring

Fig. 10
1 Planetary-gear carrier with planet gears
2 Sun gear
3 Internal gear

- No free radial forces occur in the planetary-gear set. Rolling bearings can be replaced by cost-effective plain bearings.
- Multiplate clutches, multiplate brakes, band brakes, and one-way clutches can be arranged concentrically to the planetary-gear set, thus providing more space for the hydraulic control system.

Different planetary-gear set combinations are used in transmissions:

- Simpson (3-speed, two systems),
- Ravigneaux (4-speed, two systems),
- Wilson (5-speed, two systems).

Two types of planetary-gear set have been successfully used in automatic transmissions and are characterized by the following easy-to-distinguish features:

Ravigneaux Set

In the Ravigneaux set (Figure 11), two different planetary sets and sun gears operate in a single internal gear.

Simpson Set

In the Simpson set (Figure 12), two planetary sets and internal gears run on one joint sun gear.

Parking Lock

The function of the parking lock (Figure 13) is to secure the vehicle against rolling off. Its reliable operation is therefore fundamental to safety.

The driver must press the brake pedal before the selector lever can be moved from the P (Park) position. This mechanism prevents the vehicle from being set in motion by accidental operation of the selector lever.

11 Ravigneaux set (schematic)

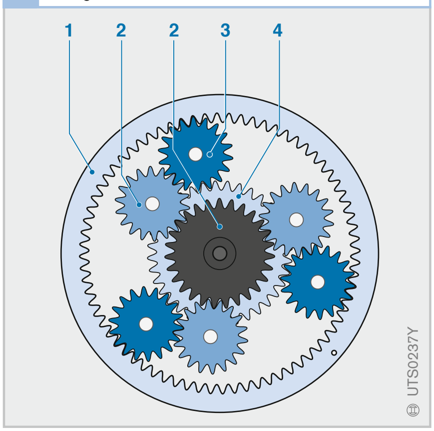


Fig. 11
1 Internal gear
2 Sun gear and planetary-gear set 1
3 Planetary-gear set 2
4 Sun gear 2

12 Simpson set (schematic)

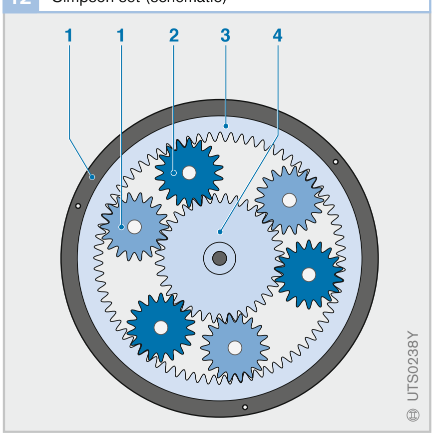


Fig. 12
1 Planetary-gear set 1 and internal gear 1
2 Planetary-gear set 2
3 Internal gear 2
4 Sun gear

13 Parking lock

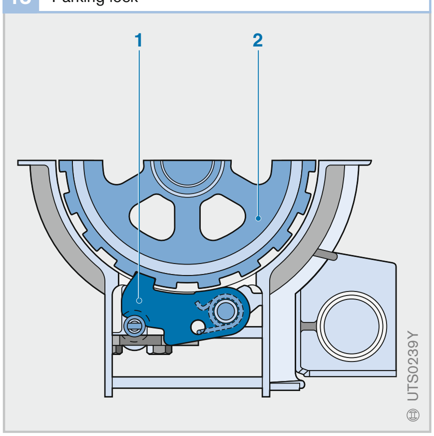


Fig. 13
1 Pawl
2 Parking-lock gear

Continuously Variable Transmission (CVT)

Application

Drive concepts with continuously variable transmissions (CVT) are characterized by high driving convenience, outstanding ride characteristics, and low fuel consumption.

VDT (Van Doorne's Transmissie) has for many years specialized in developing CVT components and prototype transmissions. Since its takeover of VDT in 1995, Bosch now covers the entire field of CVT system developments through to complete drivetrain-management systems. All the continuously variable automatic transmissions listed in Table 1 are operated with a push-belt (Figure 1). One exception is the Audi Multitronic with a link-chain manufactured by LuK (Figure 2).

The main components of a CVT can be activated by an electrohydraulic module. In addition to the push-belt – in mass production since 1985 – pulleys, pumps, and electrohydraulic modules are being developed for volume production launch. There are different types of push-belt for mid-range engine torques up to 400 Nm (e.g. Nissan Murano V6 with 3.5 *l* displacement and max. 350 Nm at 4000 rpm, with converter).

The know-how within the Bosch Group provides the software for optimum CVT activation. Naturally there is full flexibility with regard to software sharing so that vehicle manufacturers can also develop and implement special functions themselves.

1 Current availability (worldwide) of vehicles with CVT

Vehicle manufacturer	CVT designation	Vehicle
Audi	Multitronic	A4, A6
BMW	CVT	Mini
GM	CVT	Saturn
Honda	Multimatic	Capa, Civic, HR-V, Insight, Logo
Hyundai	CVT	Sonata
Kia	CVT	Optima
Lancia	CVT	Y 1.2l
MG	CVT	F, ZR, ZS
Mitsubishi	CVT	Lancer-Cedia, Wagon
Nissan	Hyper-CVT ICVT Extroid-CVT	Almera, Avensis, Bluebird, Cube Micra, Murano, Primera, Serano, Tino, Cedric Gloria
Rover	CVT	25/45
Subaru	ICVT	Pleo
Toyota	Super-CVT Hybrid-CVT	Previa, Opa Prius

1 CVT for front-wheel drive, transverse (section)

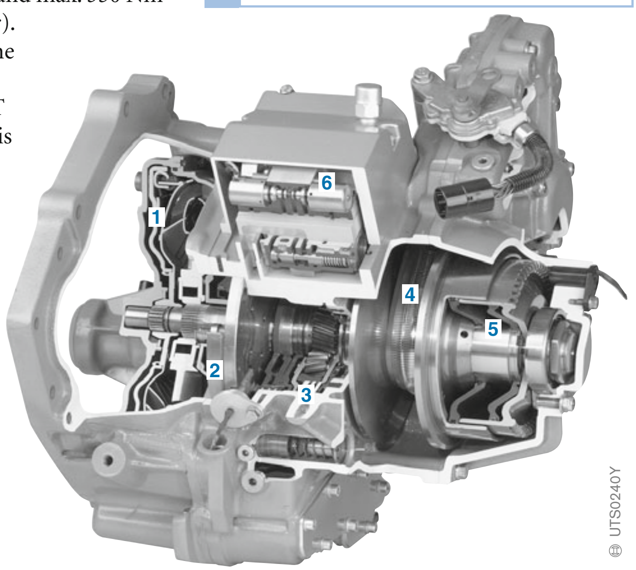
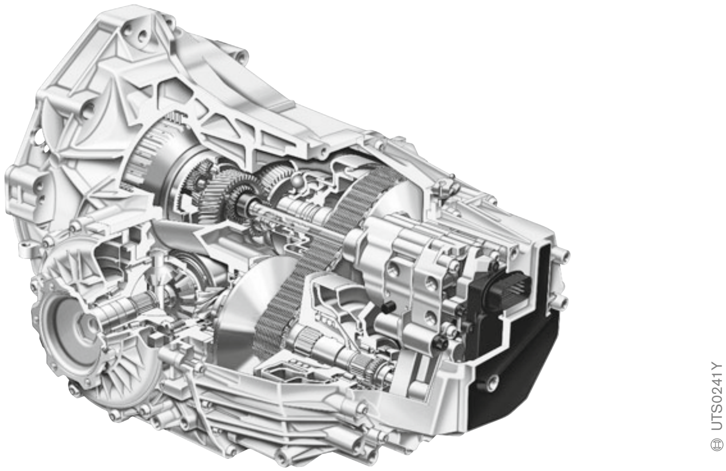


Fig. 1
1 Torque converter
2 Pump
3 Planetary-gear set with forward/reverse clutch
4 Push-belt
5 Variator
6 Control module

2 CVT for front-wheel drive, longitudinal (Audi Multitronic with link-chain, source: Audi)



A distinction is drawn within the CVT functions between a basic functionality and an expansion stage. All the functions of the first group have already been implemented and tested and are in use in various vehicles.

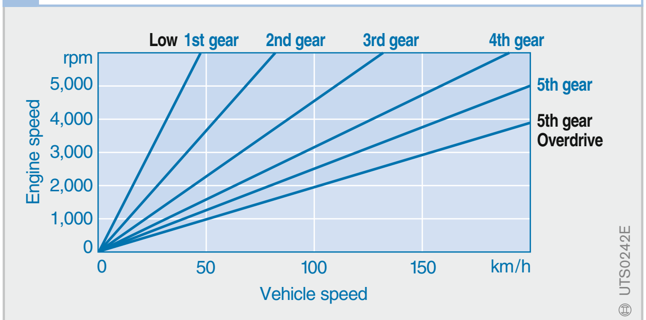
Suitable tools for efficient representation and testing such as ASCET-SD are available and used in joint projects.

The large gear-ratio span of continuously variable transmissions moves the engine operating point into ranges that are favorable to fuel consumption.

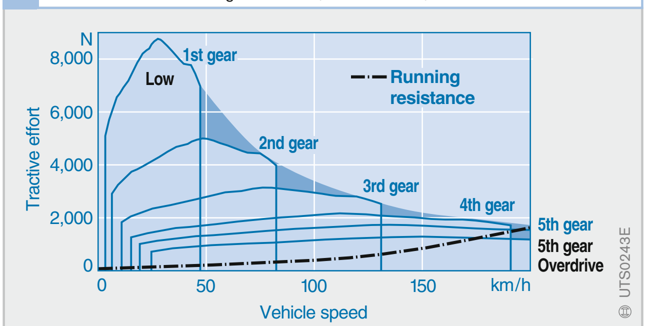
Starting out from the span shown in Figure 3, this produces the allocation of the tractive force to the gear ratio shown in Figure 4.

Conflicting requirements can be satisfied with the aid of electronic control and appropriate prioritization.

3 Span of a CVT transmission compared with 5-speed automatic (characteristic)



4 Tractive force and running resistance (characteristics)



5 Mechanical variator adjustment (schematic)

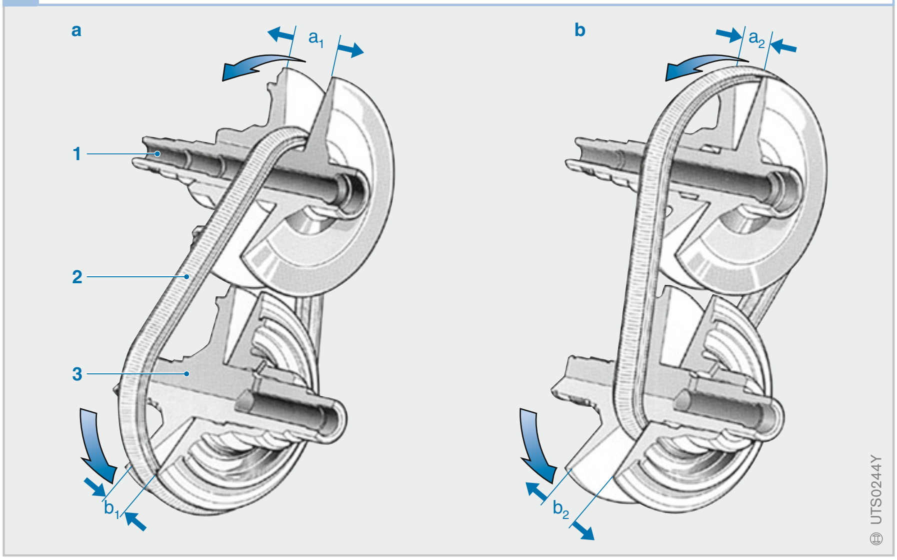
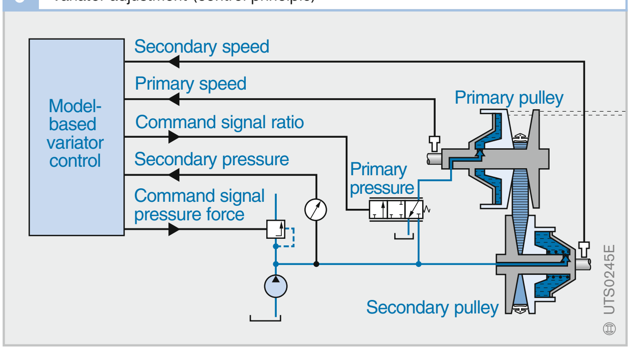


Fig. 5
a “Low” ratio
b “Overdrive” ratio

1 Input (primary) pulley
2 Push-belt or chain
3 Output (secondary) pulley

a_1, b_1 “Low” ratio
a_2, b_2 “Overdrive” ratio

6 Variator adjustment (control principle)



7 Model-based variator adjustment

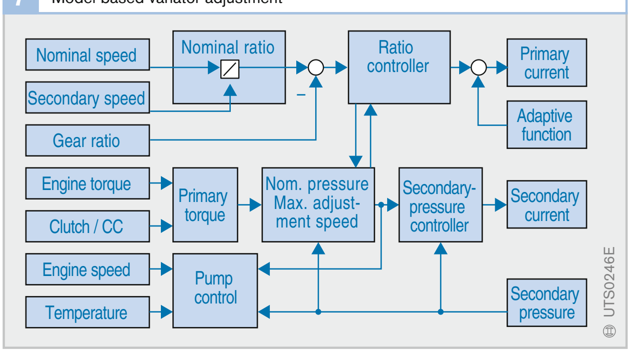


Figure 5 shows the mechanical adjustment of the gear ratio from “low” to “overdrive”. The control setup shown in Figure 6 is used for this purpose.

The model-based variator control system featured in Figure 7 processes the following operations:

- Adjustment of primary speed or gear ratio with PI controller.
- Adjustment of contact pressure for the primary and secondary pulleys.
- Coupling of the control of gear ratio and contact-pressure control and control of the pump.
- Adaptive function for compensating tolerances.

Design

The converter or the multiplate clutch acts as the power take-up element, and the reverse gear is shifted via a planetary-gear set.

The gear ratio is continuously varied with V-pulleys and a belt or a chain (variator).

High-pressure hydraulics provide the necessary contact pressure and variator adjustment.

All the functions are controlled by the electrohydraulic control system. The various components of the CVT transmission are depicted in Figure 8.

Features

One advantage of CVT transmissions is that they do not cause any interruption of tractive force when the gear ratio is changed. These transmissions offer a high level of convenience since gearshifts are not necessary.

In the entire engine map, operation is matched to optimum fuel consumption/ maximum acceleration. A high ratio span is also possible.

Although the high-pressure pump requires a certain level of drive power, the overall efficiency is satisfactory.

8 Model-based variator control

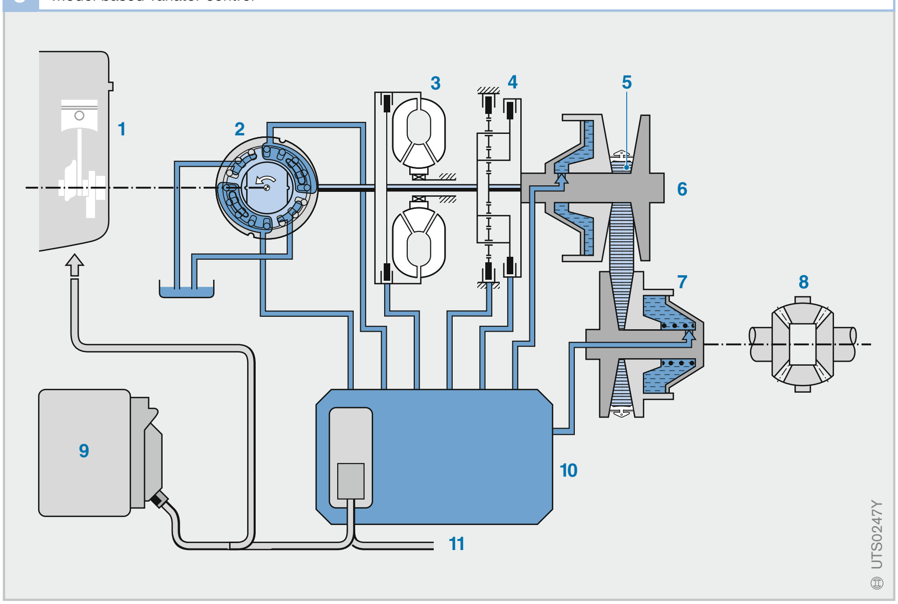


Fig. 8
1 Engine
2 Pump
3 Converter
4 Planetary-gear set
5 Push-belt
6 Input (primary) pulley
7 Output (secondary) pulley
8 Differential
9 Electronic engine management
10 Electrohydraulic module (hydraulic valves, sensors, actuators)
11 Vehicle wiring harness

CVT Components

Variator

The variator consists of two V-pulleys which move in relation to each other (Figures 9 and 10).

The pressure *p* of the transmission fluid moves the moving parts of the variator (1) in relation to each other. This alters the position of the push-belt (3) between the two pulleys and changes the gear ratio.

As power transmission is based solely on the friction between the belt and the variator, this type of adjustment requires a high system pressure.

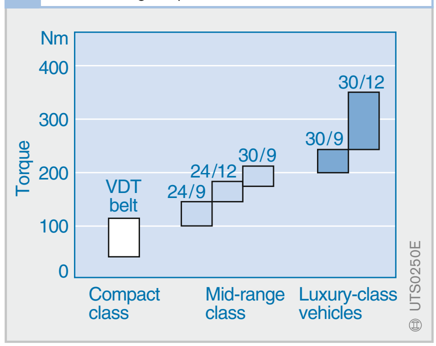
Push-Belt

The company Van Doorne's Transmissie holds a worldwide patent for the push-belt. Figure 11 shows the different types of belt and their areas of application in relation to the engine torque to be transferred.

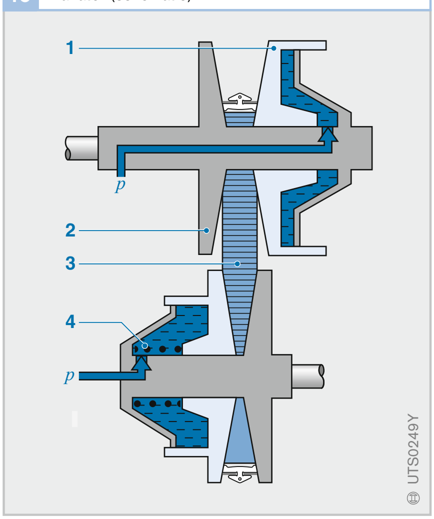
The push-belt (Figure 12) consists of push elements 2 mm thick and 24...30 mm wide, which are arranged at an inclination angle of 11° to each other. The chain is held by two packs, each with 8 to 12 steel belts. The coefficient of friction of the chain is at least 0.9.

9 Variator (view)

11 Product range of push-belts



10 Variator (schematic)



12 Push-belt (view with excerpt)

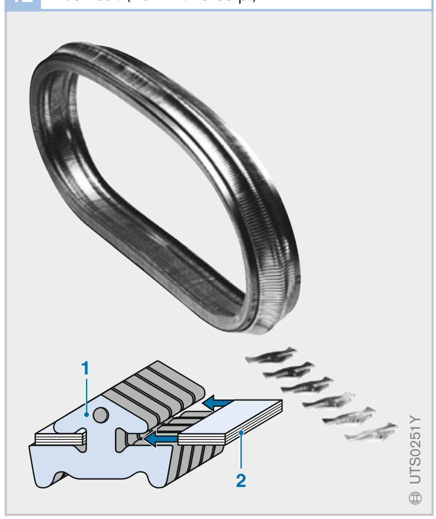


Fig. 10
1 Moving pulley
2 Fixed pulley
3 Push-belt
4 Spring
p Pressure of transmission fluid applied

Fig. 12
1 Push element
2 Steel-belt pack

The following nomenclature is used for the belt designations:

24/12/1.5/208.8
- Belt diameter
- Thickness of belts
- Number of belts
- Width of push elements in mm

Link-chain

Instead of the push-belt usually used in CVT transmissions, Audi uses a link-chain manufactured by the company LuK in its Multitronic transmission (this chain is based on the pin chain manufactured by the company P.I.V. Reimers).

This link-chain is made completely of steel and yet is almost as flexible as a V-belt. It is composed of various positions of links next to each other and therefore of such robust design that it can transfer very high torques (transferable engine torque 350 Nm) and forces.

The chain (Figure 13) consists of 1025 individual links, each with 13...14 chain links lined up next to each other. Pins with a width of 37 mm and an inclination angle of 11° connect the links (1) to each other. The ends of the pins (2) press against the conical surfaces in the variator.

The tractive force of the chain is transferred to the variator pulleys at the support points created. The mini slip created in the process is so minimal that the pins are subject to wear of no more than one to two tenths of a millimeter over the entire working life of the transmission.

The link-chain has the further advantage that it can be routed over a circumference that is smaller still than other belts. By running on this minimum wrap diameter, it has the capacity to transfer maximum forces and torques. In this event, only nine pairs of pins are in contact with the inside surfaces of the pulleys. However, the specific contact pressure is so high that they do not slip even under maximum load.

CVT Oil Pump

Since the process of adjusting the pulleys in the CVT requires a high fluid pressure, a high-power oil pump is used to generate this pressure (Figure 14).

13 Link-chain for Audi Multitronic (source: Audi)

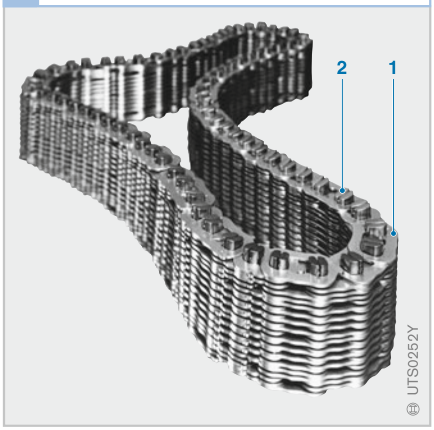
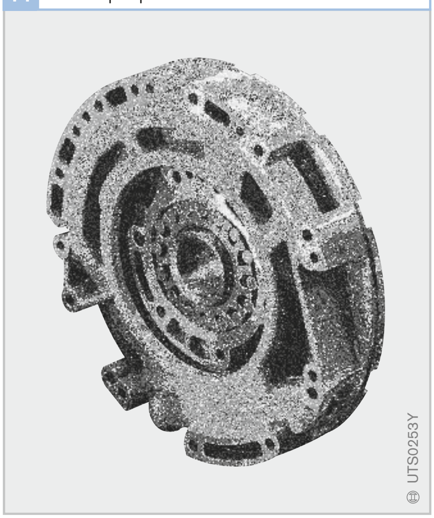


Fig. 13
1 Links
2 Pin

14 CVT oil pump



Toroid Transmission

Application

The toroid transmission is currently only used in Japan in the Cedric and Gloria models built by Nissan.

Design

As a special type of continuously variable transmission (Figures 1 and 2), the toroid transmission is also known as a friction-gear CVT. It is characterized by the following design features:

- converter as power take-up element,
- reverse gear via planetary-gear set,
- power transmission via torus wheels with intermediate rollers,
- continuously variable change of ratio by hydraulic angle adjustment of intermediate rollers,
- high-pressure hydraulics for preloading the torus wheels, and
- electrohydraulic control.

Features

The main features are as follows:

- no interruption of tractive force,
- no gearshifts (high convenience),
- adapted operation in the engine map for optimum fuel consumption/maximum acceleration,
- can be used for high torques,
- rapid ratio adjustment,
- high drive power for the high-pressure pump (overall efficiency therefore only satisfactory), and
- special ATF (automatic transmission fluid) with high shear strength required.

1 Toroid transmission (schematic)

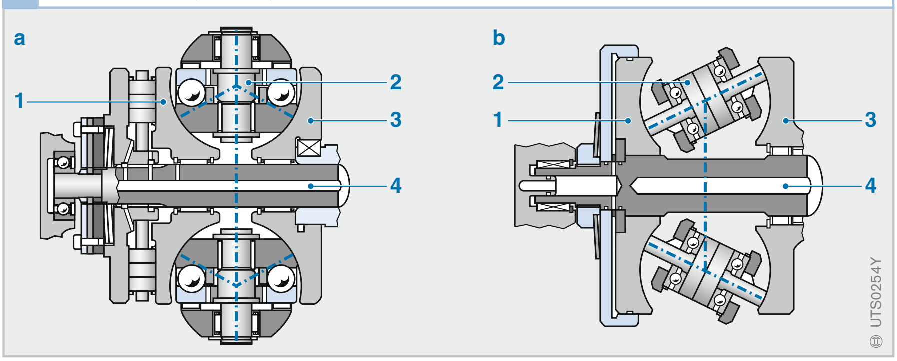


Fig. 1
a Half toroid
b Full toroid

1 Input wheel
2 Variator
3 Output wheel
4 Output

2 Toroid transmission (version)

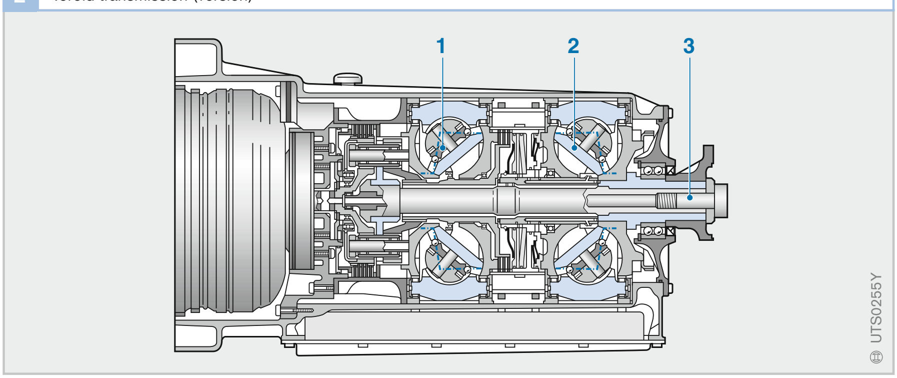


Fig. 2
1 Input wheel
2 Variator
3 Output

Transmission history 2

Daimler/Maybach Steel-Wheel Carriage from 1889 with Four-Speed Transmission

Power transmission in an automobile must guarantee the functions of starting, engine-speed, and torque conversion for forward and reverse travel. These functions call for actuators and shifting elements which intervene in the power-flow and perform engine-speed and torque conversion.

In the very early days of automobile history, many vehicles transferred their engine's motive force to the road with belt and chain drives. Only in the output stage, the final drive, were gear and chain drives soon to be in use due to the high torques involved.

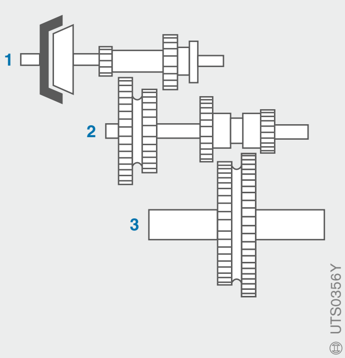
The steel-wheel carriage from Daimler and his designer Maybach from 1889 was the *first four-wheel vehicle* with an internal-combustion engine not to be simply a converted horse-drawn carriage, but to be conceived in its entirety specifically for motorized road traffic. The power-flow of its vertically mounted two-cylinder V-engine with a power output of 2 HP (1.45 kW) was already transferred to the powered axle with a clutch, a four-speed manually shifted gear transmission, and a differential. A gear transmission could specifically carry out the conversion of rotational speed, torque, and direction of rotation in the tightest of spaces.

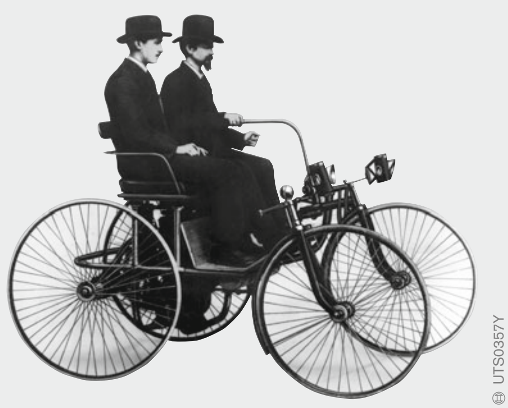
The four-speed transmission to be operated using two shift levers consisted of different straight-toothed gear pairs, one pair of which could always be engaged with the aid of two sliding-gear clusters. The top speed that could be reached ranged between 5 km/h (1st gear) and 16 km/h (4th gear). For the purpose of starting and shifting, the power transmission from the engine to the transmission could be interrupted with a bevel clutch.

Despite the introduction of the variable-ratio gear transmission, the belt drive maintained its position for some time as the power take-up element in the subsequent course of vehicle development because it permitted a certain starting slip and a greater spacing to the other drivetrain components. There were also combinations of belt drives, manual gear transmissions, and chain drives. The chain drive remained in use in passenger cars until roughly 1910. But as engine power output continued to increase, there was no longer any way past the variable-ratio gear transmission with bevel clutch on account of the high forces that were created.

After 1920 the positive connection (with constantly engaged gears) was established by displacing dog clutches with low displacement travel. Then helical gears and synchronization became standard in manually shifted transmissions. Finally, there came the introduction of automatic transmissions, which are usually fitted with planetary-gear sets on account of the high power density.

Daimler/Maybach steel-wheel carriage from 1889 with its four-speed transmission (source: DaimlerChrysler Classic)





1 Transmission input with bevel clutch
2 Sliding-gear cluster 1
3 Sliding-gear cluster 2

Motor-vehicle safety

In addition to the components of the drivetrain (engine, transmission), which provide the vehicle with its means of forward motion, the vehicle systems that limit movement and retard the vehicle also have an important role to play. Without them, safe use of the vehicle in road traffic would not be possible. Furthermore, systems that protect vehicle occupants in the event of an accident are also becoming increasingly important.

Safety systems

There are a many factors that affect vehicle safety in road traffic situations:

- the condition of the vehicle (e.g. level of equipment, condition of tires, component wear),
- the weather, road surface and traffic conditions (e.g. side winds, type of road surface and density of traffic), and
- the capabilities of the driver, i.e. his/her driving skills and physical and mental condition.

In the past, it was essentially only the braking system (apart, of course, from the vehicle lights) consisting of brake pedal, brake lines and wheel brakes that contributed to vehicle safety. Over the course of time though, more and more systems that actively intervene in braking-system operation have been added. Because of their active intervention, these safety systems are also referred to as *active safety systems.*

The motor-vehicle safety systems that are found on the most up-to-date vehicles substantially improve their safety.

The brakes are an essential component of a motor vehicle. They are indispensable for safe use of the vehicle in road traffic. At the slow speeds and with the small amount of traffic that were encountered in the early years of motoring, the demands placed on the braking system were far less exacting than they are today. Over the course of time, braking systems have become more and more highly developed. In the final analysis, the high speeds that cars can be driven at today are only possible because there are reliable braking systems which are capable of slowing down the vehicle and bringing it safely to a halt even in hazardous situations. Consequently, the braking system is a key part of a vehicle's safety systems.

As in all other areas of automotive engineering, electronics have also become established in the safety systems. The demands now placed on safety systems can only be met with the aid of electronic equipment.

1 Safety when driving on roads (concepts and influencing variables)

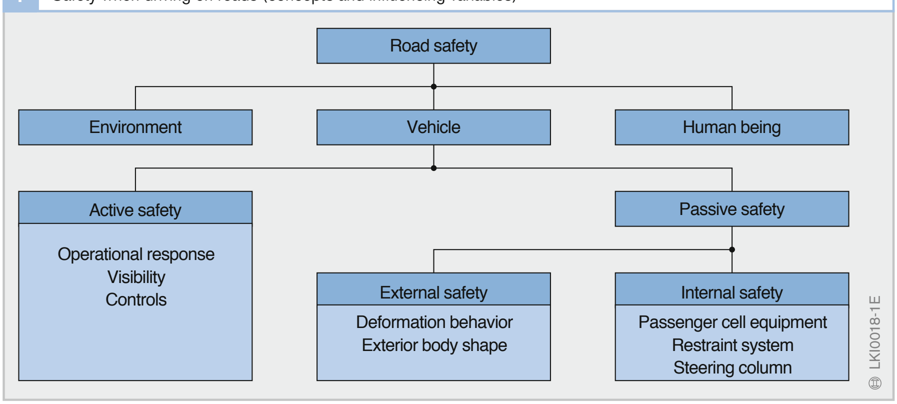


Table 1

K. Reif (Ed.), *Fundamentals of Automotive and Engine Technology*,
DOI 10.1007/978-3-658-03972-1_9, © Springer Fachmedien Wiesbaden 2014

1 Motor-vehicle safety systems

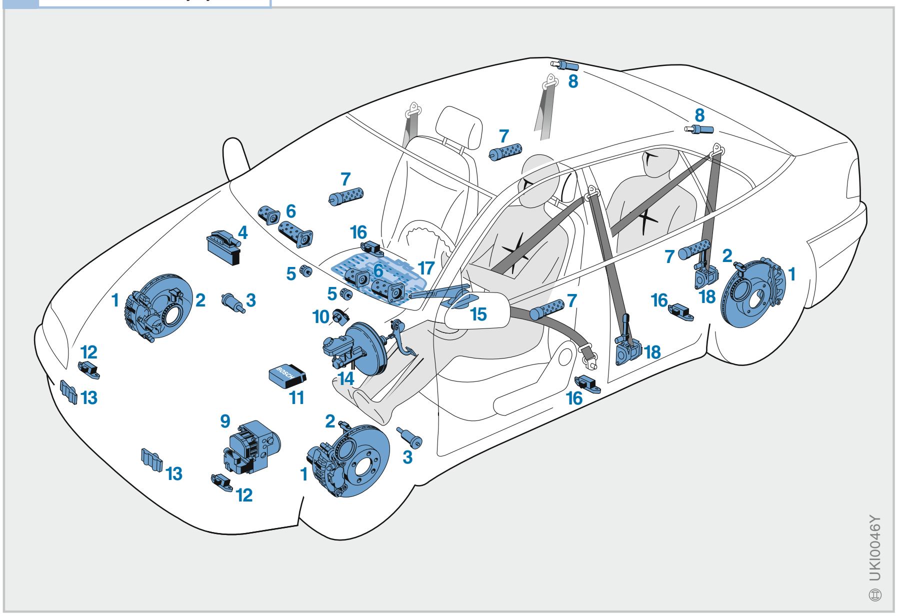


Fig. 1
1 Wheel brake with brake disk
2 Wheel-speed sensor
3 Gas inflator for foot airbag
4 ESP control unit (with ABS and TCS function)
5 Gas inflator for knee airbag
6 Gas inflators for driver and passenger airbags (2-stage)
7 Gas inflator for side airbag
8 Gas inflator for head airbag
9 ESP hydraulic modulator
10 Steering-angle sensor
11 Airbag control unit
12 Upfront sensor
13 Precrash sensor
14 Brake booster with master cylinder and brake pedal
15 Parking brake lever
16 Acceleration sensor
17 Sensor mat for seat-occupant detection
18 Seat belt with seat-belt tightener

Active safety systems

These systems help to prevent accidents and thus make a preventative contribution to road safety. Examples of active vehicle safety systems include

- ABS (**A**ntilock **B**raking **S**ystem),
- TCS (**T**raction **C**ontrol **S**ystem), and
- ESP (**E**lectronic **S**tability **P**rogram).

These safety systems stabilize the vehicle's handling response in critical situations and thus maintain its steerability.

Apart from their contribution to vehicle safety, systems such as Adaptive Cruise Control (ACC) essentially offer added convenience by maintaining the distance from the vehicle in front by automatically throttling back the engine or applying the brakes.

Passive safety systems

These systems are designed to protect vehicle occupants from serious injury in the event of an accident. They reduce the risk of injury and thus the severity of the consequences of an accident.

Examples of passive safety systems are the seat-belts required by law, and airbags – which can now be fitted in various positions inside the vehicle such as in front of or at the side of the occupants.

Fig. 1 illustrates the safety systems and components that are found on modern-day vehicles equipped with the most advanced technology.

Basics of vehicle operation

Driver behavior

The first step in adapting vehicle response to reflect the driver and his/her capabilities is to analyze driver behavior as a whole. Driver behavior is broken down into two basic categories:

- vehicle guidance, and
- response to vehicle instability.

The essential feature of the "vehicle guidance" aspect is the driver's aptitude in anticipating subsequent developments; this translates into the ability to analyze current driving conditions and the associated interrelationships in order to accurately gauge such factors as:

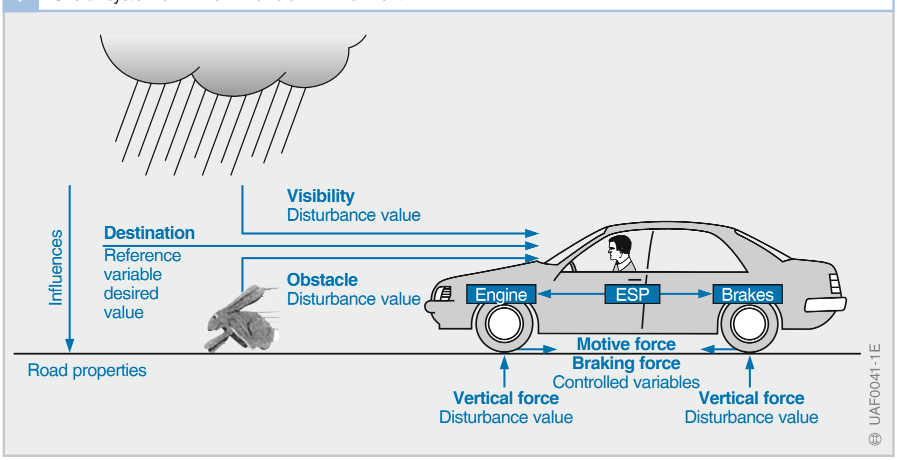
- the amount of initial steering input required to maintain consistently optimal cornering lines when cornering,
- the points at which braking must be initiated in order to stop within available distances, and
- when acceleration should be started in order to overtake slower vehicles without risk.

Steering angle, braking and throttle application are vital elements within the guidance process. The precision with which these functions are discharged depends upon the driver's level of experience.

While stabilizing the vehicle (response to vehicle instability), the driver determines that the actual path being taken deviates from the intended course (the road's path) and that the originally estimated control inputs (steering angle, accelerator pedal pressure) must be revised to avoid traction loss or prevent the vehicle leaving the road. The amount of stabilization (correction) response necessary after initiation of any given maneuver is inversely proportional to the driver's ability to estimate initial guidance inputs; more driver ability leads to greater vehicle stability. Progressively higher levels of correspondence between the initial control input (steering angle) and the actual cornering line produce progressively lower correction requirements; the vehicle reacts to these minimal corrections with "linear" response (driver input is transferred to the road surface proportionally, with no substantial deviations).

Experienced drivers can accurately anticipate both how the vehicle will react to their control inputs and how this reactive motion will combine with predictable external factors and forces (when approaching curves and road works etc.). Novices not only need more time to complete this adaptive process, their results will also harbor a greater potential for error. The conclusion is that inexperi-

1 Overall system of "Driver – Vehicle – Environment"



enced drivers concentrate most of their attention on the stabilization aspect of driving.

When an unforeseen development arises for driver and vehicle (such as an unexpectedly sharp curve in combination with restricted vision, etc.), the former may react incorrectly, and the latter can respond by going into a skid. Under these circumstances, the vehicle responds non-linearly and transgresses beyond its physical stability limits, so that the driver can no longer anticipate the line it will ultimately take. In such cases, it is impossible for either the novice or the experienced driver to retain control over his/her vehicle.

Accident causes and prevention

Human error is behind the vast majority of all road accidents resulting in injury. Accident statistics reveal that driving at an inappropriate speed is the primary cause for most accidents. Other accident sources are

- incorrect use of the road,
- failure to maintain the safety margin to the preceding vehicle,
- errors concerning right-of-way and traffic priority,
- errors occurring when making turns, and
- driving under the influence of alcohol.

Technical deficiencies (lighting, tires, brakes, etc.) and defects related to the vehicle in general are cited with relative rarity as accident sources. Accident causes beyond the control of the driver more frequently stem from other factors (such as weather).

These facts demonstrate the urgency of continuing efforts to enhance and extend the scope of automotive safety technology (with special emphasis on the associated electronic systems). Improvements are needed to

- provide the driver with optimal support in critical situations,
- prevent accidents in the first place, and
- reduce the severity of accidents when they do occur.

The designer's response to critical driving conditions must thus be to foster "predictable" vehicle behavior during operation at physical limits and in extreme situations. A range of parameters (wheel speed, lateral acceleration, yaw velocity, etc.) can be monitored for processing in one or several electronic control units (ECUs). This capability forms the basis of a concept for virtually immediate implementation of suitable response strategies to enhance driver control of critical processes.

The following situations and hazards provide examples of potential "limit conditions":

- changes in prevailing road and/or weather conditions,
- "conflicts of interest" with other road users,
- animals and/or obstructions on the road, and
- a sudden defect (tire blow-out, etc.) on the vehicle.

Critical traffic situations

The one salient factor that distinguishes critical traffic situations is abrupt change, such as the sudden appearance of an unexpected obstacle or a rapid change in road-surface conditions. The problem is frequently compounded by operator error. Owing to lack of experience, a driver who is travelling too fast or is not concentrating on the road will not be able to react with the judicious and rational response that the circumstances demand.

Because drivers only rarely experience this kind of critical situation, they usually fail to recognize how close evasive action or a braking maneuver has brought them to the vehicle's physical limits. They do not grasp how much of the potential adhesion between tires and road surface has already been "used up" and fail to perceive that the vehicle may be at its maneuverability limit or about to skid off the road. The driver is not prepared for this and reacts either incorrectly or too precipitously. The ultimate results are accidents and scenaria that pose threats to other road users.

These factors are joined by still other potential accident sources including outdated technology and deficiencies in infrastructure (badly designed roads, outdated traffic-guidance concepts).

Terms such as "improvements in vehicle response" and "support for the driver in critical situations" are only meaningful if they refer to mechanisms that produce genuine long-term reductions in both the number and severity of accidents. Lowering or removing the risk from these critical situations entails executing difficult driving maneuvers including
- rapid steering inputs including countersteering,
- lane changes during emergency braking,
- maintaining precise tracking while negotiating curves at high speeds and in the face of changes in the road surface.

These kinds of maneuvers almost always provoke a critical response from the vehicle, i.e., lack of tire traction prevents the vehicle reacting in the way that the driver would normally expect; it deviates from the desired course.

Due to lack of experience in these borderline situations, the driver is frequently unable to regain active control of the vehicle, and often panics or overreacts. Evasive action serves as an example. After applying excessive steering input in the moment of initial panic, this driver then countersteers with even greater zeal in an attempt to compensate for his initial error. Extended sequences of steering and countersteering with progressively greater input angles then lead to a loss of control over the vehicle, which responds by breaking into a skid.

Driving behavior

A vehicle's on-the-road handling and braking response are defined by a variety of influences. These can be roughly divided into three general categories:
- vehicle characteristics,
- the driver's behavior patterns, ability and reflexes, and
- peripheral circumstances/or influences from the surroundings or from outside.

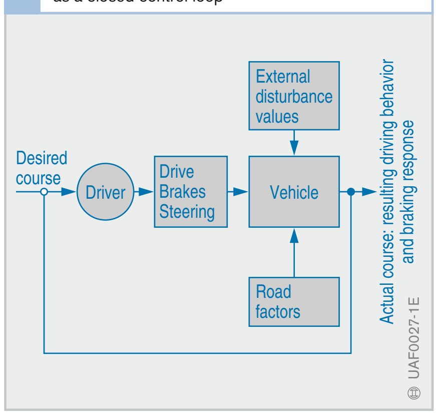
A vehicle's handling, braking and overall dynamic response are influenced by its structure and design.

Handling and braking responses define the vehicle's reactions to driver inputs (at steering wheel, accelerator pedal, brakes, etc.) as do external interference factors (road-surface condition, wind, etc.).

Good handling is characterized by the ability to precisely follow a given course and thus comply in full with driver demand.
The driver's responsibilities include:
- adapting driving style to reflect traffic and road conditions,
- compliance with applicable traffic laws and regulations,
- following the optimal course as defined by the road's geometry as closely as possible, and
- guiding the vehicle with foresight and circumspection.

The driver pursues these objectives by continuously adapting the vehicle's position and motion to converge with a subjective conception of an ideal status. The driver relies upon personal experience to anticipate developments and adapt to instantaneous traffic conditions.

2 Overall system of "driver – vehicle – environment" as a closed control loop



Evaluating driver behavior

Subjective assessments made by experienced drivers remain the prime element in evaluations of vehicle response. Because assessments based on subjective perceptions are only relative and not absolute, they cannot serve as the basis for defining objective "truths". As a result, subjective experience with one vehicle can be applied to other vehicles only on a comparative, relative basis.

Test drivers assess vehicle response using selected maneuvers conceived to reflect "normal" traffic situations. The overall system (including the driver) is judged as a closed loop. While the element "driver" cannot be precisely defined, this process provides a replacement by inputting objective, specifically defined interference factors into the system. The resulting vehicular reaction is then analyzed and evaluated. The following maneuvers are either defined in existing ISO standards or currently going through the standardization process. These dry-surface exercises serve as recognized procedures for assessing vehicular stability:

- steady-state skid-pad circulation,
- transition response,
- braking while cornering,
- sensitivity to crosswinds,
- Straight-running properties (tracking stability), and
- load change on the skid pad.

In this process, prime factors such as road geometry and assignments taken over by the driver assume vital significance. Each test driver attempts to gather impressions and experience in the course of various prescribed vehicle maneuvers; the subsequent analysis process may well include comparisons of the impressions registered by different drivers. These often hazardous driving maneuvers (e.g. the standard VDA evasive-action test, also known as the "elk test") are executed by a series of drivers to generate data describing the dynamic response and general handling characteristics of the test vehicle. The criteria include:

- stability,
- steering response and brake performance, and
- handling at the limit. The tests are intended to describe these factors as a basis for implementing subsequent improvements.

The advantages of this procedure are:

- it allows assessment of the overall, synergistic system ("driver – vehicle – environment") and
- supports realistic simulation of numerous situations encountered under everyday traffic conditions.

The disadvantages of this procedure are:

- the results extend through a broad scatter range, as drivers, wind, road conditions and initial status vary from one maneuver to the next,
- subjective impressions and experience are colored by the latitude for individual interpretation, and
- the success or failure of an entire test series can ultimately be contingent upon the abilities of a single driver.

Table 1 (next page) lists the essential vehicle maneuvers for evaluating vehicle response within a closed control loop.

Owing to the subjective nature of human behavior, there are still no definitions of dynamic response in a closed control loop that are both comprehensive and objectively grounded (closed-loop operation, meaning with driver, Fig. 2).

Despite this, the objective driving tests are complimented by various test procedures capable of informing experienced drivers about a vehicle's handling stability (example: slalom course).

1 Evaluating driver behavior

Vehicle response	Driving maneuver (Driver demand and current conditions)	Driver makes continuous corrections	Steering wheel firmly positioned	Steering wheel released	Steering angle input
Linear response	Straight-running stability – stay in lane	●	●	●	
	Steering response/turning	●			
	Sudden steering – releasing the steering			●	
	Load-change reaction	●	●	●	
	Aquaplaning	●	●	●	
	Straight-line braking	●	●	●	
	Crosswind sensitivity	●	●	●	
	High-speed aerodynamic lift		●		
	Tire defect	●	●	●	
Transition input/ transmission response	Sudden steering-angle change				●
	Single steering and countersteering inputs				●
	Multiple steering and countersteering inputs				●
	Single steering impulse				●
	“Random” steering-angle input	●			●
	Driving into a corner	●			
	Driving out of a corner	●			
	Self-centering			●	
	Single lane change	●			
	Double lane change	●			
Cornering	Steady-state skid-pad circulation		●		
	Dynamic cornering	●	●		
	Load-change reaction when cornering	●	●		
	Steering release			●	
	Braking during cornering	●	●		
	Aquaplaning in curve	●	●		
Alternating directional response	Slalom course around marker cones	●			
	Handling test (test course with sharp corners)	●			
	Steering input/acceleration			●	
Overall characteristics	Tilt resistance	●			●
	Reaction and evasive action tests	●			

Table 1

Driving maneuvers

Steady-state skid-pad circulation

Steady-state cornering around the skid pad is employed to determine maximum lateral acceleration. This procedure also provides information on the transitions that dynamic handling undergoes as cornering forces climb to their maximum. This information can be used to define the vehicle's intrinsic handling (self-steering) properties (oversteer, understeer, neutral cornering response).

3 Evasive maneuver ("Elk Test")

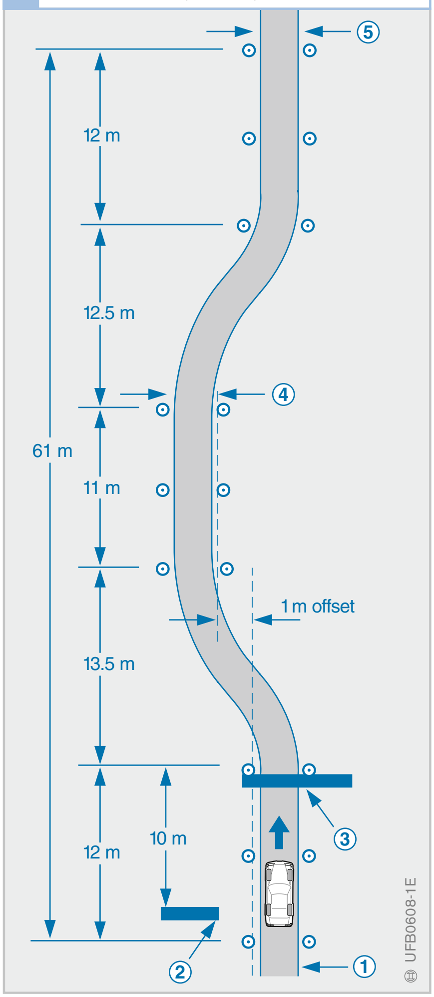


Fig. 3
Test start:
Phase 1:
Top gear (manual transmission)
Position D at 2,000 rpm (automatic transmission)

Phase 2:
Accelerator released

Phase 3:
Speed measurement with photoelectric light barrier

Phase 4:
Steering to the right

Phase 5:
End of test

Transition response

Transition response joins steady-state self-steering properties (during skid-pad circulation) as a primary assessment parameter. This category embraces such maneuvers as suddenly taking rapid evasive action when driving straight ahead.

The "elk test" simulates an extreme scenario featuring sudden evasive action to avoid an obstacle. A vehicle traveling over a 50 meter stretch of road must safely drive around an obstacle 10 meters in length projecting outward onto the track by a distance of 4 meters (Fig. 3).

Braking during cornering – load-change reactions

One of the most critical situations encountered in every-day driving – and thus one of the most vital considerations for vehicle design – is braking during cornering.

From the standpoint of the physical forces involved, whether the driver simply releases the accelerator or actually depresses the brake pedal is irrelevant; the physical effects will not differ dramatically. The resulting load shift from rear to front increasing the rear slip angle while reducing that at the front, and since neither the given cornering radius nor the vehicle speed modifies the lateral force requirement, the vehicle tends to adopt an oversteering attitude.

With rear-wheel drive, tire slip exerts less influence on the vehicle's intrinsic handling response than with front-wheel drive; this means that RWD vehicles are more stable under these conditions.

Vehicle reaction during this maneuver must represent the optimal compromise between steering response, stability and braking efficiency.

Parameters

The primary parameters applied in the assessment of dynamic handling response are:

- steering-wheel angle,
- lateral acceleration,
- longitudinal acceleration or longitudinal deceleration,
- yaw velocity,
- side-slip angle and roll angle.

Additional data allow more precise definition of specific handling patterns as a basis for evaluating other test results:

- longitudinal and lateral velocity,
- steering angles of front/rear wheels,
- slip angle at all wheels,
- steering-wheel force.

Reaction time

Within the overall system "driver-vehicle-environment", the driver's physical condition and state of mind, and thus his/her reaction times, join the parameters described above as decisive factors. This lag period is the time that elapses between perception of an obstacle and initial application of pressure to the brake pedal. The decision to act and the foot movement count as intermediate stages in this process. This period is not consistent; depending upon personal factors and external circumstances it is at least 0.3 seconds.

Special examinations are required to quantify individual reaction patterns (as conducted by medical/psychological institutes).

Motion

Vehicle motion may be consistent in nature (constant speed) or it may be inconsistent (during acceleration from a standing or rolling start, or deceleration and braking with the accompanying change in velocity).

The engine generates the kinetic energy required to propel the vehicle. Forces stemming either from external sources or acting through the engine and drivetrain must always be applied to the vehicle as a basic condition for changes in the magnitude and direction of its motion.

Handling and braking response in commercial vehicles

Objective evaluation of handling and braking response in heavy commercial vehicles is based on various driving maneuvers including steady-state skid-pad cornering, abrupt steering-angle change (vehicle reaction to "tugging" the steering wheel through a specified angle) and braking during cornering.

The dynamic lateral response of tractor and trailer combinations generally differs substantially from that of single vehicles. Particular emphasis is placed on tractor and trailer loading, while other important factors include design configuration and the geometry of the linkage elements within the combination.

The worst-case scenario features an empty truck pulling a loaded central-axle trailer. Operating a combination in this state

4 Actions: reaction, brake and stop

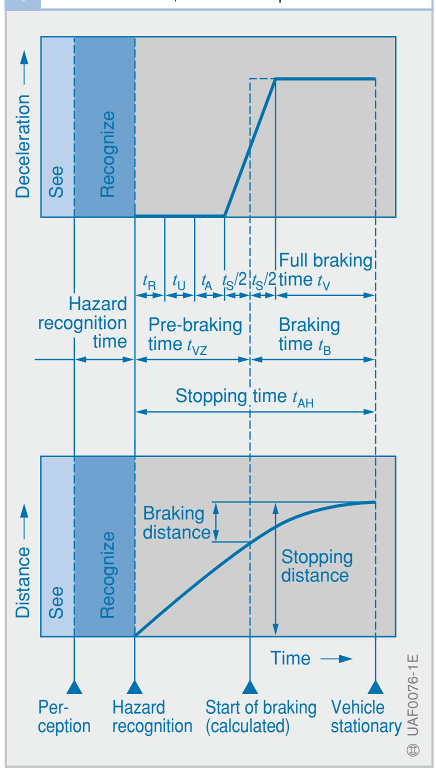


Fig. 4

- t_R Reaction time
- t_U Conversion time
- t_A Response time
- t_S Pressure buildup

requires a high degree of skill and circumspection on the part of the driver.

Jack-knifing is also a danger when tractor-trailer combinations are braked in extreme situations. This process is characterized by a loss of lateral traction at the tractor's rear axle and is triggered when "overbraking" on slippery road surfaces, or by extreme yaw rates on μ-split surfaces (with different friction coefficients at the center and on the shoulder of the lane). Jack-knifing can be avoided with the aid of antilock braking systems (ABS).

2 Personal reaction-time factors

→	Psychophysical reaction ←			→ Muscular reaction		→
Perceived object	Perception	Comprehension	Decision	Mobilization	Motion	Object of action
(e.g. traffic sign)	Visual acuity	Perception and registration	Processing	Movement apparatus	Personal implementation speed	(e.g. brake pedal)

Table 2

3 Reaction time as a function of personal and external factors

Short reaction time ←	**→ Long reaction time**
Personal factors, driver	
Trained reflex action	Ratiocinative reaction
Good condition, optimal performance potential	Poor condition, e.g. fatigue
High level of driving skill	Low level of driving skill
Youth	Advanced age
Anticipatory attitude	Inattention, distraction
Good physical and mental health	Physical or mental impediment
	Panic, alcohol
External Factors	
Simple, unambiguous, predicable and familiar traffic configuration	Complex, unclear, incalculable and unfamiliar traffic conditions
Conspicuous obstacle	Inconspicuous obstacle
Obstacle in line of sight	Obstacle on visual periphery
Logical and effective arrangement of the controls in the vehicle	Illogical and ineffective control arrangement in vehicle

Table 3

Basic principles of vehicle dynamics

A body can only be made to move or change course by the action of forces. Many forces act upon a vehicle when it is being driven. An important role is played by the tires as any change of speed or direction involves forces acting on the tires.

Tires

Task

The tire is the connecting link between the vehicle and the road. It is at that point that the safe handling of a vehicle is ultimately decided. The tire transmits motive, braking and lateral forces within a physical environment whose parameters define the limits of the dynamic loads to which the vehicle is subjected. The decisive criteria for the assessment of tire quality are:

- Straight-running ability
- Stable cornering properties
- Ability to grip on a variety of road surfaces
- Ability to grip in a variety of weather conditions
- Steering characteristics
- Ride comfort (vibration absorption and damping, quietness)
- Durability and
- Economy

Design

There are a number of different tire designs that are distinguished according to the nature and sophistication of the technology employed. The design of a conventional tire is determined by the characteristics required of it in normal conditions and emergency situations.

Legal requirements and regulations specify which tires must be used in which conditions, the maximum speeds at which different types of tire may be used, and the criteria by which tires are classified.

Radial tires

In a radial tire, the type which has now become the standard for cars, the cords of the tire-casing plies run radially, following the shortest route from bead to bead (Fig. 1).
A reinforcing belt runs around the perimeter of the relatively thin, flexible casing.

1 Structure of a radial car tire

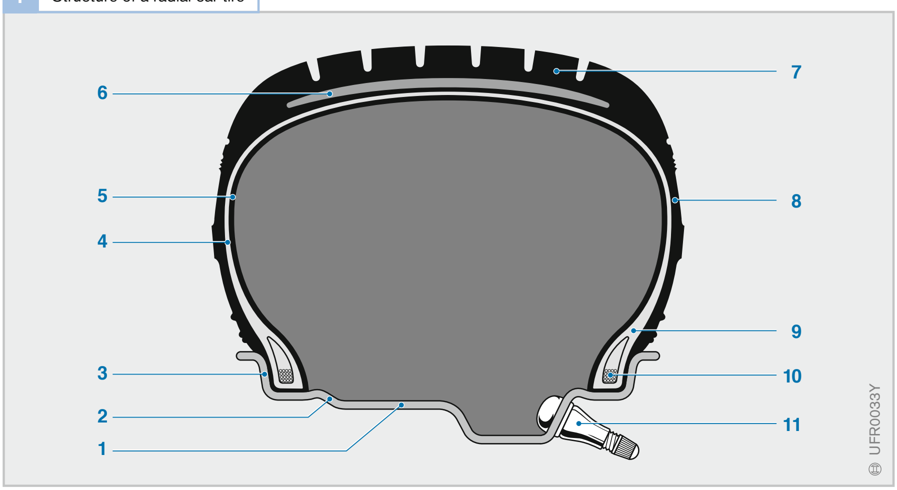


Fig. 1
1 Rim bead seat
2 Hump
3 Rim flange
4 Casing
5 Air-tight rubber layer
6 Belt
7 Tread
8 Sidewall
9 Bead
10 Bead core
11 Valve

K. Reif (Ed.), *Fundamentals of Automotive and Engine Technology*,
DOI 10.1007/978-3-658-03972-1_10, © Springer Fachmedien Wiesbaden 2014

Cross-ply tires

The cross-ply tire takes its name from the fact that the cords of alternate plies of the tire casing run at right angles to one another so that they cross each other. This type of tire is now only of significance for motorcycles, bicycles, and industrial and agricultural vehicles. On commercial vehicles it is increasingly being supplanted by the radial tire.

Regulations

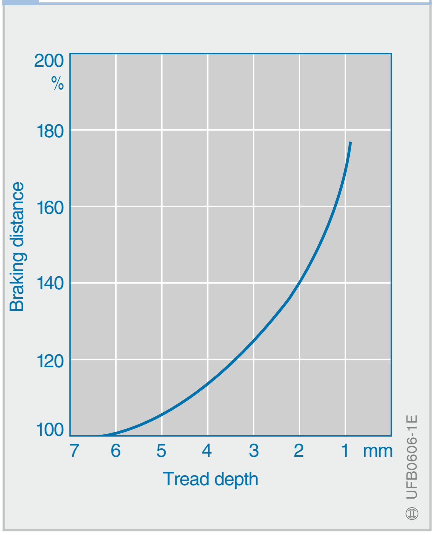
In Europe, the Council Directives, and in the USA the *FMVSS (Federal Motor Vehicle Safety Standard)* require that motor vehicles and trailers are fitted with pneumatic tires with a tread pattern consisting of grooves with a depth of at least 1.6 mm around the entire circumference of the tire and across the full width of the tread.

Cars and motor vehicles with a permissible laden weight of less than 2.8 tonnes and designed for a maximum speed of more than 40 km/h, and trailers towed by them, must be fitted either with cross-ply tires all round or with radial tires all round; in the case of vehicle-and-trailer combinations the requirement applies individually to each unit of the combination. It does not apply to trailers towed by vehicles at speeds of up to 25 km/h.

Application

To ensure correct use of tires, it is important the correct tire is selected according to the recommendations of the vehicle or tire manufacturer. Fitting the same type of tire to all wheels of a vehicle ensures the best handling results. The specific instructions of the tire manufacturer or a tire specialist regarding tire care, maintenance, storage and fitting should be followed in order to obtain maximum durability and safety.

2 Increase in braking distance on wet road surface as a function of tread depth at 100 km/h



When the tires are in use, i.e. when they are fitted to the wheel, care should be taken to ensure that

- the wheels are balanced so as to guarantee optimum evenness of running,
- all wheels are fitted with the same type of tire and the tires are the correct size for the vehicle,
- the vehicle is not driven at speeds in excess of the maximum allowed for the tires fitted, and
- the tires have sufficient depth of tread.

The less tread there is on a tire, the thinner is the layer of material protecting the belt and the casing underneath it. And particularly on cars and fast commercial vehicles, insufficient tread depth on wet road surfaces has a decisive effect on safe handling characteristics due to the reduction in grip. Braking distance increases disproportionately as tread depth reduces (Fig. 2). An especially critical handling scenario is aquaplaning in which all adhesion between tires and road surface is lost and the vehicle is no longer steerable.

Tire slip

Tire slip, or simply "slip", is said to occur when there is a difference between the theoretical and the actual distance traveled by a vehicle.

This can be illustrated by the following example in which we will assume that the circumference of a car tire is 2 meters. If the wheel rotates ten times, the distance traveled should be 20 meters. If tire slip occurs, however, the distance actually traveled by the braked vehicle is greater.

3 Effect of braking on a rolling wheel

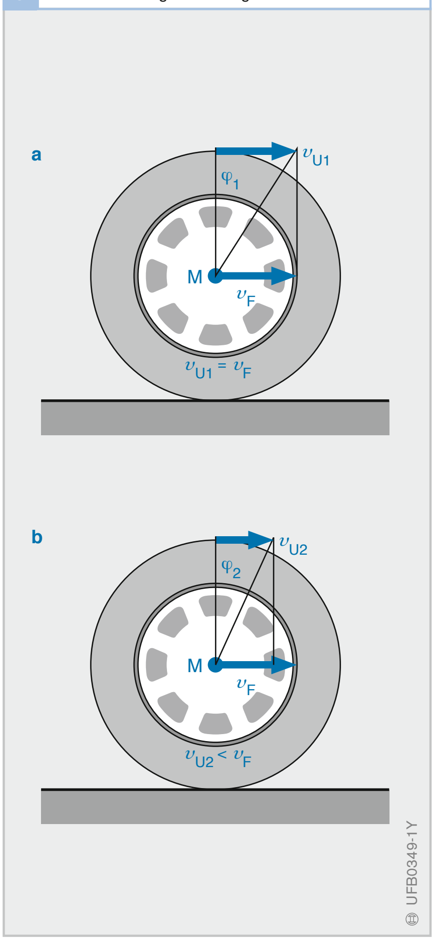


Fig. 3
- a Rolling wheel (unbraked)
- b Braked wheel
- v_F Vehicle speed at wheel center, M
- v_U Circumferential speed

On a braked wheel, the angle of rotation, φ, per unit of time is smaller (slip)

Causes of tire slip

When a wheel rotates under the effect of power transmission or braking, complex physical processes take place in the contact area between tire and road which place the rubber parts under stress and cause them to partially slide, even if the wheel does not fully lock. In other words, the elasticity of the tire causes it to deform and "flex" to a greater or lesser extent depending on the weather conditions and the nature of the road surface. As the tire is made largely of rubber, only a proportion of the "deformation energy" is recovered as the tread moves out of the contact area. The tire heats up in the process and energy loss occurs.

Illustration of slip

The slip component of wheel rotation is referred to by λ, where

$$\lambda = (v_F - v_U)/v_F$$

The quantity v_F is the vehicle road speed, v_U is the circumferential velocity of the wheel (Fig. 3). The formula states that brake slip occurs as soon as the wheel is rotating more slowly than the vehicle road speed would normally demand. Only under that condition can braking forces or acceleration forces be transmitted.

Since the tire slip is generated as a result of the vehicle's longitudinal movement, it is also referred to as "longitudinal slip". The slip generated during braking is usually termed "brake slip".

If a tire is subjected to other factors in addition to slip (e.g. greater weight acting on the wheels, extreme wheel positions), its force transmission and handling characteristics will be adversely affected.

Forces acting on a vehicle

Theory of inertia
Inertia is the property possessed by all bodies, by virtue of which they will naturally maintain the status in which they find themselves, i.e. either at rest or in motion. In order to bring about a change to that status, a force has to be applied to the body. For example, if a car's brakes are applied when it is cornering on black ice, the car will carry on in a straight line without altering course and without noticeably slowing down. That is because on black ice, only very small tire forces can be applied to the wheels.

Turning forces
Rotating bodies are influenced by turning forces. The rotation of the wheels, for example, is slowed down due to the braking torque and accelerated due to the drive torque.

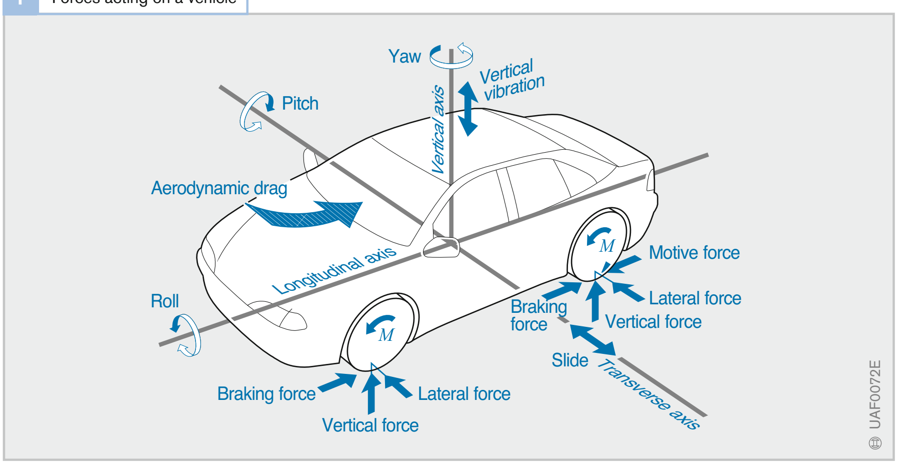
Turning forces act on the entire vehicle. If the wheels on one side of the vehicle are on a slippery surface (e.g. black ice) while the wheels on the other side are on a road surface with normal grip (e.g. asphalt), the vehicle will slew around its vertical axis when the brakes are applied (μ-split braking). This rotation is caused by the yaw moment, which arises due to the different forces applied to the sides of the vehicle.

Distribution of forces
In addition to the vehicle's weight (resulting from gravitational force), various different types of force act upon it regardless of its state of motion (Fig. 1). Some of these are
- forces which act along the longitudinal axis of the vehicle (e.g. motive force, aerodynamic drag or rolling friction); others are
- forces which act laterally on the vehicle (e.g. steering force, centrifugal force when cornering or crosswinds). The tire forces which act laterally on the vehicle are also referred to as lateral forces.

The longitudinal and the lateral forces are transmitted either "downwards" or "sideways" to the tires and ultimately to the road. The forces are transferred through
- the chassis (e.g. wind),
- the steering (steering force),
- the engine and transmission (motive force), or
- the braking system (braking force).

Opposing forces act "upwards" from the road onto the tires and thence to the vehicle because every force produces an opposing force.

1 Forces acting on a vehicle



Basically, in order for the vehicle to move, the motive force of the engine (engine torque) must overcome all forces that resist motion (all longitudinal and lateral forces) such as are generated by road gradient or camber.

In order to assess the dynamic handling characteristics or handling stability of a vehicle, the forces acting between the tires and the road, i.e. the forces transmitted in the contact areas between tire and road surface (also referred to as "tire contact area" or "footprint"), must be known.

With more practice and experience, a driver generally learns to react more effectively to those forces. They are evident to the driver when accelerating or slowing down as well as in cross winds or on slippery road surfaces. If the forces are particularly strong, i.e. if they produce exaggerated changes in the motion of the vehicle, they can also be dangerous (skidding) or at least are detectable by squealing tires (e.g. when accelerating aggressively) and increased component wear.

2 Components of tire force and pressure distribution over the footprint of a radial tire

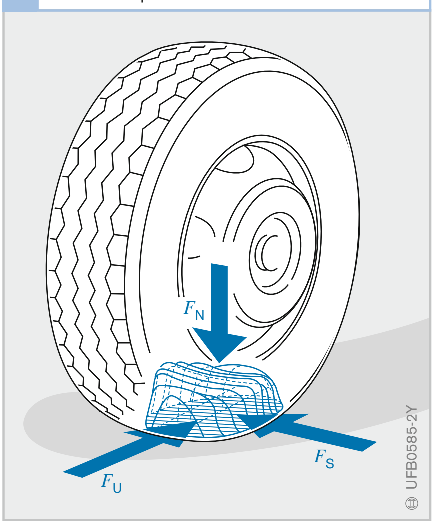


Fig. 2

F_N Vertical tire force, or normal force

F_U Circumferential force (positive: motive force; negative: braking force)

F_S Lateral force

Tire forces

A motor vehicle can only be made to move or change its direction in a specific way by forces acting through the tires. Those forces are made up of the following components (Fig. 2):

Circumferential force

The circumferential force F_U is produced by power transmission or braking. It acts on the road surface as a linear force in line with the longitudinal axis of the vehicle and enables the driver to increase the speed of the vehicle using the accelerator or slow it down with the brakes.

Vertical tire force (normal force)

The vertical force acting downwards between the tire and road surface is called the vertical tire force or normal force F_N. It acts on the tires at all times regardless of the state of motion of the vehicle, including, therefore, when the vehicle is stationary.

The vertical force is determined by the proportion of the combined weight of vehicle and payload that is acting on the individual wheel concerned. It also depends on the degree of upward or downward gradient of the road that the vehicle is standing on. The highest levels of vertical force occur on a level road.

Other forces acting on the vehicle (e.g. heavier payload) can increase or decrease the vertical force. When cornering, the force is reduced on the inner wheels and increased on the outer wheels.

The vertical tire force deforms the part of the tire in contact with the road. As the tire sidewalls are affected by that deformation, the vertical force cannot be evenly distributed. A trapezoidal pressure-distribution pattern is produced (Fig. 2). The tire sidewalls absorb the forces and the tire deforms according to the load applied to it.

Lateral force

Lateral forces act upon the wheels when steering or when there is a crosswind, for example. They cause the vehicle to change direction.

Braking torque

When the brakes are applied, the brake shoes press against the brake drums (in the case of drum brakes) or the brake pads press against the disks (in the case of disk brakes). This generates frictional forces, the level of which can be controlled by the driver by the pressure applied to the brake pedal.

The product of the frictional forces and the distance at which they act from the axis of rotation of the wheel is the braking torque M_B.

That torque is effective at the circumference of the tire under braking (Fig. 1).

Yaw moment

The yaw moment around the vehicle's vertical axis is caused by different longitudinal forces acting on the left and right-hand sides of the vehicle or different lateral forces acting at the front and rear axles. Yaw moments are required to turn the vehicle when cornering. Undesired yaw moments, such as can occur when braking on μ-split (see above) or if the vehicle pulls to one side when braking, can be reduced using suitable design measures. The kingpin offset is the distance between the point of contact between the tire and the road and the point at which the wheel's steering axis intersects the road surface (Fig. 3). It is negative if the point at which the steering axis intersects the road surface is on the outside of the point of contact between tire and road. Braking forces combine with positive and negative kingpin offset to create a lever effect that produces a turning force at the steering which can lead to a certain steering angle at the wheel. If the kingpin offset is negative, this steering angle counters the undesired yaw moment.

3 Kingpin offset

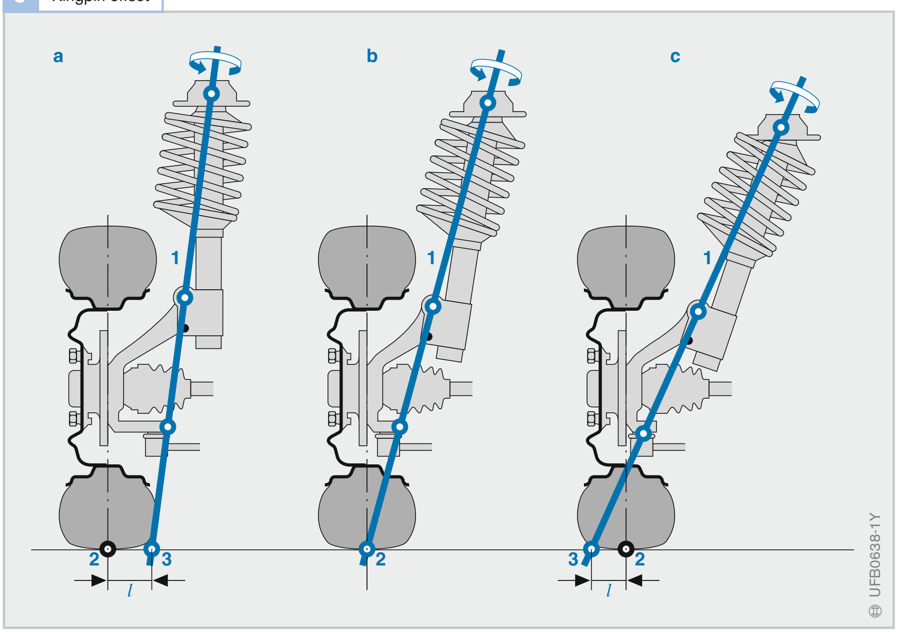


Fig. 3
- a Positive kingpin offset: $M_{Ges} = M_T + M_B$
- b Zero kingpin offset: no yaw moment
- c Negative kingpin offset: $M_{Ges} = M_T - M_B$

- 1 Steering axis
- 2 Wheel contact point
- 3 Intersection point
- l Kingpin offset
- M_{Ges} Total turning force (yaw moment)
- M_T Moment of inertia
- M_B Braking torque

Friction force

Coefficient of friction

When braking torque is applied to a wheel, a braking force F_B is generated between the tire and the road surface that is proportional to the braking torque under stationary conditions (no wheel acceleration). The braking force transmitted to the road (frictional force F_R) is proportional to the vertical tire force F_N:

$$F_R = \mu_{HF} \cdot F_N$$

The factor μ_{HF} is the coefficient of friction. It defines the frictional properties of the various possible material pairings between tire and road surface and the environmental conditions to which they are exposed.

The coefficient of friction is thus a measure of the braking force that can be transmitted. It is dependent on
- the nature of the road surface,
- the condition of the tires,
- the vehicle's road speed, and
- the weather conditions.

The coefficient of friction ultimately determines the degree to which the braking torque is actually effective. For motor-vehicle tires, the coefficient of friction is at its highest on a clean and dry road surface; it is at its lowest on ice. Fluids (e.g. water) or dirt between the tire and the road surface reduce the coefficient of friction. The figures quoted in Table 1 apply to concrete and tarmacadam road surfaces in good condition.

On wet road surfaces in particular, the coefficient of friction is heavily dependent on vehicle road speed. At high speeds on less than ideal road surfaces, the wheels may lock up under braking because the coefficient of friction is not high enough to provide sufficient adhesion for the tires to grip the road surface. Once a wheel locks up, it can no longer transmit side forces and the vehicle is thus no longer steerable. Fig. 5 illustrates the frequency distribution of the coefficient of friction at a locked wheel at various road speeds on wet roads.

The friction or adhesion between the tire and the road surface determines the wheel's ability to transmit force. The ABS (**A**ntilock **B**raking **S**ystem) and TCS (**T**raction **C**ontrol **S**ystem) safety systems utilize the available adhesion to its maximum potential.

4 Linear wheel velocity, v_x, with braking force, F_B, and braking torque, M_B

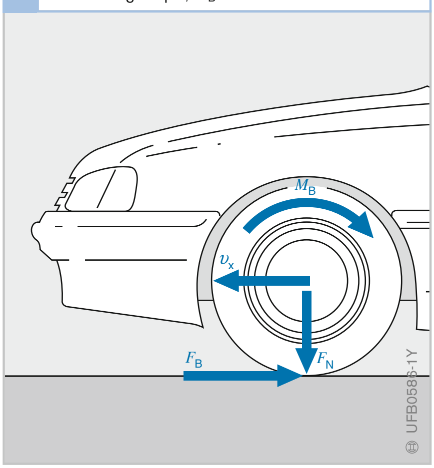


Fig. 4
v_x Linear velocity of wheel
F_N Vertical tire force (normal force)
F_B Braking force
M_B Braking torque

5 Frequency distribution of the coefficient of friction at a locked wheel at various road speeds on wet roads

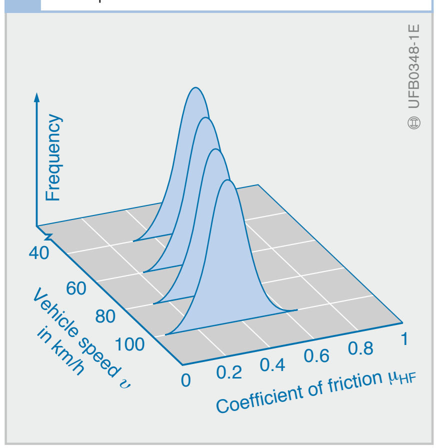


Fig. 5
Source: Forschungsinstitut für Kraftfahrwesen und Fahrzeugmotoren, Stuttgart, Germany (research institute for automotive engineering and automotive engines)

Aquaplaning

The amount of friction approaches zero if rainwater forms a film on the road surface on which the vehicle then "floats". Contact between the tires and the road surface is then lost and the effect known as aquaplaning occurs. Aquaplaning is caused by a "wedge" of water being forced under the entire contact area of the tire with the road surface, thereby lifting it off the ground. Aquaplaning is dependent on:

- the depth of water on the road,
- the speed of the vehicle,
- the tire tread pattern, tire width and level of wear, and
- the force pressing the tire against the road surface.

Wide tires are particularly susceptible to aquaplaning. When a vehicle is aquaplaning, it cannot be steered or braked. Neither steering movements nor braking forces can be transmitted to the road.

Kinetic friction

When describing processes involving friction, a distinction is made between static friction and kinetic friction. With solid bodies, the static friction is greater than kinetic friction. Accordingly, for a rolling rubber tire there are circumstances in which the coefficient of friction is greater than when the wheel locks. Nevertheless, the tire can also slide while it is rolling, and on motor vehicles this is referred to as slip.

Effect of brake slip on coefficient of friction

When a vehicle is pulling away or accelerating – just as when braking or decelerating – the transmission of forces from tire to road depends on the degree of adhesion between the two. The friction of a tire basically has a constant relationship to the level of adhesion under braking or acceleration.

Fig. 6 shows the progression of the coefficient of friction μ_{HF} under braking. Starting from a zero degree of brake slip, is rises steeply to its maximum at between 10% and 40% brake slip, depending on the nature of the road surface and the tires, and then drops away again. The rising slope of the

6 Coefficient of friction, μ_{HF}, and lateral-force coefficient, μ_S, relative to brake slip

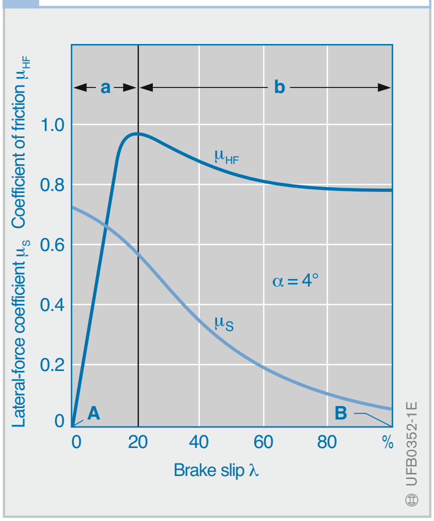


Fig. 6
a Stable zone
b Unstable zone
α Slip angle
A Rolling wheel
B Locked wheel

1 Coefficients of friction, μ_{HF}, for tires in various conditions of wear, on various road conditions and at various speeds

Vehicle road speed km/h	Tire condition	Dry road μ_{HF}	Wet road (depth of water 0.2 mm) μ_{HF}	Heavy rain (depth of water 1 mm) μ_{HF}	Puddles (depth of water 2 mm) μ_{HF}	Icy (black ice) μ_{HF}
50	new	0.85	0.65	0.55	0.5	0.1
	worn out	1	0.5	0.4	0.25	and below
90	new	0.8	0.6	0.3	0.05	
	worn out	0.95	0.2	0.1	0.0	
130	new	0.75	0.55	0.2	0	
	worn out	0.9	0.2	0.1	0	

Table 1

curve represents the "stable zone" (partial-braking zone), while the falling slope is the "unstable zone".

Most braking operations involve minimal levels of slip and take place within the stable zone so that an increase in the degree of slip simultaneously produces an increase in the usable adhesion. In the unstable zone, an increase in the amount of slip generally produces a reduction in the level of adhesion. When braking in such situations, the wheel can lock up within a fraction of a second, and under acceleration the excess power-transmission torque rapidly increases the wheel's speed of rotation causing it to spin.

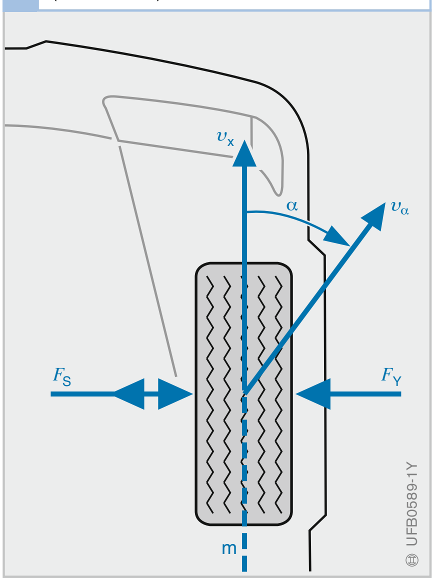
When a vehicle is traveling in a straight line, ABS and TCS prevent it entering the unstable zone when braking or accelerating.

Sideways forces

If a lateral force acts on a rolling wheel, the center of the wheel moves sideways. The ratio between the lateral velocity and the velocity along the longitudinal axis is referred to as "lateral slip". The angle between the resulting velocity, v_α, and the forward velocity, v_x, is called the "lateral slip angle α" (Fig. 7). The side-slip angle, γ, is the angle between the vehicle's direction of travel and its longitudinal axis. The side-slip angle encountered at high rates of lateral acceleration is regarded as an index of controllability, in other words the vehicle's response to driver input.

Under steady-state conditions (when the wheel is not being accelerated), the lateral force F_S acting on the center of the wheel is in equilibrium with the lateral force applied to the wheel by the road surface. The relationship between the lateral force acting through the center of the wheel and the wheel contact force F_N is called the "lateral-force coefficient μ_S".

7 Lateral slip angle, α, and the effect of lateral force, F_S, (overhead view)



8 Position of tire contact area relative to wheel in a right-hand bend showing lateral force, F_S, (front view)

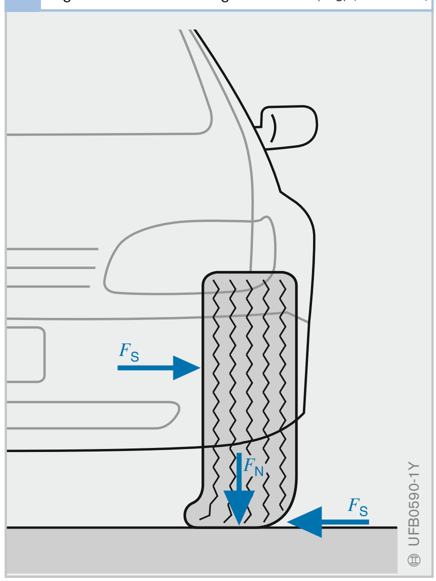


Fig. 7
v_α Velocity in lateral slip direction
v_x Velocity along longitudinal axis
F_S, F_y Lateral force
α Slip angle

Fig. 8
F_N Vertical tire force (normal force)
F_S Lateral force

There is a nonlinear relationship between the slip angle α and the lateral-force coefficient μ_S that can be described by a lateral slip curve. In contrast with the coefficient of friction μ_{HF} that occurs under acceleration and braking, the lateral-force coefficient μ_S is heavily dependent on the wheel contact force F_N. This characteristic is of particular interest to vehicle manufacturers when designing suspension systems so that handling characteristics can be enhanced by stabilizers.

With a strong lateral force, F_S, the tire contact area (footprint) shifts significantly relative to the wheel (Fig. 8). This retards the buildup of the lateral force. This phenomenon greatly affects the transitional response (behavior during transition from one dynamic state to another) of vehicles under steering.

Effect of brake slip on lateral forces

When a vehicle is cornering, the centrifugal force acting outwards at the center of gravity must be held in equilibrium by lateral forces on all the wheels in order for the vehicle to be able to follow the curve of the road.

However, lateral forces can only be generated if the tires deform flexibly sideways so that the direction of movement of the wheel's center of gravity at the velocity, v_α, diverges from the wheel center plane "m" by the lateral slip angle, α (Fig. 7).

Fig. 6 shows the lateral-force coefficient, μ_S, as a function of brake slip at a lateral slip angle of 4°. The lateral-force coefficient is at its highest when the brake slip is zero. As brake slip increases, the lateral-force coefficient declines gradually at first and then increasingly rapidly until it reaches its lowest point when the wheel locks up. That minimum figure occurs as a result of the lateral slip angle position of the locked wheel, which at that point provides no lateral force whatsoever.

Friction – tire slip – vertical tire force

The friction of a tire depends largely on the degree of slip. The vertical tire force plays a subordinate role, there being a roughly linear relationship between braking force and vertical tire force at a constant level of slip.

The friction, however, is also dependent on the tire's lateral slip angle. Thus the braking and motive force reduces as the lateral slide angle is increased at a constant level of tire slip. Conversely, if the braking and motive force remains constant while the lateral slip angle is increased, the degree of tire slip increases.

Dynamics of linear motion

If the rim of a wheel is subjected both to a lateral force and braking torque, the road surface reacts to this by exerting a lateral force and a braking force on the tire. Accordingly, up to a specific limit determined by physical parameters, all forces acting on the rotating wheel are counterbalanced by equal and opposite forces from the road surface.

Beyond that limit, however, the forces are no longer in equilibrium and the vehicle's handling becomes unstable.

Total resistance to motion

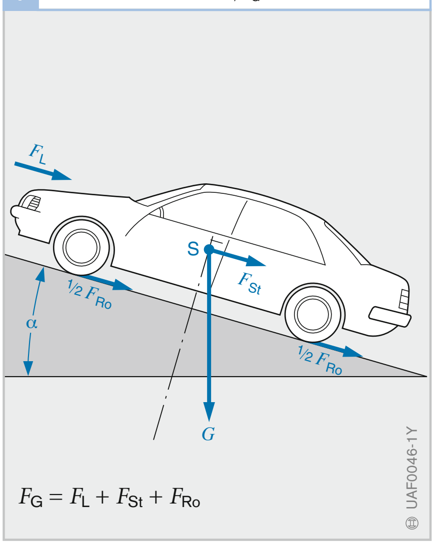
The total resistance to vehicle motion, F_G, is the sum of the rolling resistance, aerodynamic drag and climbing resistance (Fig. 1). In order to overcome that total resistance, a sufficient amount of motive force has to be applied to the driven wheels. The greater the engine torque, the higher the transmission ratio between the engine and the driven wheels and the smaller the power loss through the drivetrain (efficiency η is approx. 0.88...0.92 with engines mounted in line, and approx. 0.91...0.95 with transversely mounted engines), the greater is the motive force available at the driven wheels.

A proportion of the motive force is required to overcome the total resistance to motion. It is adapted to suit the substantial increase in motion resistance on uphill gradients by the use of a choice of lower gearing ratios (multi-speed transmission). If there is a "surplus" of power because the motive force is greater than the resistance to motion, the vehicle will accelerate. If the overall resistance to motion is greater, the vehicle will decelerate.

Rolling resistance when traveling in a straight line

Rolling resistance is produced by deformation processes which occur where the tire is in contact with the road. It is the product of weight and rolling resistance coefficient and increases with a smaller wheel diameter and the greater the degree of deformation of the tire, e.g. if the tire is under-inflated. However, it also increases as the weight on the wheel and the velocity increases. Furthermore, it varies according to type of road surface – on asphalt, for example, it is only around 25% of what it is on a dirt track.

1 Total resistance to motion, F_G



$$F_G = F_L + F_{St} + F_{Ro}$$

Fig. 1
F_L Aerodynamic drag
F_{Ro} Rolling resistance
F_{St} Climbing resistance
F_G Total resistance to motion
G Weight
α Incline angle/gradient angle
S Center of gravity

Table 1

1 Examples of drag coefficient, c_W, for cars

Vehicle body shape	c_W
Convertible with top down	0.5 ... 0.7
Box-type	0.5 ... 0.6
Conventional saloon [1]	0.4 ... 0.55
Wedge shape	0.3 ... 0.4
Aerodynamic fairings	0.2 ... 0.25
Tear-drop	0.15 ... 0.2

[1] "Three-box" design

Table 2

2 Examples of drag coefficient, c_W, for commercial vehicles

Vehicle body shape	c_W
Standard tractor unit	
– without fairings	≥ 0.64
– with some fairings	0.54 ... 0.63
– with all fairings	≤ 0.53

Rolling resistance when cornering

When cornering, the rolling resistance is increased by an extra component, cornering resistance, the coefficient of which is dependent on vehicle speed, the radius of the bend being negotiated, suspension characteristics, type of tires, tire pressure and lateral-slip characteristics.

Aerodynamic drag

The aerodynamic drag F_L is calculated from the air density ϱ, the drag coefficient c_W (dependent on the vehicle body shape, Tables 1 and 2), vehicle's frontal cross-sectional area A and the driving speed v (taking account of the headwind speed).

$$F_L = c_W \cdot A \cdot v^2 \cdot \varrho/2$$

Climbing resistance

Climbing resistance, F_{St} (if positive), or gravitational pull (if negative) is the product of the weight of the vehicle, G, and the angle of uphill or downhill gradient, α.

$$F_{St} = G \cdot \sin \alpha$$

Acceleration and deceleration

Steady acceleration or deceleration in a straight line occurs when the rate of acceleration (or deceleration) is constant. The distance required for deceleration is of greater significance than that required for acceleration because braking distance has direct implications in terms of vehicle and road safety.

The braking distance is dependent on a number of factors including

- Vehicle speed: at a constant rate of deceleration, braking distance increases quadratically relative to speed.
- Vehicle load: extra weight makes braking distances longer.
- Road conditions: wet roads offer less adhesion between road surface and tires and therefore result in longer braking distances.
- Tire condition: insufficient tread depth increases braking distances, particularly on wet road surfaces.
- Condition of brakes: oil on the brake pads/shoes, for example, reduces the friction between the pads/shoes and the disk/drum. The lower braking force thus available results in longer braking distances.
- Fading: The braking power also diminishes due to the brake components overheating.

The greatest rates of acceleration or deceleration are reached at the point when the motive or braking force is at the highest level possible without the tires starting to lose grip (maximum traction).

The rates actually achievable under real conditions, however, are always slightly lower because the vehicle's wheels are not all at the point of maximum adhesion at precisely the same moment. Electronic traction, braking and vehicle-handling control systems (TCS, ABS and ESP) are active around the point of maximum force transmission.

Dynamics of lateral motion

Response to crosswinds

Strong crosswinds can move a vehicle off course, especially if it is traveling at a high speed and its shape and dimensions present a large surface area for the wind to catch (Fig. 1). Sudden crosswind gusts such as may be encountered when exiting a road cutting can cause substantial sideways movement (yaw) of high-sided vehicles. This happens too quickly for the driver to react and may provoke incorrect driver response.

When a vehicle is driving through a crosswind, the wind force, F_W, produces a lateral component in addition to the longitudinal aerodynamic drag, F_L. Although its effect is distributed across the entire body surface, it may be thought of as a single force, the lateral wind force, F_{SW}, acting at a single point of action "D". The actual location of the point of action is determined by the vehicle's body shape and angle of incidence α of the wind.

1 Vehicle in crosswind

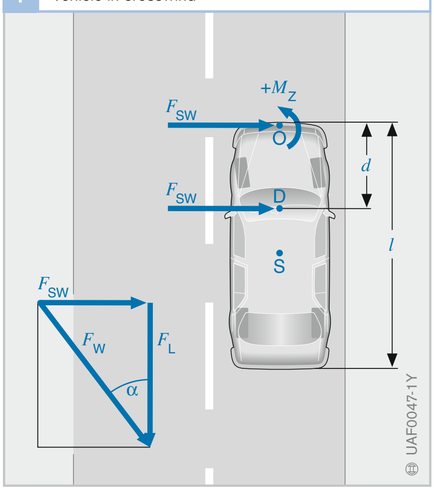


Fig. 1
D Point of action
O Reference point
S Center of gravity
F_W Wind force
F_L Aerodynamic drag
F_{SW} Lateral wind force
M_Z Yaw moment
α Angle of incidence
l Vehicle length
d Distance of point of action, D, from reference point, O
F_S and M_Z acting at O corresponds to F_S acting at D (in aerodynamics it is normal to refer to dimensionless coefficients instead of forces)

The point of action is generally in the front half of the vehicle. On conventionally shaped saloon cars ("three-box" design) it is largely static and is closer to the center of the vehicle than on vehicles with a more streamlined body shape (sloping back), where it can move according to the angle of incidence of the wind.

The position of the center of gravity, S, on the other hand depends on the size and distribution of the vehicle load. In view of these variable factors, therefore, in order to arrive at a general representation of the effect of a crosswind (that is not affected by the relative position of the wheels and suspension to the body), a reference point 0 on the center line of the vehicle at the front is adopted.

When specifying lateral wind force at a reference point other than the true point of action, the turning force of the crosswind around the point of action, that is the yaw moment, M_Z, must also be considered. The crosswind force is resisted by the lateral cornering forces at the wheels. The degree of lateral cornering force which a pneumatic tire can provide depends on various factors in addition to lateral slip angle and wheel load, such as tire design and size, tire pressure and the amount of grip afforded by the road surface.

A vehicle will have good directional stability characteristics in a crosswind if the point of action is close to the vehicle's center of gravity. Vehicles that tend to oversteer will deviate less from their course in a crosswind if the point of action is forward of the center of gravity. The best position for the point of action on vehicles with a tendency to understeer is slightly behind the center of gravity.

Understeer and oversteer

Cornering forces between a rubber-tired wheel and the road can only be generated when the wheel is rotating at an angle to its plane. A lateral slip angle must therefore be present. A vehicle is said to understeer when, as lateral acceleration increases, the lateral slip angle at the front axle increases more than it does at the rear axle. The opposite is true of a vehicle which oversteers (Fig. 2).

For safety reasons, vehicles are designed to slightly understeer. As a result of drive slip, however, a front-wheel drive vehicle can quickly change to sharply understeer or a rear-wheel drive vehicle to oversteer.

Centrifugal force while cornering

Centrifugal force, F_{cf}, acts at the center of gravity, S, (Fig. 3). Its effect depends on a number of factors such as

- the radius of the bend,
- the speed of the vehicle,
- the height of the vehicle's center of gravity,
- the mass of the vehicle,
- the track of the vehicle,
- the frictional characteristics of the tire and road surface (tire condition, type of surface, weather conditions), and
- the load distribution in the vehicle.

Potentially hazardous situations will occur when cornering if the centrifugal force reaches a point where it threatens to overcome the lateral forces at the wheels and the vehicle cannot be held on its intended course. This effect can be partially counteracted by positive camber or banked corners.

If the vehicle slips at the front wheel, it understeers; if it slips at the wheel axle, it oversteers. In both cases the Electronic Stability Program (ESP) detects an undesirable rotation about the vertical axle. By active intervention in the form of selective braking of individual wheels, it is then able to correct the imbalance.

2 Vehicle oversteer and understeer

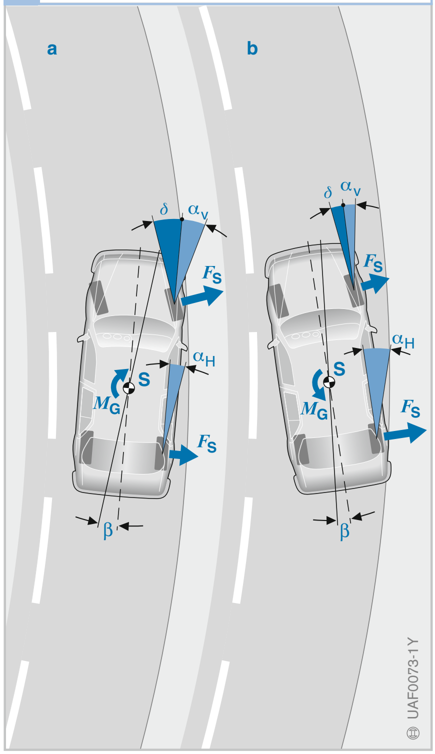


Fig. 2
a Understeer
b Oversteer
α_V Front lateral slip angle
α_H Rear lateral slip angle
δ Steering angle
β Side-slip angle
F_S Lateral force
M_G Yaw moment

3 Centrifugal force while cornering

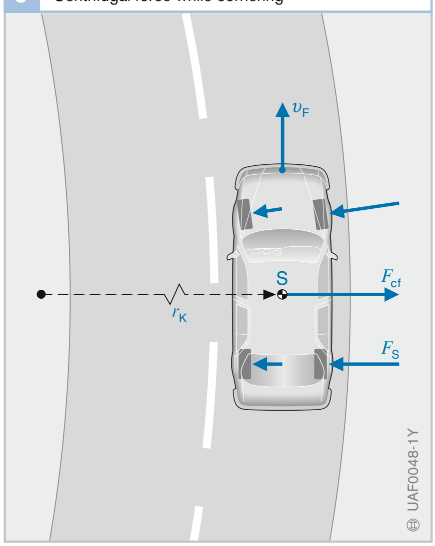


Fig. 3
F_{cf} Centrifugal force
v_F Vehicle speed
F_S Lateral force at individual wheels
r_K Radius of bend
S Center of gravity

Definitions

Braking sequence

As defined in ISO 611, the term "braking sequence" refers to all operations that take place between the point at which operation of the (brake) actuation device begins and the point at which braking ends (when the brake is released or the vehicle is at a standstill).

Variable braking

A type of braking system which allows the driver at any time to increase or reduce the braking force to a sufficiently precise degree by operating the actuation device within its normal effective range.

If operating the actuation device in a particular manner increases the braking force, then the opposite action must reverse the effect and reduce the braking force.

Braking-system hysteresis

Braking system hysteresis is the difference between the actuating forces when the brake is applied and released at a constant braking torque.

Brake hysteresis

Brake hysteresis is the difference between the application forces when the brake is actuated and released at a constant braking torque.

Forces and torques

Actuating force

The actuating force, F_C, is the force that is applied to the actuation device.

Application force

On a friction brake, the application force is the total force exerted on the brake-pad mount, together with attached friction material, in order to generate the friction required for the braking force.

Total braking force

The total braking force, F_f, is the sum total of braking forces at each of the wheels that are produced by the effect of the braking system and which oppose the vehicle's motion or its tendency to move.

Braking torque

The braking torque is the product of the frictional forces generated in the brake by the application forces and the distance of the point of action of those forces from the axis of rotation of the wheel.

Braking-force distribution

The braking-force distribution indicates in terms of percentage share how the total braking force, F_f, is distributed between the front and rear wheels, e.g. front wheels 60%, rear wheels 40%.

External brake coefficient, C

The external brake coefficient, C, is the ratio of the output torque to the input torque or the output force to the input force of a brake.

Internal brake coefficient, C*

The internal brake coefficient, C*, is the ratio of the total tangential force acting at the effective radius of a brake to the application force, F_S.

Typical values: for drum brakes, values of up to $C^* = 10$ may be obtained, for disc brakes $C^* \approx 1$.

Time periods

The braking sequence is characterised by a number of time periods which are defined with reference to the ideal curves shown in Figure 1.

Period of movement of actuation device

The period of movement of the actuation device is the time from the point at which force is first applied to the actuation device (t_0), to the point at which it reaches its final position (t_3) as determined by the actuating force or the actuation travel. The same applies by analogy to the release of the brakes.

Response time
The response time, t_a, is the time that elapses from the point at which force is first applied to the actuation device to the point at which braking force is first produced (pressure generated in the brake lines) ($t_1 - t_0$).

Pressure build-up time
The pressure build-up time, t_s, is the time from the point at which braking force is first produced to the point at which the pressure in the brake lines reaches its highest level ($t_5 - t_1$).

Total braking time
The braking time, t_b, is the time that elapses from the point at which force is first applied to the actuation device to the point at which braking force ceases ($t_7 - t_0$). If the vehicle comes to a halt, then the moment at which the vehicle is first stationary is the moment at which the braking time ends.

Effective braking time
The effective braking time, t_w, is the time that elapses from the moment at which braking force is first produced to the moment at which braking force ceases ($t_7 - t_2$). If the vehicle comes to a halt, then the moment at which the vehicle is first stationary is the moment at which the effective braking time ends.

Distances
Braking distance
The braking distance, s_1, is the distance travelled by a vehicle during the period of the effective braking time ($t_7 - t_2$).

Total braking distance
The total braking distance s_0 is the distance travelled by a vehicle during the period of the total braking time ($t_7 - t_0$). That is the distance travelled from the point at which the driver first applies force to the actuation device to the point at which the vehicle is at a standstill.

Braking deceleration
Momentary deceleration
The momentary deceleration, a, is the quotient of the reduction in speed and the elapsed time.
$a = dv/dt$

Average deceleration over the total braking distance
From the vehicle speed v_0 at the time t_0, the average deceleration, a_{ms}, over the stopping distance, s_0, is calculated using the formula
$a_{ms} = v_0^2/2s_0$

Mean fully developed deceleration
The figure for mean fully developed deceleration, a_{mft}, represents the average deceleration during the period in which deceleration is at its fully developed level ($t_7 - t_6$).

Braking factor
The braking factor, Z, is the ratio between total braking force, F_f, and total static weight, G_S, (vehicle weight) acting on the axle or axles of the vehicle. That is equivalent to the ratio of braking deceleration, a, to gravitational acceleration, g ($g = 9.81$ m/s²).

1 Vehicle braking sequence to the point of standstill (ideal case)

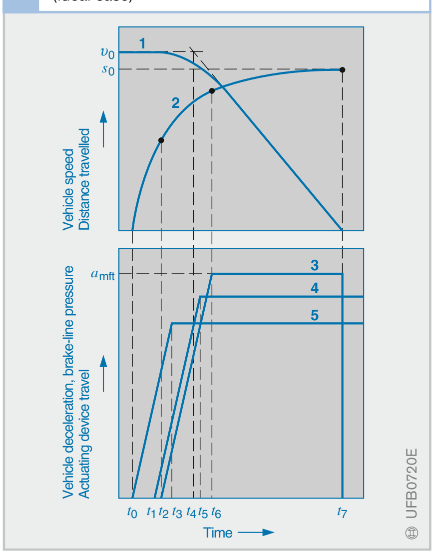


Fig. 1
1 Vehicle speed
2 Distance travelled while braking
3 Vehicle deceleration
4 Brake-line pressure (brake pressure)
5 Actuation device travel
t_0 Time at which the driver first applies force to actuation device
t_1 Brake-line pressure (brake pressure) starts to rise
t_2 Vehicle deceleration begins
t_3 Actuation device has reached intended position
t_4 Intersection of extended speed curve sections
t_5 Brake-line pressure has reached stabilised level
t_6 Vehicle deceleration has reached stabilised level
t_7 Vehicle comes to a halt

Car braking systems

Braking systems are indispensable for the roadworthiness and safe operation of a motor vehicle in road traffic conditions. They are therefore subject to strict legal requirements. The increasing effectiveness and sophistication demanded of braking systems over the course of time has meant that the mechanical systems have been continually improved. With the advent of microelectronics, the braking system has become a complex electronic system.

Overview

Car braking systems must perform the following fundamental tasks:

- Reduce the speed of the vehicle
- Bring the vehicle to a halt
- Prevent unwanted acceleration during downhill driving
- Keep the vehicle stationary when it is stopped

The first three of those tasks are performed by the service brakes. The driver controls the service brakes by operating the brake pedal. The parking brake ("hand brake") keeps the vehicle stationary once it is at a standstill.

Conventional braking systems

On conventional braking systems, the braking sequence is initiated exclusively by means of force applied to the brake pedal. In the braking system's master cylinder, that force is converted into hydraulic pressure. Brake fluid acts as the transmission medium between the master cylinder and the brakes (Figure 1).

On power-assisted braking systems such as are most frequently used on cars and light commercial vehicles, the actuation pressure is amplified by a brake servo unit (brake booster).

1 Example of a power-assisted car braking system

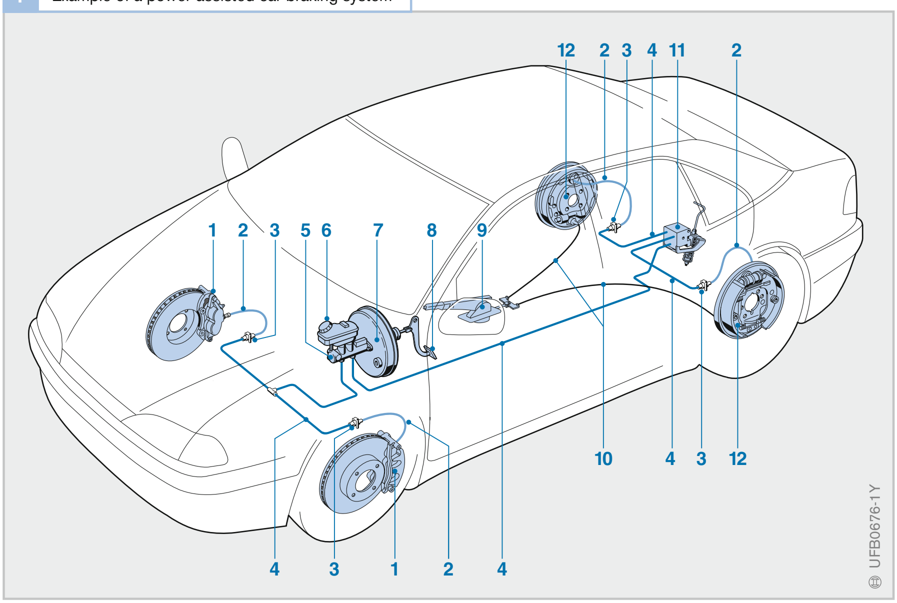


Fig. 1

1 Front brake (disc brake)
2 Brake hose
3 Connecting union between brake pipe and brake hose
4 Brake pipe
5 Master cylinder
6 Brake-fluid reservoir
7 Brake servo unit
8 Brake pedal
9 Handbrake lever
10 Brake cable (parking brake)
11 Braking-force reducer
12 Rear brake (drum brake in this case)

K. Reif (Ed.), *Fundamentals of Automotive and Engine Technology*,
DOI 10.1007/978-3-658-03972-1_11, © Springer Fachmedien Wiesbaden 2014

Electronic braking systems

Antilock braking system (ABS)

An electronic braking system was first used on a volume-production vehicle in 1978. ABS (**A**ntilock **B**raking **S**ystem) prevents the wheels locking up and the vehicle becoming uncontrollable under heavy braking.

As with conventional systems, an ABS system has a mechanical link between the brake pedal and the brakes. But it also incorporates an additional component, the hydraulic modulator. Solenoid valves in the hydraulic modulator are controlled in such a way that if the degree of wheel slip exceeds a certain amount, the brake pressure in the individual wheel cylinders is selectively limited to prevent the wheels locking.

ABS has been continually improved and developed to the extent that it is now standard equipment on virtually all new vehicles sold in western Europe.

Electrohydraulic brakes (SBC)

SBC (**S**ensotronic **B**rake **C**ontrol) represents a new generation of braking systems. Under normal operating conditions, it has no mechanical link between the brake pedal and the wheel cylinders. The SBC electrohydraulic system detects the brake pedal travel electronically using duplicated sensor systems and analyses the sensor signals in an ECU. This method of operation is sometimes referred to as "brake by wire". The hydraulic modulator controls the pressure in the individual brakes by means of solenoid valves. Operation of the brakes is still effected hydraulically using brake fluid as the transmission medium.

Electromechanical brakes (EMB)

In the future there will be another electronic braking system, EMB (**E**lectro**m**echanical **B**rakes), which will operate electromechanically rather than employing hydraulics. Electric motors will force the brake pads against the discs in order to provide the braking action. The link between the brake pedal and the brakes will be purely electronic.

Electronic vehicle-dynamics systems

Continuing development of the ABS system led to the creation of TCS (**T**raction **C**ontrol **S**ystem). This system, which was first seen on volume-production cars in 1987, prevents wheel spin under acceleration and thus improves vehicle handling. Consequently, it is not a braking system in the strict sense of the word. Nevertheless, it makes use of and actively operates the braking system to prevent a wheel from spinning.

Another vehicle-dynamics system is the ESP (**E**lectronic **S**tability **P**rogram), which prevents the vehicle entering a skid within physically determined parameters. It too makes use of and actively controls the braking system in order to stabilise the vehicle.

Ancillary functions of electronic systems

Electronic processing of data also makes possible a number of ancillary functions that can be integrated in the overall electronic braking and vehicle-dynamics systems.

- **B**rake **A**ssistant (BA) increases brake pressure if the driver is hesitant in applying the full force of the brakes in an emergency.
- **E**lectronic **B**raking-force distribution controls the braking force at the rear wheels so that the best possible balance between front and rear wheel braking is achieved.
- **H**ill **D**escent **C**ontrol (HDC) automatically brakes the vehicle on steep descents.

History of the brake

Origin and development

The first use of the wheel is dated to 5,000 B.C. Usually, cattle were used as draft animals; later, horses and donkeys were also used. The invention of the wheel made it necessary to invent the brake. After all, a horse-drawn carriage traveling downhill had to be slowed down, not only to keep its speed within controllable limits, but also to prevent it running into the back of the horses. This was likely done using wooden rods braced against the ground or the wheel disks. Beginning around 700 B.C., wheels acquired iron tires to prevent premature wear of the wheel rim.

Beginning in 1690, coach drivers used a "chock" to brake their carriages. While driving downhill, they used its handle push it under a wheel, which then was immobilized and slid onto the chock.

In 1817, at the dawn of the industrial age, Baron Karl Drais rode from Karlsruhe in southern Germany to Kehl, proving to a stunned public that it is possible to ride on two wheels without falling over. As he surely had difficulty stopping when driving downhill, his last, 1820 model featured a friction brake on the rear wheel (Fig. 1).

Finally, in 1850, the iron axle was introduced in carriage construction, along with the shoe brake. In this type of brake, brake shoes were pressed against the metallic running surface of the iron-coated wooden wheels. The shoe brake could be operated from the driver's seat with the aid of a crank handle and a gear linkage (Fig. 2).

The low speed and sluggish drive train of the first automobiles did not place any great demands on the effectiveness of the brakes. In the early days, the shoe, band and wedge brakes, which were manually or foot-operated using levers, hinges and cables on the fixed rear axles, were sufficient for this purpose.

At first, the rear wheels were braked; occasionally, an intermediate shaft or only the cardan shaft was braked. Only about 35 years after the automobile was invented were the front wheels equipped with (cable-operated) brakes. Even more years passed before automobiles began to be equipped with hydraulically operated brakes, which, at the time, were only drum brakes. Use of the old method of cable activation continued in a few models, such as the VW Beetle, until after World War II. Other important milestones were the use of disk brakes and, in the present era, the introduction and incremental development of various driving stability systems.

1 Baron Karl Drais' wheel with friction brake, 1820

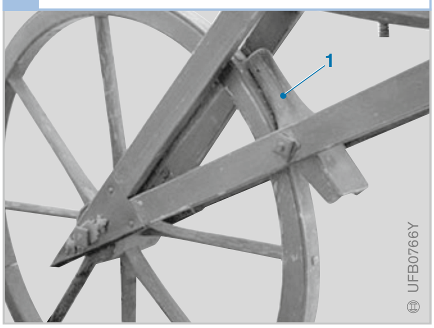
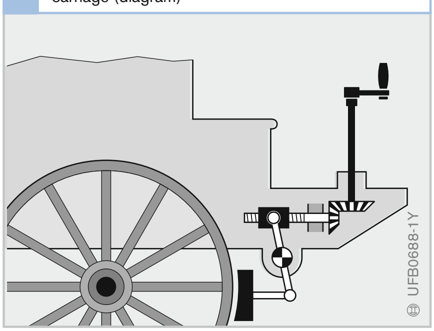


Fig. 1
1 Friction brake on wheel

2 Shoe brake with crank and linkage on a carriage (diagram)



Shoe and external shoe brakes on the wheel running surfaces

The first motor vehicles drove on wooden wheels with steel or rubber tires, or rubber-tired, spoked steel wheels. For braking, levers (as for the horse-drawn carriages) pushed brake shoes or external shoe brakes with friction linings against the running surfaces of the rear wheels. An initial example is the "riding carriage" developed by Gottlieb Daimler as an experimental vehicle in 1885 (the first motorcycle, with an engine performance of 0.5 horsepower and a top speed of 12 km/h). A cable led from the brake actuating lever, located at the front, close to the steering arm, to the *external shoe brake* on the rear wheel (Figures 3a, b).

In 1886, the first passenger cars with internal combustion engines were introduced: the Daimler motor carriage (1.1 hp, 16 km/h), which was derived from the horse carriage, and the Benz motorcar, which was newly designed as an automobile. Both of them had shoe brakes, as did the world's first truck, built in 1896. The shoe brake was installed in front of the rear wheels of each vehicle (Figures 3c, d, e, f.).

3 Historic motor vehicles and their wheel brakes (examples)

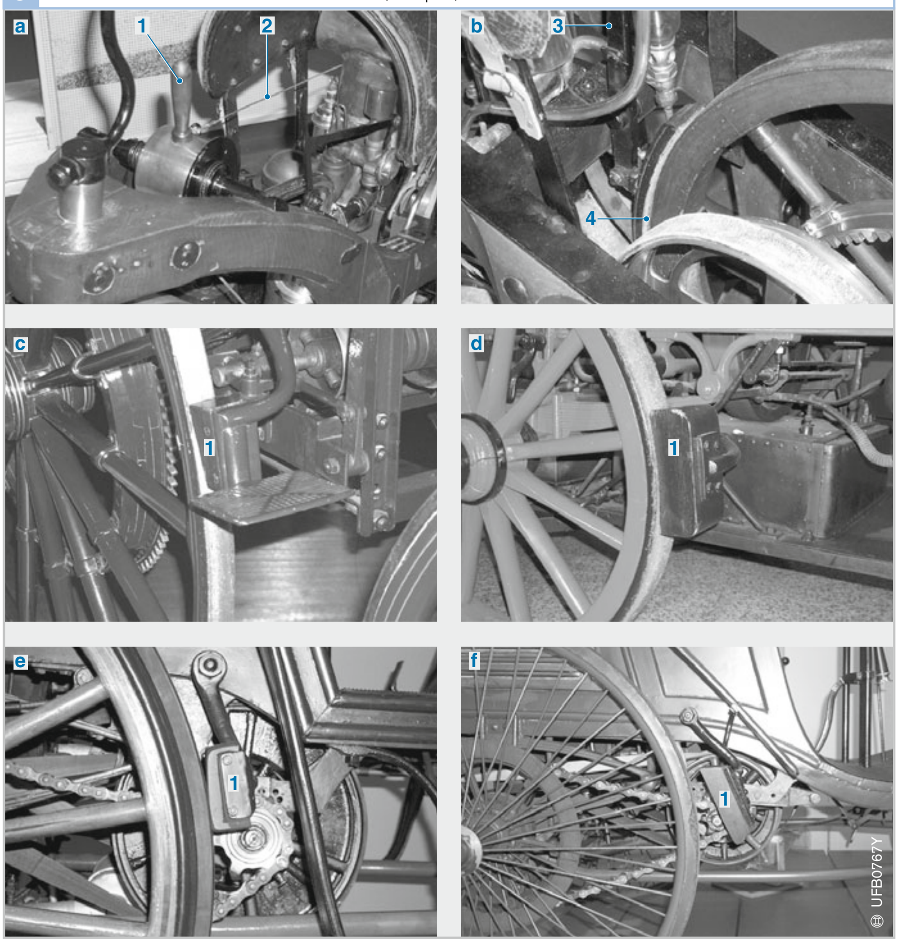


Fig. 3
a, b Daimler riding carriage 1885
1 Brake actuating lever
2 Cable to brake lever
3 Brake lever
4 External shoe brake on rear wheel

c Daimler motor carriage, 1886
1 Shoe brake, which also braked in "automatic" state when the flanged step was stepped on

d Daimler fire truck, 1890
1 Shoe brake

e Benz Viktoria, 1893
1 Shoe brake

f Benz Velo, 1894
1 Shoe brake

4 Daimler steel-wheeled car with band brake, 1889

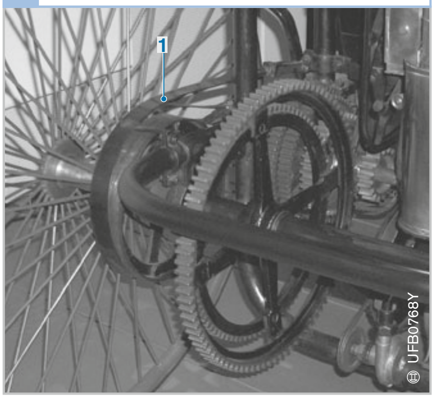


Fig. 4
1 Band brake on the rear axle

5 Daimler Phoenix, 1889, drive shaft (front view)

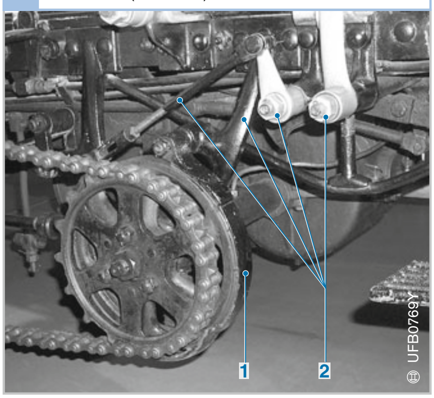


Fig. 5
1 External shoe brake, front section
2 Brake lever and brake linkage

6 Daimler Phoenix, 1889, drive shaft (rear view)

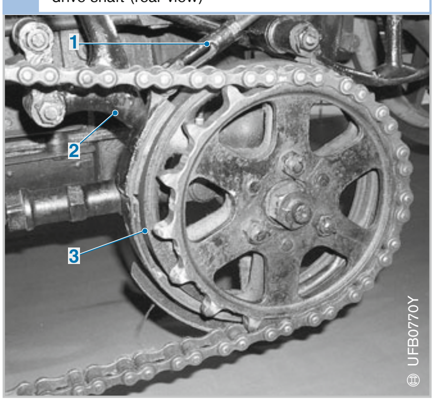


Fig. 6
1 Brake rod
2 Brake lever
3 External shoe brake, rear section

Band and external shoe brakes

As solid rubber tires quickly became established for motor vehicles (beginning with the triangular Benz motorcar in 1886 and the Daimler steel-wheeled car of 1889) and were soon replaced by air-filled rubber tires for a more comfortable ride, the era of the shoe brake in automobiles had already come to an end.

From then on, *band brakes* (flexible steel brake bands that braked either directly or via several brake shoes riveted to the inside) or *external shoe brakes* (rigid cast iron or steel brake shoes with brake linings) began to be used. These pedal-operated brakes worked with external brake drums that were normally installed at the front on the intermediate drive shaft or on the drive axle in the rear wheel area.

For example, the Fahrzeugfabrik Eisenach produced the first Wartburg motorcar in 1898. Model 1 featured an exposed transmission and drive pinions. *Band brakes* braked both the axle drive and the two rear wheels.

In 1899, the Daimler steel-wheeled car had solid rubber tires and *steel band brakes* on the rear wheels (Fig. 4). The Daimler “Phoenix”, also dating from 1899, still had solid rubber tires, but these were soon replaced by air-filled tires. A footbrake acted as an *external shoe brake* on the front drive shaft (Figures 5 and 6), and the handbrake acted on the rear wheels. In addition, this car featured – as did, for example, the Benz racing car of 1899 (Fig. 7) – a “sprag brake”, a strong rod mounted on the rear that had the purpose of being driven into the (usually relatively soft) road.

An excerpt from the original text of the user manual for the “Phaeton” by Benz & Co. Rheinische Gasmotoren-Fabrik A.G. Mannheim from 1902 reads as follows: “In addition to a handbrake attached to its left side, the car is braked primarily by depressing the left foot pedal, which acts as a band brake on the brake disks fastened to the two rear wheels. Simultaneously, as mentioned above, the belt is automatically moved out. To stop the car immediately, both the left and the right pedals are depressed at the same time, which causes the band brake connected to the right pedal to act on the brake disk and thus brake the reduction gear."

Internal shoe drum brakes with mechanical cable activation

Over time, vehicles became faster and heavier. Therefore, they required a more effective brake system. Thus band and external shoe brakes soon gave way to the *internal shoe drum brake,* for which Louis Renault applied for a patent in 1902. A spreading mechanism pushed two crescent-shaped brake shoes against the inner surface of the cast iron or steel brake drums, which were connected to the wheel. Due to its self-reinforcing effect, the drum brake features low operating forces compared to the braking forces, long maintenance periods and long-lasting linings.

At first, the braking force was transmitted to the two drum brakes of the rear wheels using brake cables.

For example, the Mercedes Simplex already featured additional, *cable-operated rear wheel drum brakes* (Fig. 8) in addition to the cardan shaft band brake. Due to higher engine performance (40 horsepower), a second footbrake (Fig. 9) was added, which also acted as a *band brake* on the intermediate shaft of the chain drive. By the way, all four brakes were cooled by a water spray which, during braking, dripped onto the friction surfaces from a reservoir.

Beginning in about 1920, vehicles were fitted with *drum brakes on all four wheels.* The braking force was still transmitted using mechanical means, i.e. *levers, joints and brake cables.*

7 Benz racecar, 1899, with external band brake and suspended "sprag brake"

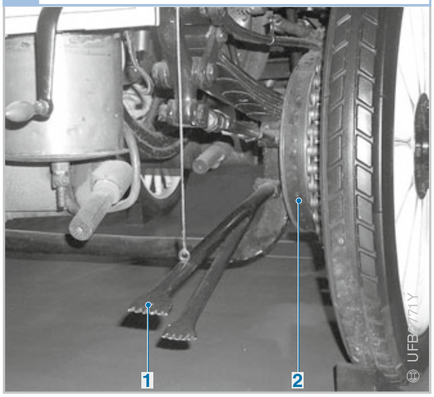


Fig. 7
1 "Sprag brake"
2 External band brake with brake shoes riveted to the inside

8 Daimler-Simplex, 1902, with cable-operated drum brake on the rear wheel

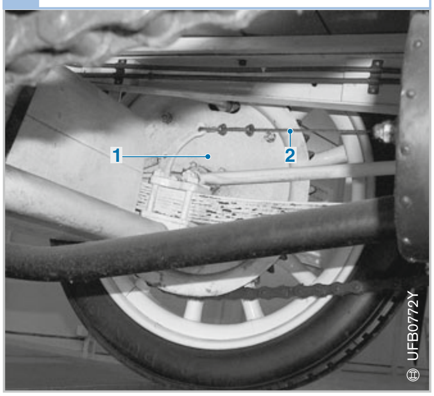


Fig. 8
1 Drum brake
2 Bowden cable

9 Daimler-Simplex, 1902, pedal and lever mechanism on the driver's seat

These cable-operated drum brakes remained in use for a long time. One example was the standard VW model of the 1950s (Fig. 10):

The primary element of this brake system was a brake pressure rail (Item 1). The four brake cables (2) attached to this element ran backwards through cable sleeves to the wheel brakes (drum brakes) of the four wheels (3). The rear part of the rail was supported by a short lever that sat on the brake pedal shaft. When the brake pedal of the footbrake (4) was depressed, the brake pressure rail was pushed forwards along with the four cables. The cables transmitted the force to the wheel brakes.

The lever for the handbrake (5) was further back in the car. However, via a decoupled rod, the handbrake ultimately acted on the same mechanism as the footbrake, and thus likewise acted on all four wheels.

10 Standard model VW, cable brake

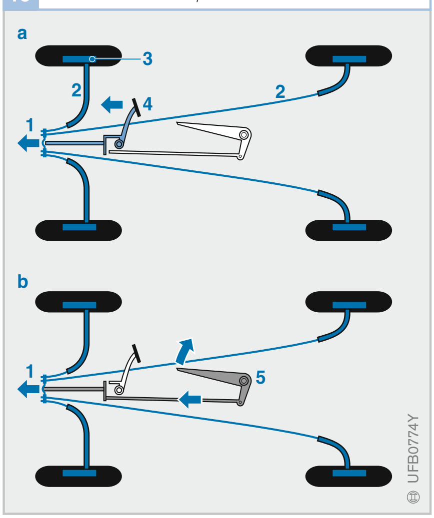


Fig. 10
a Activation of the footbrake
b Activation of the handbrake

1 Brake pressure rail
2 Brake cables
3 Wheel brakes
4 Brake pedal of the footbrake
5 Lever of the handbrake

Hydraulic brake activation

The main problem of the cable brake was the great maintenance effort and the uneven braking effect caused by uneven friction during mechanical transmission.

This was remedied when Lockheed introduced a hydraulically actuated brake in 1919. A special brake fluid now transmitted the brake pedal force uniformly to the actuating cylinders of the wheel brakes over metal lines and hoses, without the need for levers, joints and cables.

Hydraulic brake activation also made it possible to amplify the foot pressure applied by the driver by using intake manifold depression as a source of power for a brake servo system. The principle was patented in 1919 by Hispano-Suiza.

On commercial vehicles and railway rolling stock, air brakes established themselves as the system of choice.

In 1926, the “Adler Standard” was the first car in Europe to be equipped with a hydraulic brake system. The first hydraulic braking force reinforcement in auto racing were used in the Mercedes-Benz “Silver Arrows” in 1954. This ultimately became standard equipment for many mass-production vehicles.

Because a possible failure of the brake circuit could completely disable the early single-circuit brakes, the dual-circuit brake was later prescribed by law. According to VW Golf developer Dr. Ernst Fiala, the early “Beetles” (the standard model VWs) still had a cable-operated brake for that very reason: at the time it was feared that a hose in the hydraulic brakes could explode. Later, however – if only for competitive reasons – the VW Export and VW Transporter featured hydraulic braking systems.

Disk brake

Although British automaker Lancaster had patented the disk brake in 1902, it was a long time until this type of brake was introduced. Not until some fifty years later, beginning in 1955, did the legendary Citroën DS-19 become the first mass-produced car to be fitted with disk brakes. The disk brake was derived from the multi-plate brake and was initially developed for the aircraft industry.

In the disk brake, one brake lining presses the brake disk from the inside and outside. The brake disk (which is normally made of cast iron or, less commonly, of steel) is connected to the wheel. Its advantage is its simple and easy-to-assemble structure. It also counteracts the reduction in braking effect caused by overheating and prevents misalignment of the wheels of an axle.

The first German car with disk brakes on the front wheels was the BMW 502 in 1959. The first German cars to have disk brakes on all four wheels were the Mercedes 300 SE, the Lancia Flavia and the Fiat 2300 in 1961. Today, virtually all cars have a disk braking system, at least on the front wheels. In 1974, the first Formula 1 racecars with carbon fiber composite brake disks were introduced. These disks are considered especially light and heat resistant and thus have gained widespread use in motorsports and aviation.

Brake pads and shoes

Suitable brake linings had to be developed for drum and disk brakes, for which asbestos proved to be particularly effective. Not until it became known that asbestos fibers were harmful to health was the material replaced by plastic fiber.

Driving stability systems

The age of electronic brake systems dawned in 1978 with the arrival of the antilock braking system (ABS) for cars developed by Bosch. During braking, ABS provides early detection of the incipient lock of one or more wheels and prevents wheel locking. It ensures the steerability of the vehicle and substantially reduces the danger of skidding. In 1986, it was followed by the **t**raction **c**ontrol **s**ystem (TCS) with which Bosch extended system capability to the control of wheel spin under acceleration.
Fig. 11 shows road tests of these systems on the Bosch proving grounds in Boxberg in southern Germany.

As a further improvement of driving safety, Bosch introduced the **e**lectronic **s**tability **p**rogram (ESP) in 1995, which integrates the functions of ABS and TCS. It not only prevents the vehicle wheels from locking and spinning, it also keeps the vehicle from pulling to the side. Alternative systems, such as four-wheel steering and rear-axle kinematics, which were developed in the 1980s and 90s and were installed in some mass-production vehicles, did not catch on because they weighed too much, cost too much or were not effective enough.

Meanwhile, the (electrohydraulic) sensotronic brake control has found its place in automobile construction. It provides all of the ESP functions and decouples the mechanical operation of the brake pedal by means of an electronic control system. For safety purposes, a hydraulic fallback system is automatically available.

11 Steep uphill drive in the Bosch proving grounds in Boxberg for testing the driving stability systems of passenger cars and commercial vehicles

Classification of car braking systems

The entirety of the braking systems on a vehicle is referred to as braking equipment. Car braking systems can be classified on the basis of
- design and
- method of operation

Designs

Based on legal requirements, the functions of the braking equipment are shared among three braking systems:
- the service brakes,
- the secondary-brake system, and
- the parking brake

On commercial vehicles, the braking equipment also includes a continuous-operation braking system (e.g. retarder) that allows the driver to keep the vehicle at a steady speed on a long descent. The braking equipment of a commercial vehicle also includes an automatic braking system that operates the brakes of a trailer if it is detached from its towing vehicle either deliberately or by accident.

Service brakes

The service-brake system ("foot brake") is used to slow down the vehicle, to keep its speed constant on a descent, or to bring it completely to a halt.

The driver can infinitely vary the braking effect by means of the pressure applied to the brake pedal.
The service-brake system applies the brakes on all four wheels.

Secondary-brake system

The secondary-brake system must be capable of providing at least some degree of braking if the service-brake system fails. It must be possible to infinitely vary the level of braking applied.

The secondary-brake system does not have to be an entirely separate third braking system (in addition to the service brakes and the parking brake) with its own separate actuation device. It can consist of the remaining intact circuit of a dual-circuit service-brake system on which one circuit has failed, or it can be a parking-brake that is capable of graduated application.

Parking-brake system

The parking brake (hand brake) performs the third function required of the braking equipment. It must prevent the vehicle from moving when stopped or parked, even on a gradient and when the vehicle is unattended.

According to the legal requirements, the parking-brake system must also have an unbroken mechanical link, e.g. a rod linkage or a cable, between the actuation device and the brakes that it operates.

The parking-brake system is generally operated by means of a hand-brake lever positioned near the driver's seat, or in some cases by a foot pedal. This means that the service and parking-brake systems of a motor vehicle have separate actuation devices and means of force transmission.

The parking-brake system is capable of graduated application and operates the brakes on one pair of wheels (front or rear) only.

Methods of operation
Depending on whether they are operated entirely or partially by human effort or by another source of energy, braking systems can be classed either as
- muscular-energy (unassisted) braking systems,
- power-assisted braking systems, or
- power-brake systems,

Muscular-energy braking systems
On this type of braking system frequently found on cars and motorcycles, the effort applied to the brake pedal or hand-brake lever is transmitted to the brakes either mechanically (by means of a rod linkage or cable) or hydraulically. The energy by which the braking force is generated is produced entirely by the physical strength of the driver.

Power-assisted braking systems
The power-assisted braking system is the type most commonly used on cars and light commercial vehicles.
It amplifies the force applied by the driver by means of a brake servo which utilises another source of energy (vacuum or hydraulic power). The amplified muscle power is transmitted hydraulically to the brakes.

Power-brake systems
Power-brake systems are generally used on commercial vehicles but are occasionally fitted on large cars in conjunction with an integral ABS facility. This type of braking system is operated entirely without muscular-energy.

The system is operated by hydraulic power (which is based on fluid pressure) transmitted by hydraulic means. The brake fluid is stored in energy accumulators (hydraulic accumulators) which also contain a compressed gas (usually nitrogen). The gas and the fluid are kept apart by a flexible diaphragm (diaphragm accumulator) or a piston with a rubber seal (piston accumulator). A hydraulic pump generates the fluid pressure, which is always in equilibrium with the gas pressure in the energy accumulator. A pressure regulator switches the hydraulic pump to idle as soon as the maximum pressure is reached.

Since brake fluid can be regarded as practically incompressible, small volumes of brake fluid can transmit large brake-system pressures.

Components of a car braking system

Figure 1 shows the schematic layout of a car braking system. It consists of the following main component groups:

- Energy supply system,
- Actuation device,
- Force transmission system, and
- Wheel brakes.

Energy supply system

The energy supply system encompasses those parts of the braking system that provide, control and (in some cases) condition the energy required to operate the brakes. It ends at the point where the force transmission system begins, i.e. where the various circuits of the braking system are isolated from the energy supply system or from each other.

Car braking systems are in the main power-assisted braking systems in which the physical effort of the driver, amplified by the vacuum in the brake servo unit, provides the energy for braking.

Actuation device

The actuation device encompasses those parts of a braking system that are used to initiate and control the action of that braking system. The control signal may be transmitted within the actuation device, and the use of an additional energy source is also possible.

The actuation device starts at the point at which the actuation force is directly applied. It may be operated in the following ways:

- by direct application of force by hand or foot by the driver,
- by indirect control of force by the driver.

The actuation device ends at the point where distribution of the braking-system energy begins or where a portion of that energy is diverted for the purpose of controlling braking. Among the essential components of the actuation device are the vacuum servo unit and the master cylinder.

Force transmission system

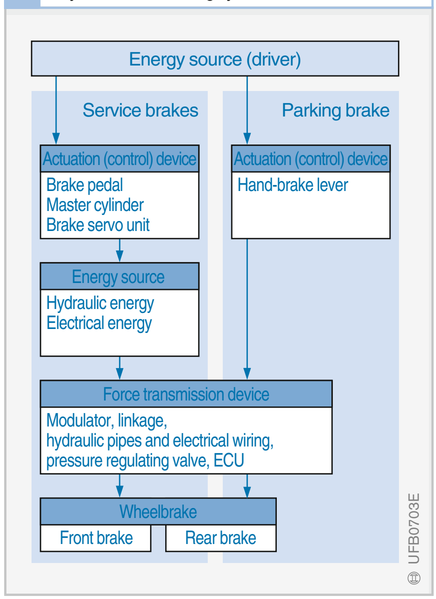
The force transmission system encompasses those parts of the braking system that transmit the energy introduced by the energy supply system(s) and controlled by the actuation device. It starts at the point where the actuation device and the energy supply system end. It ends at the point where it interfaces with those parts of the braking system that generate the forces that inhibit or retard vehicle motion. It may be mechanical or hydromechanical in design.

The components of the force transmission system include the transmission medium, hoses, pipes and, on some systems, a pressure regulating valve for limiting the braking force at the rear wheels.

Wheelbrakes

The wheelbrakes consist of those parts of the braking system in which the forces that inhibit or retard the movement of the vehicle are generated. On car braking systems, they are friction brakes (disc or drum brakes).

1 Layout of a car braking system



Brake-circuit configuration

Legal requirements demand that braking systems incorporate a dual-circuit forcetransmission system.

According to DIN 74 000, there are five ways in which the two brake circuits can be split (Figure 1). It uses the following combinations of letters to designate the five different configurations: II, X, HI, LL and HH. Those letters are chosen because their shapes roughly approximate to the layout of the brake lines connecting the master cylinder and the brakes.

Of those five possibilities, the II and X configurations, which involve the minimum amount of brake piping, hoses, disconnectable joints and static or dynamic seals, have become the most widely established. That characteristic means that the risk of failure of each of the individual circuits due to fluid leakage is as low as it is for a single-circuit braking system. In the event of brake-circuit failure due to overheating of one of the brakes, the HI, LL and HH configurations have a critical weakness because the connection of individual brakes to both circuits means that failure of one brake can result in total failure of the braking system as a whole.

In order to satisfy the legal requirements regarding secondary-braking effectiveness, vehicles with a forward weight-distribution bias are fitted with the X configuration. The II configuration is particularly suited to use on cars with a rearward weight-distribution bias.

II configuration
This layout involves a front-axle/rear-axle split – one circuit operates the rear brakes, the other operates the front brakes.

X configuration
This layout involves a diagonal split – each circuit operates one front brake and its diagonally opposed rear brake.

HI configuration
This layout involves a front/front-and-rear split - one brake circuit operates the front and rear brakes, the other operates only the front brakes.

LL configuration
This arrangement involves a two-front/one-rear split. Each circuit operates both front wheels and one rear wheel.

HH configuration
The circuits are split front-and-rear/front-and-rear. Each circuit operates all four wheels.

1 Brake-circuit configuration

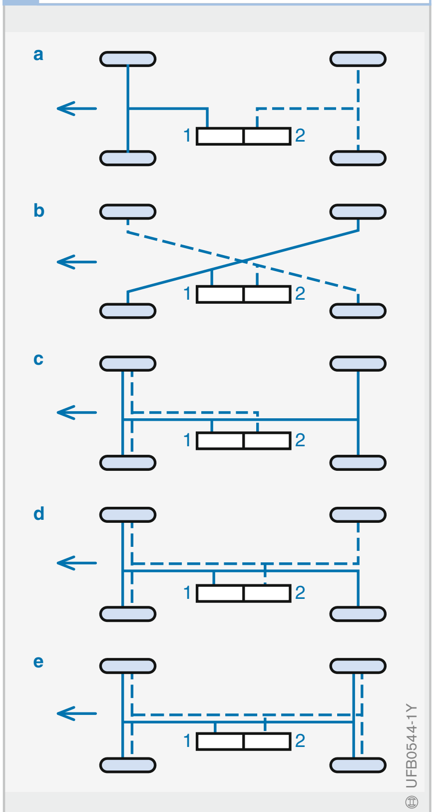


Fig. 1
a II configuration
b X configuration
c HI configuration
d LL configuration
e HH configuration
1 Brake circuit 1
2 Brake circuit 2
← Direction of travel

Vehicle electrical systems

The vehicle electrical system of a motor vehicle comprises the alternator as the energy converter, one or more batteries as the energy accumulators and the electrical equipment as consumers. The energy from the battery is supplied to the starter (consumer), which then starts the vehicle engine. During vehicle operation, the ignition and fuel-injection system, the control units, the safety and comfort and convenience electronics, the lighting, and other equipment have to be supplied with power.

Electrical energy supply in the passenger car

When the engine is running, the alternator supplies electricity which, depending on the voltage level in the vehicle electrical system (determined by the alternator speed and the consumers drawing current), is normally enough to power the consumers and charge the battery as well. If the consumer current draw I_V in the vehicle electrical system is greater than the alternator current I_G (e.g. when the engine is idling), the battery is discharged. The vehicle system voltage falls to the voltage level of the battery from which current is drawn. If the consumer current draw I_V is less than the alternator current output I_G, a proportion of the current flows to the battery and acts as a battery charging current I_B. The vehicle system voltage increases to the setpoint value specified by the voltage regulator.

With careful selection of the battery, alternator, starter and the other electrical system consumers, it must be ensured that the charge balance of the battery is indeed balanced so that:

- It is always possible for the internal-combustion engine to be started
- It allows operation of specific electrical consumers for a reasonable period of time when the engine is off

The lowest temperature at which the engine can be started depends on a number of factors, including the battery (capacity, low-temperature test current, state of charge, internal resistance, etc.) and the starter (design, size, and performance). If the engine is started at a temperature of –20 °C, for example, the battery must have a given minimum state of charge p.

Apart from the battery itself, the current output of the alternator and the power output of the consumers have a decisive influence on the charge balance of the battery.

Current output of the alternator

The current output of the alternator is speed-dependent. For an engine idling

1 Possible starting temperature versus battery charge level

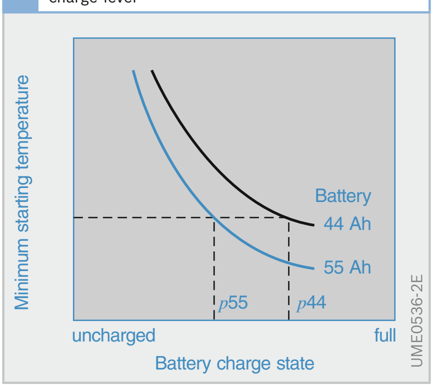


Fig. 1
p Minimum charge level

2 Alternator current output I_G relative to alternator speed

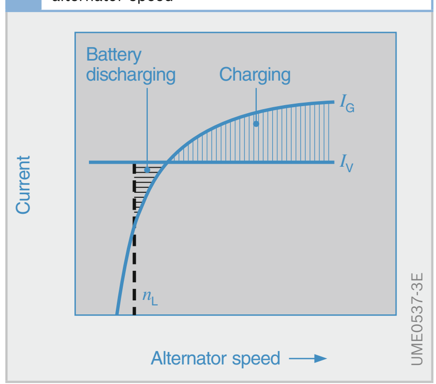


Fig. 2
I_V Consumer current draw
n_L Engine idling speed

K. Reif (Ed.), *Fundamentals of Automotive and Engine Technology*,
DOI 10.1007/978-3-658-03972-1_12, © Springer Fachmedien Wiesbaden 2014

speed of n_L, the alternator can only supply some of its rated current if it has a conventional turns ratio ranging from 1:2 to 1:3 (crankshaft to alternator). By definition, the rated current is output at an alternator speed of 6,000 rpm.

Power output of the consumers

The electrical consumers have a variety of switch-on durations. A distinction is made between continuous consumers (ignition system, fuel injection, etc.), long-time consumers (lighting, rear-window heating, etc.) and short-term consumers (turn signals, stop lamps, etc.).

The use of a number of consumers is dependent on the time of year (air-conditioning system, seat heating). The on-time of electrical radiator blowers depends on the ambient temperature and vehicle operation. In winter, the vehicle is driven with the lights on most of the time.

The electrical load requirements encountered during vehicle operation are not constant. The first minutes after startup are generally characterized by high demand, followed by a sharp drop in electrical load requirements:

- A windshield heater requires up to 2 kW for 1 to 3 minutes after the engine is started in order to de-ice the windshield
- The secondary-air pump, which injects air immediately after the combustion chamber to burn the exhaust gas, runs for about 3 minutes after the engine is started
- Other electrical consumers such as heaters (rear window, seats, mirrors, etc.), fans and lights are switched on for longer or shorter periods depending on the situation, while the engine-management system is in operation all the time

Charging the battery

Due to the chemical processes that take place in the battery, the battery charge voltage has to be higher at low temperatures and lower at high temperatures. The gassing voltage curve shows the maximum voltage at which the battery does not produce gas. A regulator limits the voltage if the alternator current I_G is greater than the sum of the consumer equipment current draw I_V and the temperature-dependent, maximum permissible battery charging current I_B. Regulators are normally mounted on the alternator. If there is a significant deviations between the temperatures of the voltage regulator and the battery electrolyte, it is better to monitor the voltage regulation temperature directly at the battery. The voltage drop across the alternator/battery charging cable can be accounted for by a regulator which

1 Installed consumers taking account of the switch-on duration (examples)

Electrical consumer	Power input	Average electrical load requirements
Motronic, electric fuel-supply pump	250 W	250 W
Radio	20 W	20 W
Side-marker lamp	8 W	7 W
Low beam (dipped beam)	110 W	90 W
License-plate lamp, tail lamp	30 W	25 W
Indicator lamp, instruments	22 W	20 W
Heated rear window	200 W	60 W
Interior heating, fan	120 W	50 W
Electrical radiator ventilator	120 W	30 W
Windshield wiper	50 W	10 W
Stop lamp	42 W	11 W
Turn signal	42 W	5 W
Fog lamps	110 W	20 W
Rear fog lamp	21 W	2 W
Total		
Installed electrical load requirements	1,145 W	
Average electrical load requirements		600 W

Table 1

measures the actual value of the voltage at the battery.

The arrangement of the alternator, battery and consumers influences the voltage drop on the charging cable and thus the charge voltage. The total current $I_G = I_B + I_V$ flows through the charging cable if all electrical consumers are connected to the battery. In response to the relatively high voltage drop, the charge voltage falls proportionately sharply.

If all consumers are connected on the alternator side, the voltage drop is less and the charging voltage is higher. In the process, consumers that are sensitive to voltage peaks or voltage ripple (electronic circuits) may be damaged or suffer malfunctions. Electrical consumers which feature a high power input and relative insensitivity to overvoltage should therefore be connected close to the alternator, whereas voltage-sensitive consumers with a low power input should be connected close to the battery.

Voltage drops can be minimized by suitable conductor cross-sections and good connections with low contact resistance, even after a long service life.

Design of the vehicle electrical system

Dynamic system characteristic curve

The dynamic system characteristic curve maps the relationship between battery voltage and battery current during a driving cycle. The envelopes reflect the interrelationships between the battery, alternator, consumers, temperature, engine speed, and engine/alternator speed ratio. A large area in the envelope means that, with this type of vehicle electrical system, significant voltage fluctuations are occurring in the selected driving cycle and that the battery undergoes more powerful cyclization, i.e. its state of charge experiences powerful temporal changes. The system characteristic curve is specific to every different combination and every set of operating conditions, and is therefore a dynamic curve. Measuring systems can be connected to the battery terminals to plot the dynamic system characteristic curve.

Charge-balance calculation

The charge-balance calculation is used as the basis for defining the design of the alternator and battery. By means of a computer program, the battery charge level is calculated from the consumer load and the alternator output power at the end of a specified driving cycle. *Rush-hour driving* (low engine speeds) combined with *winter operation* (low charging-current input to the battery) is regarded as a normal passenger-car driving cycle. The battery must maintain a stable charge balance even under these very unfavorable conditions for

3 Dynamic system characteristic curve (envelope for urban driving)

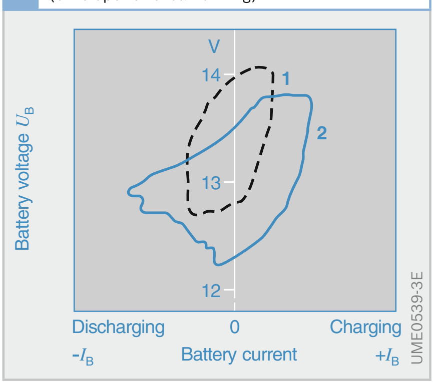
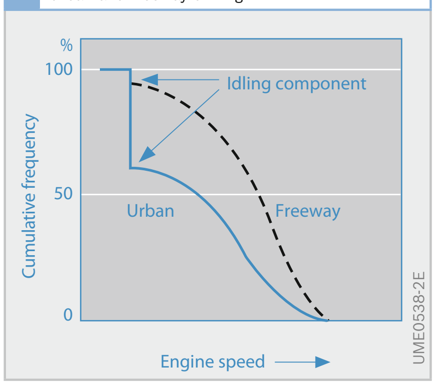


Fig. 3
1 With large alternator and small battery
2 With small alternator and large battery

4 Cumulative frequency of engine speed in urban and freeway driving



the energy balance of the vehicle electrical system.

Driving profile
The driving profile as an input variable for the charge-balance calculation is represented by the cumulative-frequency curve of the engine speed. It describes how often a specific engine speed is reached or exceeded.

In urban driving situations in rush-hour driving, a car's engine is running at idle speed for a large part of the time due to the frequency of stops at traffic lights and consequently high traffic density.

A city bus running on scheduled routes has a high idle percentage time due to interruptions in driving at bus stops. Another factor that has a negative effect on the battery's charge balance is electrical consumers that are operated when the engine is switched off. Long-distance buses generally spend only a small proportion of the time with the engine at idle speed, but on the other hand may have periods when consumers with a high power input are operated with the engine switched off.

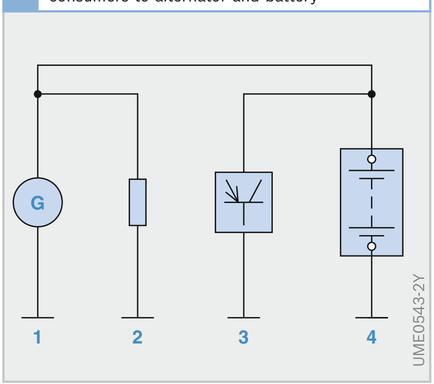
Vehicle electrical system simulation
Contrary to the momentary view using a charge-balance calculation, model-based computer simulations can calculate electrical system power supply status at any time during the trip. They can also include electrical system management systems and assess their effectiveness.

In addition to adjusting current levels in the battery, it is possible to record the vehicle-system voltage characteristic curve and the battery cycle at any time during a trip. Calculations performed using vehicle electrical system simulations are always useful when it comes to comparing electrical-system typologies and assessing the effectiveness of consumers with a high dynamic response or only brief switch-on durations.

Fuel consumption
Since one of the factors that affects the fuel consumption of a vehicle is its mass, the mass of the alternator is also an influence on consumption.

Even the generation of power by the alternator has an effect on fuel consumption: the additional consumption at 100 W of generated electrical output is in the order of 0.17 *l* per 100 km and depends on the efficiency of the alternator. For this reason, alternators with higher mid-range efficiency levels generally contribute to the engine's fuel efficiency despite being slightly heavier in weight.

5 Vehicle electrical system showing connection of consumers to alternator and battery



6 Electrical load requirements versus driving time

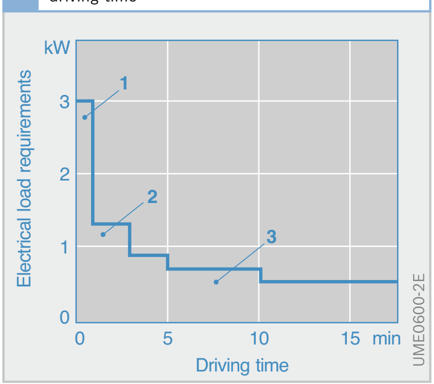


Fig. 5
1 Alternator
2 Consumer with relatively high power input
3 Consumer with low power input
4 Battery

Fig. 6
1 Windshield heater
2 Secondary-air pump
3 Heater, fan, engine management, etc.

Electrical energy management

An electrical energy management (EEM) system coordinates the interaction between alternator, voltage transformer, batteries and electrical consumers when the vehicle is in use. When the vehicle is parked, the EEM monitors the batteries, and switches standstill-draw and constant-draw consumers off as soon as the battery charge reaches a critical limit. The EEM regulates the entire electrical energy balance. It compares the power demand from the consumers with the power available within the vehicle electrical system and maintains a constant balance between supply and power output.

The basis for the EEM is the battery management. The objective of the battery management is to communicate to the EEM information about the current state of the battery and about predicted electrical behavior. Using this information, it is possible to implement operating strategies for increasing vehicle availability and profitability.

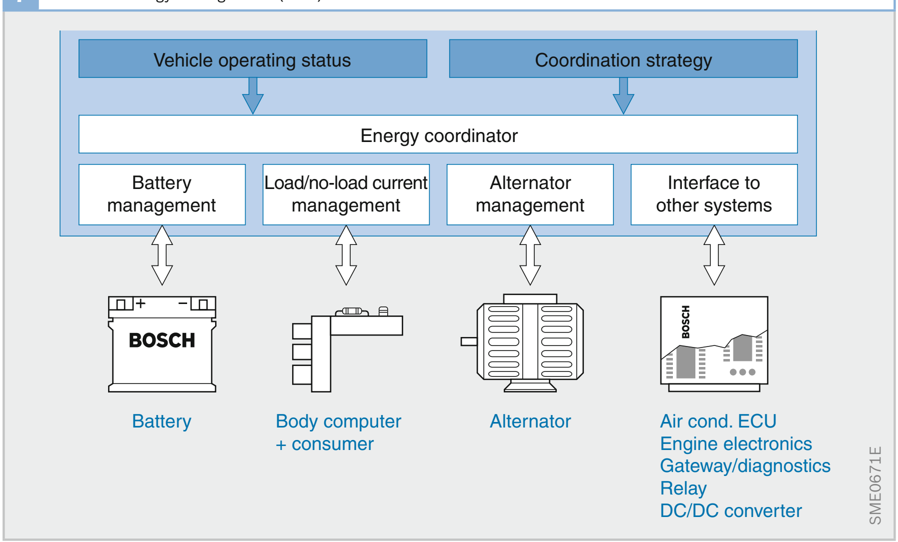
The battery management communicates to the EEM the variables relevant to the battery, e.g. the state of charge (SOC), the state of health (SOH) and the state of function (SOF) of the battery. SOF is a prediction of how the battery would react to a predefined load profile, e.g. whether a starting operation would be successful with the current battery status.

These values are calculated using complex, model-based algorithms from the measurement of battery current, voltage and temperature.

Using this battery data, the EEM is able to determine the optimal charge voltage and reduce the load on the electrical system (switch off consumers) in response to a degrading state of function and/or increase power generation (e.g. by increasing the idling speed).

If the battery's state of function falls below a specified threshold value despite the measures that have been implemented, the EEM can warn the driver that certain functions (e.g. engine start) will not be available with the current battery status.

7 Electrical energy management (EEM)



Battery status recognition
The control unit that makes battery status recognition (BSR) possible is the electronic battery sensor EBS (sometimes even EEM functions are implemented in this control unit). The sensor with integrated evaluation electronics records the fundamental battery variables of voltage, current and temperature. From these variables, it uses complex software algorithms to calculate the variables that describe the status of the car battery.

The electronic battery sensor comprises a chip, which contains the entire electronics, and a resistor element for current measurement. Together with the terminal clip, both of these form a single assembly unit that can be connected directly to the car battery and fits into the terminal recess of conventional car batteries.

The following tasks of the electrical energy management are made possible by the use of battery status recognition:

- Assurance of startability (SOF) through compliance with defined limit values of the battery state of function and an increase in vehicle availability
- Reductions in electrical load requirements and reductions in fuel consumption by means of alternator management with adaptation of alternator voltage
- Greater flexibility in the design of battery and alternator size by means of superordinate energy management (optimal economic efficiency)
- Extension of battery life (e.g. through prevention of exhaustive discharge)
- Battery change indication

For stop/start applications, the following functions can also be fulfilled:

- Prediction of startability after a defined stationary period (time, temperature, no-load current, power consumption during the stop phase)
- Assurance of a battery charge reserve in the stop phase (e.g. by switching off consumers)

8 Two-battery vehicle electrical system (layout)

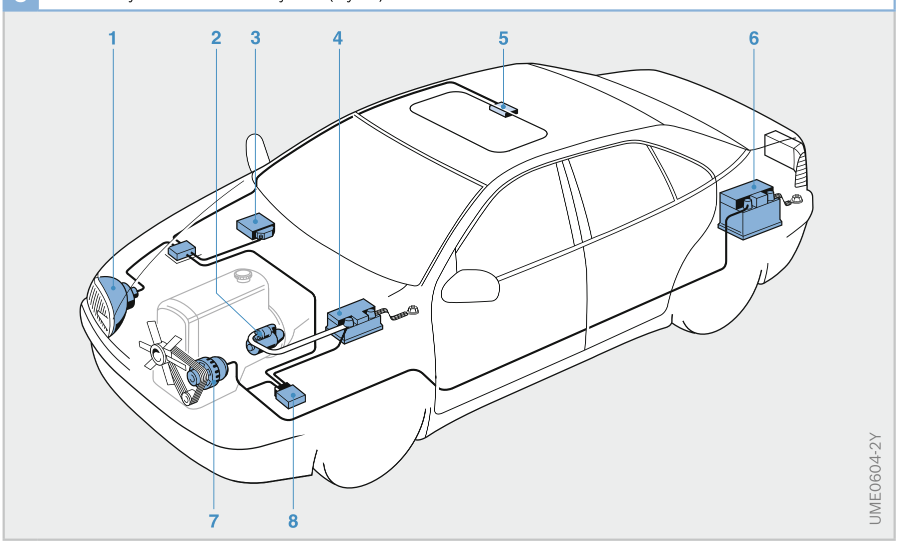


Fig. 8
1 Lighting system (vehicle electrical system)
2 Starter motor
3 Engine management (vehicle electrical system)
4 Starter battery
5 Other electrical-system consumers (e.g. power sunroof)
6 General-purpose battery
7 Alternator
8 Charging/isolating module

Two-battery vehicle electrical system

In the design of a vehicle battery, which supplies both the starter and the other consumers in the vehicle electrical system, a compromise has to be found between different requirements.

During the engine starting sequence, the battery is subjected to high current loads (300 to 500 A). The associated voltage drop has an adverse effect on certain electrical consumers (e.g. units with microcontroller) and should be as low as possible. On the other hand, only comparatively low currents flow during vehicle operation; for a reliable power supply, the capacity of the battery is the decisive factor. Neither properties – rated output nor capacity – can be optimized simultaneously.

In vehicle electrical systems with two batteries (starter battery and general-purpose battery), the "high power for starting" and "general-purpose electrical supply" functions are separated by the electrical system control unit to make it possible to avoid the voltage drop during the starting process, while ensuring reliable cold starts, even when the charge level of the general-purpose battery is low.

Starter battery

The starter battery must supply a high amount of current for only a limited period of time (during starting). It is therefore designed for a high power density (high power for low weight). Compact dimensions allow installation in the immediate vicinity of the starter motor with short connecting cables. The capacity is reduced.

General-purpose battery

This battery only supplies the vehicle electrical system (excluding the starter). It provides currents for supplying the consumers of the vehicle electrical system (e.g. approx. 20 A for the engine-management system) but has a high cyclic capability, i.e. it can supply and store large amounts of power. Dimensioning is based essentially on the capacity reserves required for activated consumers, the consumers that operate with the engine switched off (e.g. no-load-current consumers, parking lights, hazard warning flashers, immobilizer), and the minimum permissible charge level.

Vehicle electrical system control unit

The vehicle electrical system control unit (EN ECU) in a two-battery vehicle electrical system separates the starter battery and the starter from the rest of the vehicle electrical system provided this can be supplied with sufficient power by the general-purpose battery. It therefore prevents the voltage drop that occurs during starting, affecting the performance of the vehicle electrical system. When the vehicle is parked, it prevents the starter battery from becoming discharged by activated

9 Two-battery vehicle electrical system (schematic)

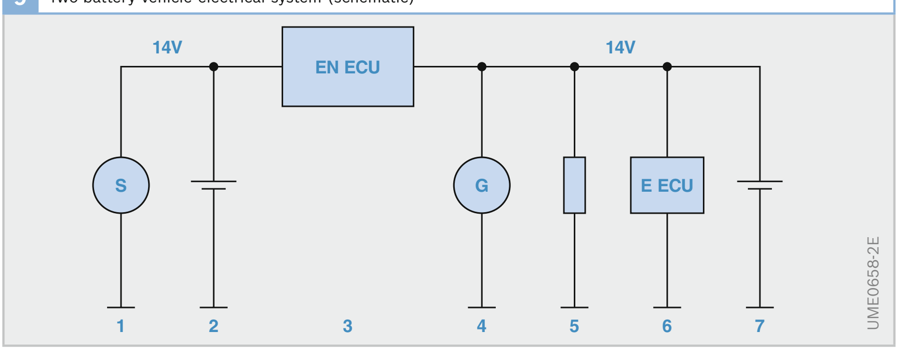


Fig. 9
1 Starter
2 Starter battery
3 Vehicle electrical system control unit
4 Alternator
5 Consumer
6 Engine control unit
7 General-purpose battery

consumers with the engine switched off and standstill-draw consumers.

By separation of the starting system from the remainder of the vehicle electrical system, there are theoretically no limits for the voltage level within the starting system. Consequently, the charge voltage can be optimally adapted to the starter battery by a DC/DC converter to minimize the charging time.

If there is no charge in the general-purpose battery, the control unit is capable of provisionally connecting both vehicle electrical systems. This means that the vehicle electrical system can be sustained using the fully-charged starter battery. In another possible configuration, the control unit for the starting operation would connect only the start-related consumers to whichever battery was fully charged.

Vehicle electrical systems for commercial vehicles

Battery changeover 12/24 V

Various heavy commercial vehicles have a combined 12/24 V system, i.e. the supply voltage can be switched between 12 and 24 V. In such cases, the alternator for voltage generation and the electrical components (starter excepted) are designed for nominal 12 V operation. The starter has a nominal voltage of 24 V. It is therefore possible to achieve the power output required to start larger diesel engines, for example.

The system consists of two 12 V batteries that are connected in parallel during vehicle operation and with the engine switched off. The voltage is no different with the parallel connection; the vehicle electrical system is supplied with 12 V. The total

10 Circuit of a starting system with battery changeover relay

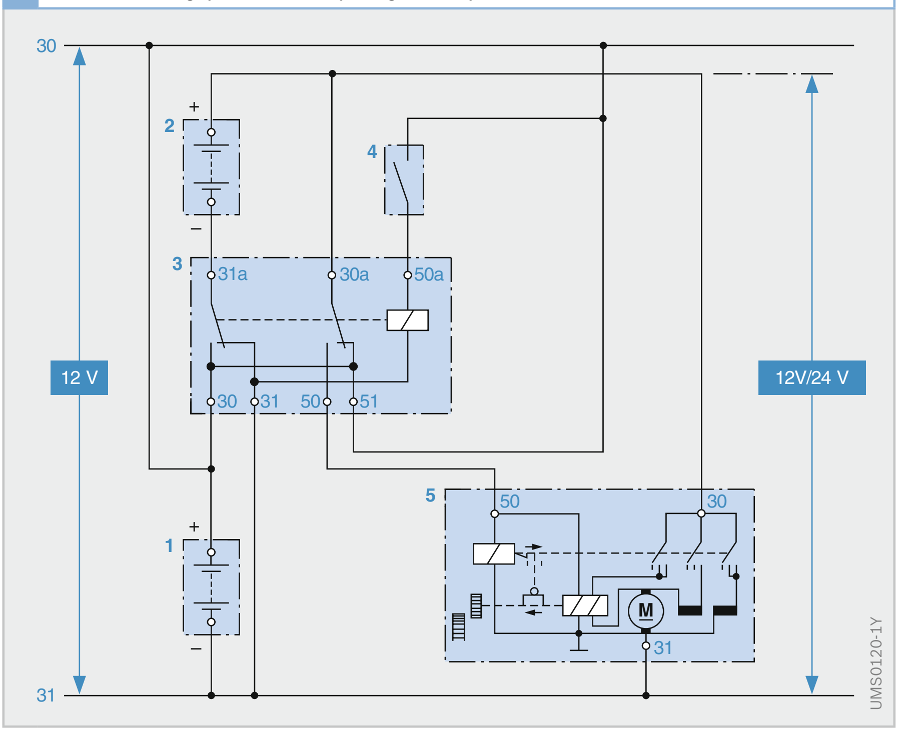


Fig. 10
1 12 V battery I
2 12 V battery II
3 Battery changeover relay
4 Ignition switch
5 24 V starting motor

capacity of the two batteries is the sum of the individual capacities.

When the ignition switch is turned, a battery changeover relay automatically switches the two batteries in series so that 24 V is applied across the starting-motor terminals during the cranking process. All other consumers are still supplied with 12 V.

As soon as the engine has started, that is when the ignition switch has been released and the starting motor switched off, the battery changeover relay automatically connects the batteries in parallel again. With the engine turning, the 12 V alternator recharges both batteries.

The capacities of two batteries connected in parallel should be equal to achieve uniform current distribution during the charging and discharging process. From the wiring viewpoint, connections should also be as symmetrical as possible, with identical lengths of connection cable and conductor cross-sections.

Components in the vehicle electrical system

At the present time, the components described can also be used in passenger-car electrical systems. Here, however, they are not common and tend to be fitted as optional equipment.

Battery master switch

Generally, the vehicle's electrical installation is wired such that when the key is pulled from the ignition switch the electrical lines leading from the switch to, for example, the ignition system, the control units (Motronic, ABS), the wipers, etc. are no longer supplied with current.

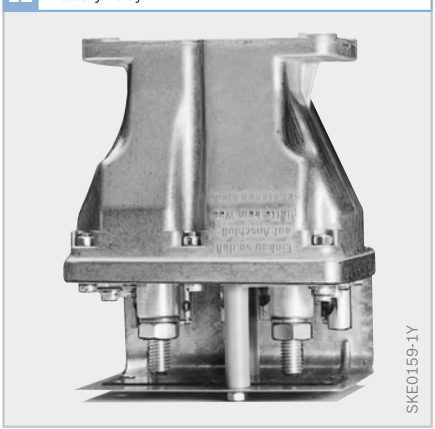
On the other hand, the lines leading to the ignition switch, to the starter, and to the light switch remain "live". In other words there is still voltage on these lines, and if they have frayed or worn-through points these can lead to low resistances which can cause leakage currents or short circuits which result in a discharged battery. The consequences are a discharged battery or the possibility of a fire. A battery master switch makes it possible to completely isolate the battery from the vehicle electrical system to eliminate the risk of these dangers.

The single-pole battery master switch is installed in the battery's ground cable (negative terminal) as near to the battery as possible. It should be within convenient reach of the driver.

On installations equipped with three-phase current alternators, due to the danger of voltage peaks (with the attendant destruction of electronic components), it is forbidden to operate the vehicle without the bat-

11 Battery master switch

12 Battery relay



tery connected. On such installations therefore, the battery master switch may only be actuated with the engine at standstill.

Battery relay

Legislation stipulates that in buses, road tankers etc. a battery relay must be installed as the master switch to separate the vehicle electrical system from the battery. Not only short circuits are avoided (during repairs for instance), but also the decomposition effects due to leakage currents on current-carrying components.

For this type of installation with three-phase current alternator, in order to prevent excessive voltage peaks it is necessary to fit a 2-pole electromagnetic battery main switch. This prevents the alternator being separated from the battery when the engine is running.

Battery-cutoff relay

The battery-cutoff relay (NO contact) separates the starter battery from a second battery used for ancillary equipment. It protects the starter battery against discharge when the three-phase current alternator is not delivering charge current. This relay is provided with a diode for reverse-polarity protection, and a decay diode to suppress the inductive voltage peaks caused by switching.

Battery charging relay

The battery charging relay is needed when an additional 12V battery is to be charged in a 24 V vehicle system voltage. It is provided with resistors across which at 10 A charging current a voltage drop takes place which reduces the charge voltage to 12 V. This of course necessitates the 24 V alternator being able to generate the additional 10 A.

13 Battery-cutoff relay symbol

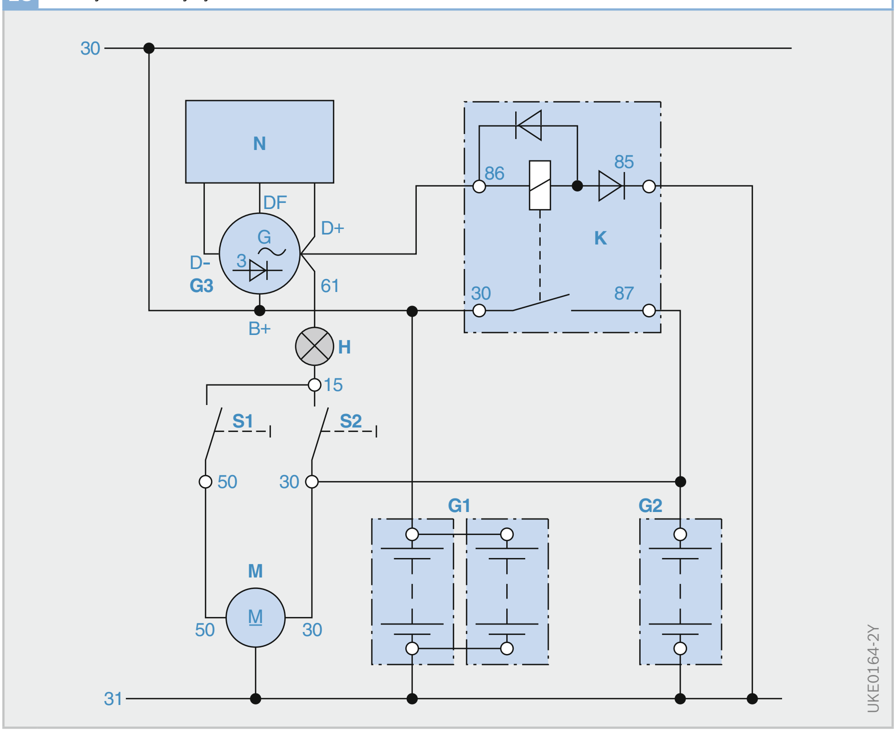


Fig. 13
- G1 Battery for ancillary equipment
- G2 Starter battery
- G3 Three-phase current alternator
- H Charge-indicator lamp
- K Battery-cutoff relay
- M Starter
- N Alternator regulator
- S1 Ignition switch
- S2 Driving switch

Wiring harnesses

Requirements

The purpose of the wiring harness is to distribute power and signals within a motor vehicle. A wiring harness in the present day, mid-class passenger car with average equipment has approximately 750 different lines, their length totaling around 1,500 meters. In recent years the number of contact points has practically doubled due to the continuous rise in functions in the motor vehicle. A distinction is made between the engine compartment and body wiring harness. The latter is subject to less demanding temperature, vibration, media and tightness requirements.

Wiring harnesses have considerable influence on the costs and quality of a vehicle. The following points must be taken into consideration in wiring harness development:

- Leak-tightness
- EMC compatibility
- Temperatures
- Damage protection for the lines
- Line routing
- Ventilation of the wiring harness

It is therefore necessary to involve wiring harness experts as soon as in the system definition stage. Figure 1 shows a wiring harness that was developed as a special intake-module wiring harness. Thanks to the optimization of routing and security in conjunction with engine and wiring harness development, it was possible to achieve an advancement of quality as well as to yield cost and weight advantages.

Dimensions and material selection

The most important tasks for the wiring harness developer are:

- Dimensioning the line cross-sections
- Material selection
- Selection of suitable plug-in connections
- Routing of lines under consideration of ambient temperature, engine vibrations, acceleration and EMC
- Consideration of the environment in which the wiring harness is routed (topology, assembly stages in vehicle manufacture and equipment on the assembly line)

1 Wiring harness (example)

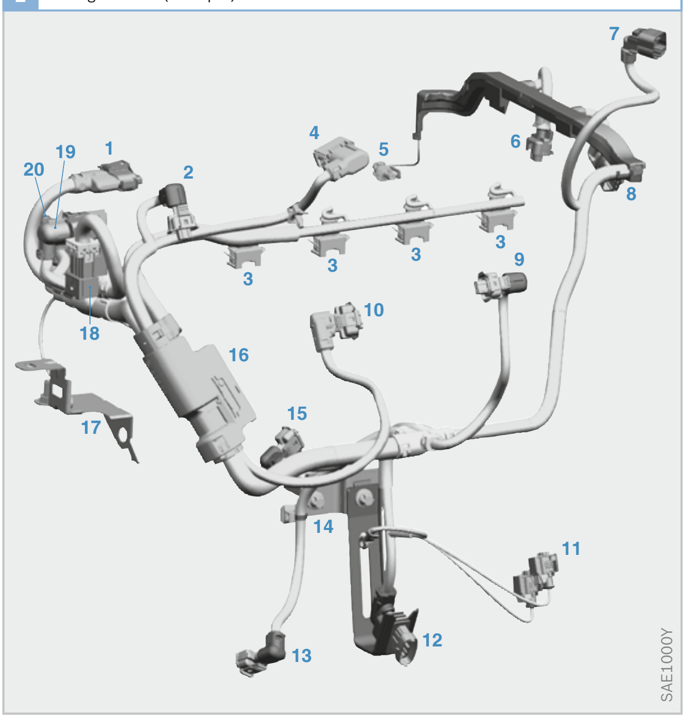


Fig. 1
1 Ignition coil module
2 Channel deactivation
3 Fuel injectors
4 Throttle device DV-E
5 Oil-pressure switch
6 Engine-temperature sensor
7 Intake-air temperature sensor
8 Camshaft sensor
9 Canister-purge valve
10 Intake-manifold pressure sensor
11 Charge-current indicator lamp
12 Downstream Lambda oxygen sensor
13 Speed sensor
14 Terminal 50, starter switch
15 Knock sensor
16 Engine control unit
17 Engine ground
18 Disconnecter plug for engine and transmission wiring harness
19 Upstream Lambda oxygen sensor
20 EGR valve

Line cross-sections

Line cross-sections are defined based on permissible voltage drops. The lower cross-section limit is determined by the line strength. Convention has it that no lines with a cross-section of less than 0.5 mm^2 are used. With additional measures (e.g. supports, protective tubes, tension relief), even a cross-section of 0.35 mm^2 may be permissible.

Materials

Copper is usually used as the conductive material. The insulation materials for the lines are defined by the temperature to which they are exposed. It is necessary to use materials that are suitable for the high temperatures of continuous operation. Here, the ambient temperature must be taken into consideration as much as the heating caused by the flow of current. The materials used are thermoplastics (e.g. PE, PA, PVC), fluoropolymers (e.g. ETFE, FEP) and elastomers (e.g. CSM, SIR).

If the lines are not routed past particularly hot parts (e.g. exhaust pipe) in the engine topology, one of the criteria for the selection of the insulation material and the cable cross-section could be the derating curve of the contact with its associated line. The derating curve represents the relationship between current, the temperature increase that it causes, and the ambient temperature of the plug-in connection. Normally, the heat generated in the contacts can only be carried away along the lines themselves. It should also be noted that the change in temperatures results in a change in the modulus of elasticity of the contact material (metal relaxation). It is possible to influence the relationships described by means of larger line cross-sections and the use of suitable contact types and more noble surfaces (e.g. gold, silver) and thus higher limit temperatures. For highly fluctuating current intensities, it is often useful to measure the contact temperature.

Plug-in connections and contacts

The type of plug-in connections and contacts used depends on various factors:

- Current intensity
- Ambient temperatures
- Vibration load
- Resistance to media and
- Installation space

Line routing and EMC measures

Lines should be routed in such a way as to prevent damage and line breaks. This is achieved by means of fasteners and supports. Vibration loads on contacts and plug-in connections are reduced by fastening the wiring harness as close to the plug as possible and at the same level as the vibration where possible. The line routing must be determined in close cooperation with the engine and vehicle developers.

Where EMC problems arise, it is recommended to route sensitive lines and lines with steep current flanks separately. Shielded lines are not straightforward to produce and are therefore expensive. They also need to be grounded. The twisting of lines is a more cost-favorable and effective measure.

Line protection

Lines need to be protected against chafing and against making contact with sharp edges and hot surfaces. Adhesive tapes are used for this purpose. The level of protection is determined by the interval and winding density. Corrugated tubing (material savings from corrugation) with the necessary connecting pieces are often used as line protection. However, tape fixing is still an essential means of preventing movement of individual lines inside the corrugated tube. Optimal protection is offered by cable ducts.

Plug-in connections

Design and requirements

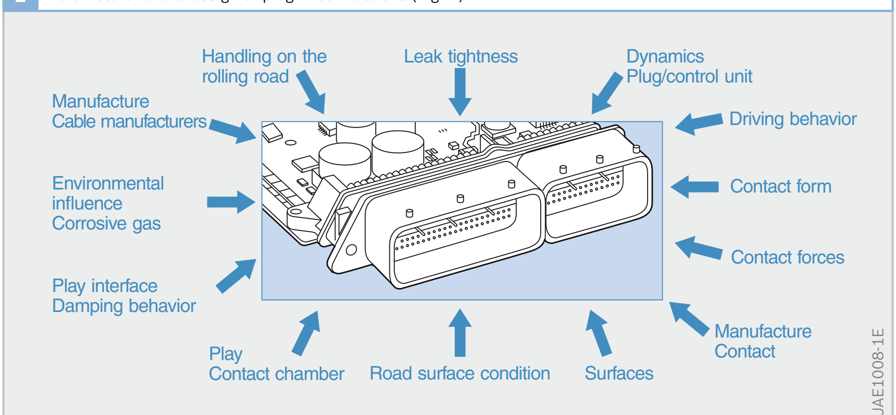
The high integration density of electronics in the motor vehicle places high demands on a car's plug-in connections. Not only do they carry high currents (e.g. activation of ignition coils), they also carry analog signal currents with low voltage and current intensity (e.g. signal voltage of the engine temperature sensor). Throughout the service life of the vehicle, the plug-in connections must ensure the reliable transmission of signals between control units and to the sensors whilst maintaining tolerances.

The increasing demands of emission-control legislation and active vehicle safety are forcing the ever more precise transmission of signals through the contacts of the plug-in connections. A large number of parameters must be taken into consideration in the design, arrangement and testing of the plug-in connections (Fig. 1).

The most common cause of failure of a plug-in connection is wearing of the contact caused by vibrations or temperature change. The wear promotes oxidation. This results in an increase in ohmic resistance – the contact may, for example, be subjected to thermal overload. The contact part may be heated beyond the melting point of the copper alloy. In the case of highly resistant signal contacts, the vehicle controller often detects a plausibility error by comparison with other signals; the controller then enters fault mode. These problem areas in the plug-in connection are diagnosed by the on-board diagnosis (OBD) required by emission-control legislation. However, it is difficult to diagnose the error in the service workshops because this defect is displayed as being a component failure. It is only possible to diagnose the faulty contact indirectly.

For the assembly of the plug-in connection, there are various functional elements on the plug housing intended to ensure that the cables with their crimped contacts can be joined to the plug-in connection reliably and defect-free. Modern plug-in connections have a joining force of < 100 N so that the assembly operative is able to reliably join the connector to the component or control-unit interface. The risk of plug-in connections being connected to the interface incorrectly increases with higher connecting forces. The plug would come loose during vehicle operation.

1 Parameters for the design of plug-in connections (Fig. A)



1 Areas of application for plug-in connections			
	No. of poles	Special features	Applications
Low-pin-count	1 to 10	No joining force support	Sensors and actuators (many different requirements)
High-pin-count	10 to 150	Joining force support by slide, lever, modules	Control units (several, similar requirements)
Special connectors	any	e.g. integrated electronics	Special applications (individual, matched requirements)

Table 1

Design and types

Plug-in connections have different areas of application (Table 1). These are characterized by the number of poles and ambient conditions. There are different classes of plug-in connection: hard engine attachment, soft engine attachment, and body attachment. Another difference is the temperature class of the installation location.

High-pin-count plug-in connections

High-pin-count plug-in connections are used for all control units in the vehicle. They differ in their number of poles and the geometry of the pins. Figure 2 shows a typical design of a high-pin-count plug-in connection. The complete plug-in connection is sealed at the connector strip of the respective control unit by means of a continuous radial seal in the plug housing. This, together with three sealing lips, ensures a reliable seal against the control unit sealing collar.

The contacts are protected against the ingress of humidity along the cable by a seal plate, through which the contacts are inserted and the line crimped to them. A silica-gel mat or silica mat is used for this purpose. Larger contacts and lines may also be sealed using a single-core seal (see "Low-pin-count plug-in connections").

When the plug is assembled, the contact with the line attached is inserted through the seal plate that is already in the plug. The contact slides home into its position in the contact carrier. The contact latches on its own by a locking spring that engages in an undercut in the plastic housing of the plug.

2 High-pin-count plug-in connection (Fig. B)

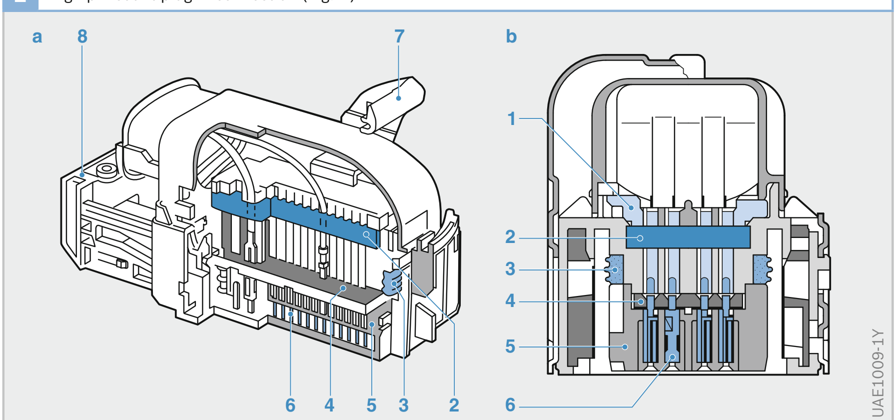


Fig. 2
a 3D view
b Sectional view

1 Pressure plate
2 Seal plate
3 Radial seal
4 Slide pin (secondary lock)
5 Contact carrier
6 Switch contact
7 Lever
8 Slide mechanism

Once all contacts are in their final position, a slide pin is inserted to provide a second contact safeguard, or secondary lock. This is an additional security measure and increases the retaining force of the contact in the plug-in connection. In addition, the sliding movement is a means of checking that the contacts are in the correct position. The operating force of the plug-in connection is reduced by a lever and a slider mechanism.

Low-pin-count plug-in connections

Low-pin-count plug-in connections are used for actuators (e.g. fuel injectors) and sensors. Their design is similar in principle to that of a high-pin-count plug-in connection (Fig. 3). The operating force of the plug-in connection is not usually supported.

The connection between a low-pin-count plug contact system and the interface is sealed with a radial seal. Inside the plastic housing, however, the lines are sealed with single-core seals secured to the contact.

Contact systems

Two-part contact systems are used in the motor vehicle. The inner part (Fig. 4) – the live part – is pressed from a high-quality copper alloy. It is protected by a steel overspring, which at the same time increases the contact forces of the contact by means of an inwardly acting spring element. A catch arm pressed out from the steel overspring engages the contact in the plastic housing part. Contacts are coated with tin, silver or gold, depending on requirements. To improve the wear characteristics of the contact point, not only are different contact coatings used but also different structural shapes. Different decoupling mechanisms are integrated into the contact part to decouple cable vibrations from the contact point (e.g. meandering routing of the supply lead).

The cables are crimped onto the contact. The crimp geometry on the contact must be adapted to the cable concerned. Pliers or fully automatic, process-monitored crimping presses with contact-specific tools are available for the crimping process.

3 Low-pin-count plug-in connection (Fig. C)

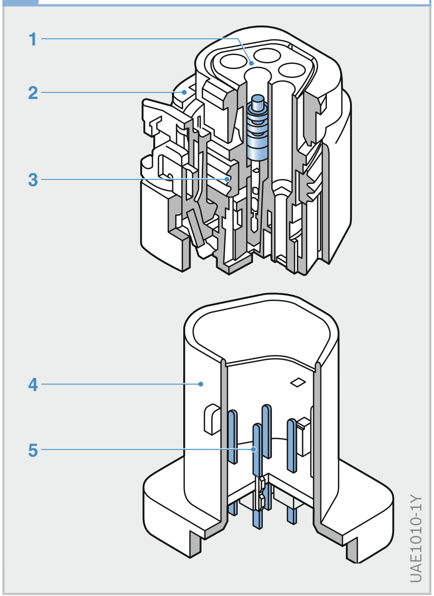


Fig. 3
1 Contact carrier
2 Housing
3 Radial seal
4 Interface
5 Flat blade

4 Contact (Fig. D)

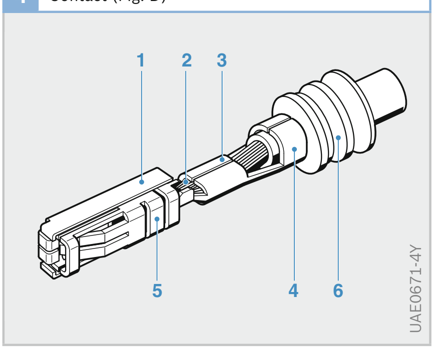
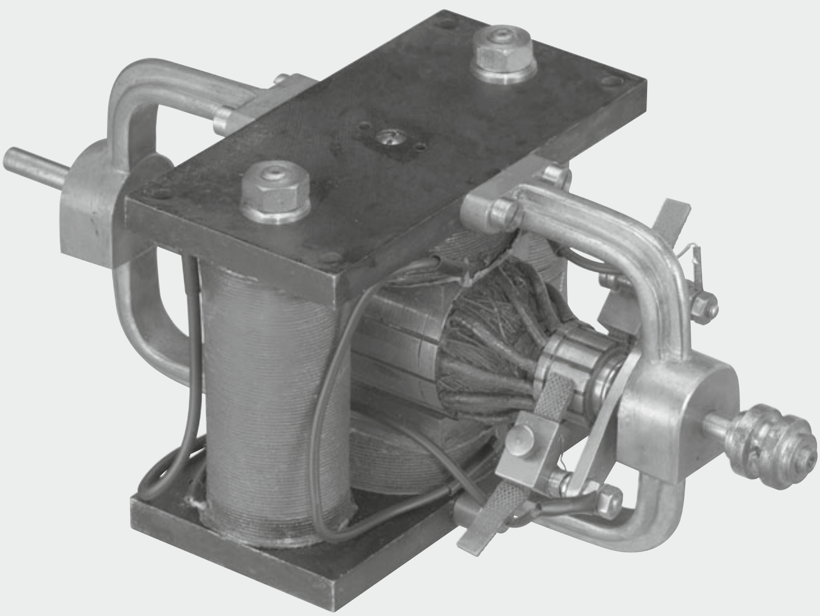


Fig. 4
1 Steel overspring
2 Single conductor (single core)
3 Conductor crimp
4 Insulation crimp
5 Wave-shaped interior design
6 Single-core seal

History of the alternator

At the turn of the century, the introduction of electrical lighting to motor vehicles to take the place of the previously used horse-and-carriage lighting meant that a suitable source of electrical power had to be available in the vehicle. The battery alone was completely unsuitable since, when discharged, it had to be removed from the vehicle for re-charging. In around 1902, the model for a dynamo (the basis for today's alternator) was created at Robert Bosch. It mainly comprised permanent magnets as stators, an armature with commutator and a contact breaker for ignition (see Fig.). The only difficulty here, though, was the fact that the dynamo's voltage was dependent on the engine's speed which varied considerably.

Endeavors, therefore, concentrated on the development of a DC dynamo with voltage regulation. Finally, electromagnetic control of the field resistor as a function of the machine's output voltage proved to be the answer. Around 1909, using the knowledge available at that time, it thus became possible to build a complete "Lighting and Starting System for Motor Vehicles".
This was introduced to the market in 1913 and comprised a dynamo (splashwater-protected, encapsulated 12-V DC dynamo with shunt regulation and a rated output of 100 W), a battery, a voltage-regulator and switch box, a freewheeling starter with pedal-operated switch, and a variety of different lighting components.



UME0664Y

Overview of electrical and electronic systems in the vehicle

The amount of electronics in the vehicle has risen dramatically in recent years and is set to increase yet further in the future. Technical developments in semiconductor technology support ever more complex functions with the increasing integration density. The functionality of electronic systems in motor vehicles has now surpassed even the capabilities of the Apollo 11 space module that orbited the Moon in 1969.

Overview

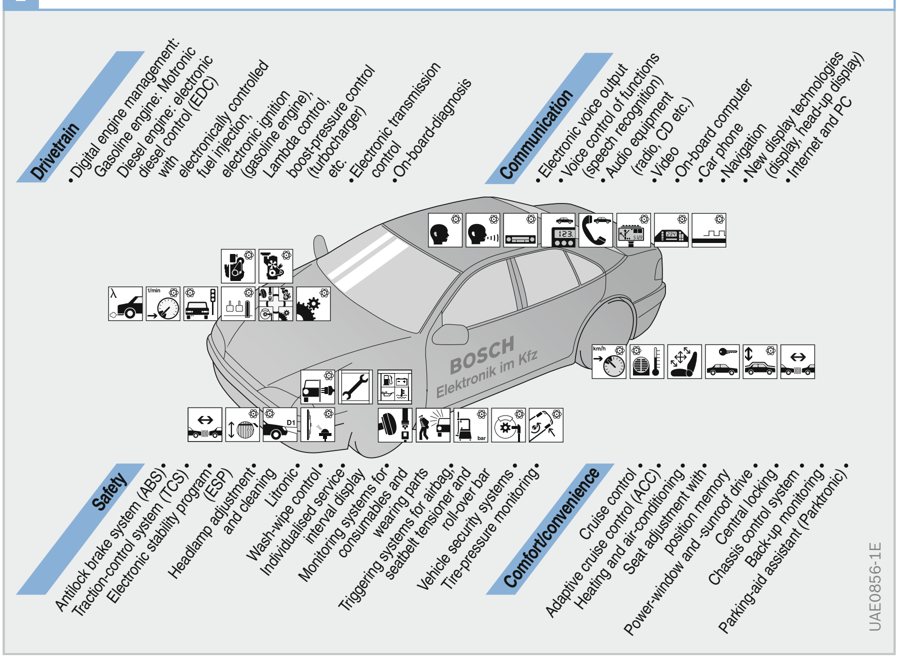
Development of electronic systems

Not least in contributing to the success of the vehicle has been the continuous string of innovations which have found their way into vehicles. Even as far back as the 1970s, the aim was to make use of new technologies to help in the development of safe, clean and economical cars. The pursuit of economic efficiency and cleanliness was closely linked to other customer benefits such as driving pleasure. This was characterized by the European diesel boom, upon which Bosch had such a considerable influence. At the same time, the development of the gasoline engine with gasoline direct injection, which would reduce fuel consumption by comparison with intake-manifold injection, experienced further advancements.

An improvement in driving safety was achieved with electronic brake-control systems. In 1978, the antilock brake system (ABS) was introduced and underwent continual development to such an extent that it is now fitted as standard on every vehicle in Europe. It was along this same line of development that the electronic stability program (ESP), in which ABS is integrated, would debut in 1995.

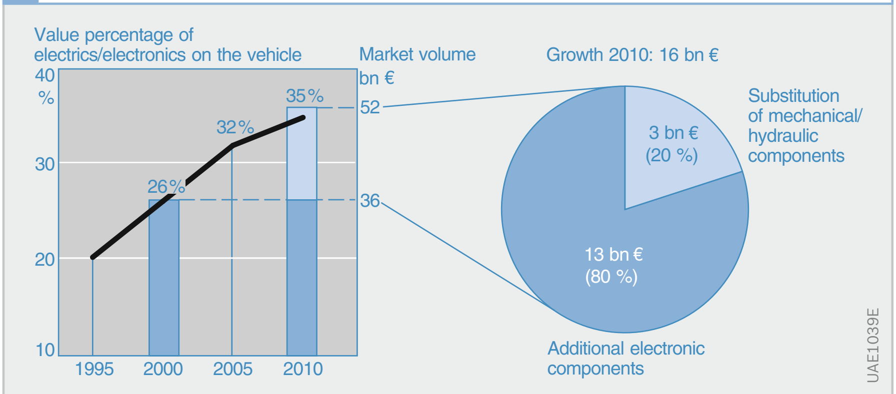
The latest developments also take comfort into account. These include the hill hold control (HHC) function, for example, which makes it easier to pull away on uphill gradients. This function is integrated in ESP.

1 Electronics in the motor vehicle



K. Reif (Ed.), *Fundamentals of Automotive and Engine Technology*,
DOI 10.1007/978-3-658-03972-1_13, © Springer Fachmedien Wiesbaden 2014

2 Market volumes of electrics/electronics in Europe (estimates)



Many kinds of new functions appear in conjunction with driver-assistance systems. Their scope extends far beyond today's standard features such as Parkpilot or electronic navigation systems. The aim is to produce the "sensitive vehicle" that uses sensors and electronics to detect and interpret its surroundings. Tapping into ultrasound, radar and video sensor technologies has led to solutions that play an important role in assisting the driver, e.g. through improved night vision or distance control.

Value creation structure for the future

The latest studies show that the production costs of an average car will increase only slightly by 2010 despite further innovations. No significant value growth for existing systems is expected in the mechanics/hydraulics domain despite the expected volume growth. One reason here being the electrification of functions that have conventionally been realized mechanically or hydraulically. Brake control systems are an impressive example of this change. While the conventional brake system was characterized more or less completely by mechanical components, the introduction of the ABS brake-control system was accompanied by a greater proportion of electronic components in the form of sensor technology and an electronic control unit. With the more recent developments of ESP, the additional functions, such as HHC, are almost exclusively realized by electronics.

Even though significant economies of scale are seen with the established solutions, the value of the electrics and electronics will increase overall (Fig. 1). By 2010, this will amount to a good third of the production costs of an average vehicle. This assumption is based not least on the fact that the majority of future functions will also be regulated by electrics and electronics.

The increase in electrics and electronics is associated with a growth in software. Even today, software development costs are no longer negligible by comparison with hardware costs. Software authoring is faced with two challenges arising from the resulting increase in complexity of a vehicle's overall system: coping with the volume and a clearly structured architecture. The Autosar initiative (Automotive Open Systems Architecture), in which various motor vehicle manufacturers and suppliers participate, is working towards a standardization of electronics architecture with the aim of reducing complexity through increased reusability and interchangeability of software modules.

Task of an electronic system

Open-loop and closed-loop control

The nerve center of an electronic system is the control unit. Figure 3 shows the system blocks of a Motronic engine-management system. All the open-loop and closed-loop algorithms of the electronic system run inside the control unit. The heart of the control unit is a microcontroller with the program memory (flash EPROM) in which is stored the program code for all functions that the control unit is designed to execute.

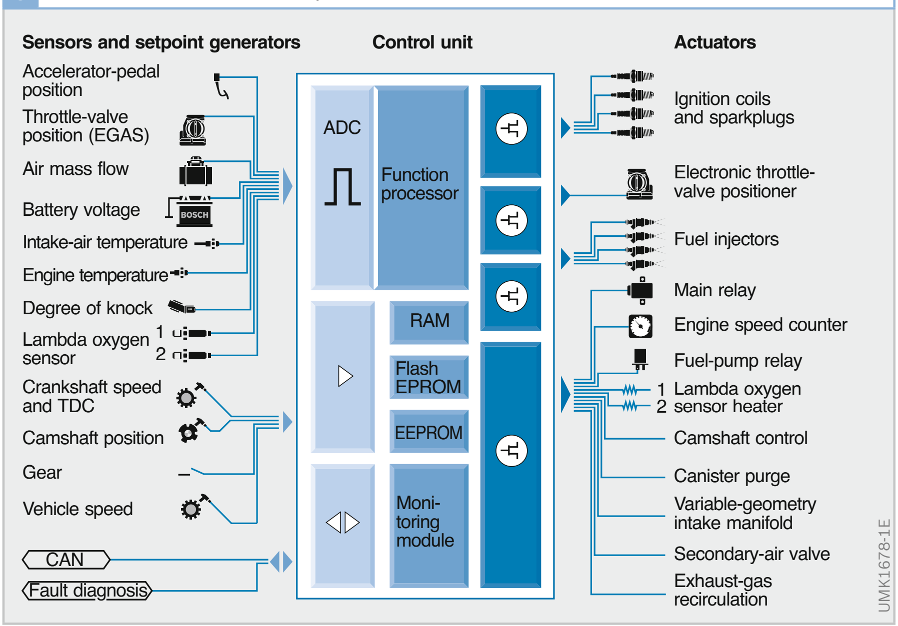
The input variables for the sequence control are derived from the signals from sensors and setpoint generators. They influence the calculations in the algorithms, and thus the triggering signals for the actuators. These convert into mechanical variables the electrical signals that are output by the microcontroller and amplified in the output stage modules. This could be mechanical energy generated by a servomotor (power-window unit), for example, or thermal energy generated by a sheathed-element glow plug.

Communication

Many systems have a mutual influence on each other. For example, it may sometimes be necessary to not only have the electronic stability program carry out a braking intervention in the event wheel spin but also to request that the engine-management system reduce torque and thus counteract wheel spin. Similarly, the control unit for the automatic transmission outputs a request to the engine-management system to reduce torque during a gearshift and thereby promote a soft gear change. To this end, the systems are networked with each other, i.e. they are able to communicate with each other on data buses (e.g. CAN, LIN).

In a premium-class vehicle, there may be up to 80 control units performing their duties. The examples below are intended to give you an insight into the operating principle of these systems.

3 Function modules of an electronic system



Control of gasoline engines

"Motronic" is the name of an engine-management system that facilitates open- and closed-loop control of gasoline engines within a single control unit.

There are Motronic variants for engines with intake-manifold injection (ME Motronic) and for gasoline direct injection (DI Motronic). Another variant is the Bifuel Motronic, which also controls the engine for operation with natural gas.

System description

Functions

The primary task of the Motronic engine-management system is:

- To adjust the torque desired and input by the driver depressing the accelerator pedal
- To operate the engine in such a way as to comply with the requirements of ever more stringent emission-control legislation
- To ensure the lowest possible fuel consumption but at the same time
- To guarantee high levels of driving comfort and driving pleasure

Components

Motronic comprises all the components which control and regulate the gasoline engine (Fig. 1, next page). The torque requested by the driver is adjusted by means of actuators or converters. The main individual components are:

- The electrically actuated throttle valve (air system): this regulates the air-mass flow to the cylinders and thus the cylinder charge
- The fuel injectors (fuel system): these meter the correct amount of fuel for the cylinder charge
- The ignition coils and spark plugs (ignition system): these provide for correctly timed ignition of the air-fuel mixture present in the cylinder

Depending on the vehicle, different measures may be required to fulfill the requirements demanded of the engine-management system (e.g. in respect of emission characteristics, power output and fuel consumption). Examples of system components able to be controlled by Motronic are:

- Variable camshaft control: it is possible to use the variability of valve timing and valve lifts to influence the ratio of fresh gas to residual exhaust gas and the mixture formation
- External exhaust-gas recirculation: adjustment of the residual gas content by means of a precise and deliberate return of exhaust gas from the exhaust train (adjustment by the exhaust-gas recirculation valve)
- Exhaust-gas turbocharging: regulated supercharging of the combustion air (i.e. increase in the fresh air mass in the combustion chamber) to increase torque
- Evaporative emission control system: for the return of fuel vapors that escape from the fuel tank and are collected in an activated charcoal canister

Operating variable acquisition

Motronic uses sensors to record the operating variables required for the open and closed-loop control of the engine (e.g. engine speed, engine temperature, battery voltage, intake air mass, intake-manifold pressure, Lambda value of the exhaust gas).

Setpoint generators (e.g. switches) record the adjustments made by the driver (e.g. position of the ignition key, cruise control).

Operating variable processing

From the input signals, the engine ECU detects the current operating status of the engine and uses this information in conjunction with requests from auxiliary systems and from the driver (accelerator-pedal sensor and operating switches) to calculate the command signals for the actuators.

K. Reif (Ed.), *Fundamentals of Automotive and Engine Technology*,
DOI 10.1007/978-3-658-03972-1_14, © Springer Fachmedien Wiesbaden 2014

1 Components used for open-loop electronic control of a DI-Motronic system (example of a naturally aspirated engine, λ = 1)

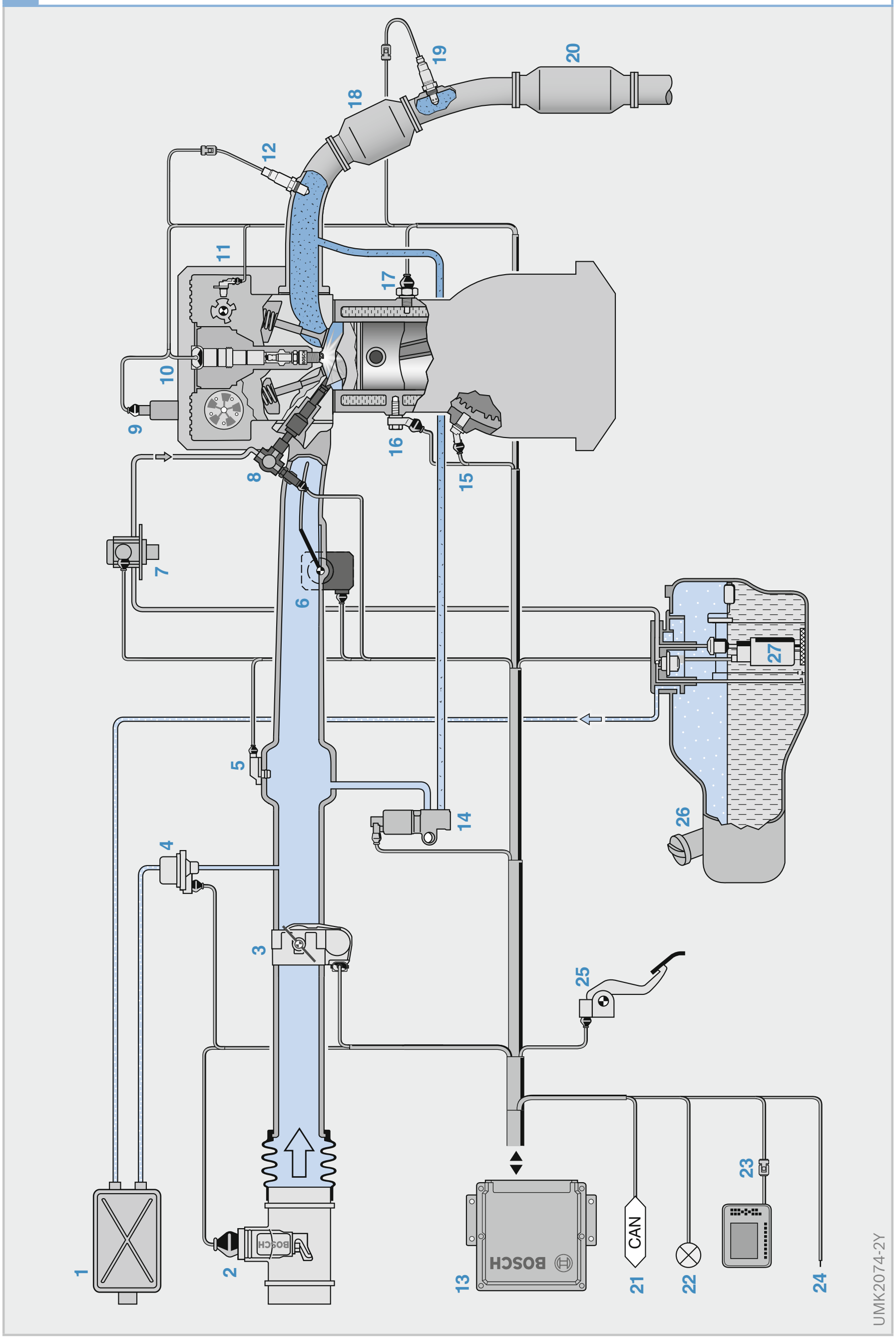


Fig. 1

1 Activated charcoal canister
2 Hot-film air-mass meter
3 Throttle device (ETC)
4 Canister-purge valve
5 Intake-manifold pressure sensor
6 Swirl control valve
7 High-pressure pump
8 Rail with high-pressure fuel injector
9 Camshaft adjuster
10 Ignition coil with spark plug
11 Camshaft phase sensor
12 Lambda oxygen sensor (LSU)
13 Motronic ECU
14 EGR valve
15 Speed sensor
16 Knock sensor
17 Engine-temperature sensor
18 Primary catalytic converter
19 Lambda oxygen sensor
20 Primary catalytic converter
21 CAN interface
22 Diagnosis lamp
23 Diagnosis interface
24 Interface with immobilizer control unit
25 Accelerator-pedal module
26 Fuel tank
27 Fuel delivery module with electric fuel-supply pump

Air system

A specific air-fuel mixture is required to achieve the desired torque. For this purpose, the throttle valve (Fig. 1, Item 3) regulates the air necessary for the mixture formation by adjusting the metering orifice in the intake port for the fresh air taken in by the cylinders. This is effected by a DC motor (Fig. 2) integrated in the throttle device that is controlled by the Motronic control unit. The position of the throttle valve is fed back to the control unit by a position sensor to make position control possible. This sensor may be in the form of a potentiometer, for example. Since the throttle device is a component relevant to safety, the sensor is designed with redundancy.

The intake air mass (air charge) is recorded by sensors (e.g. hot-film air-mass meter, intake-manifold pressure sensor).

Fuel system

The control unit (Fig. 1, Item 13) calculates the fuel volume required from the intake air mass and the current operating status of the engine (e.g. intake-manifold pressure, engine speed), and also the time at which fuel injection should take place.

In gasoline injection systems with intake manifold injection, the fuel is introduced into the intake duct upstream of the intake valves. To this end, the electric fuel-supply pump (27) delivers fuel (primary pressure up to approximately 450 kPa) to the fuel injectors. Each cylinder is assigned a fuel injector that injects the fuel at intermittent intervals. The air-fuel mixture in the intake passage flows into the cylinder during the induction stroke. Corrections are made to the injected fuel quantity, e.g. by the Lambda control (Lambda oxygen sensor, 12) and the canister purge (evaporative-emissions control system, 1, 4).

With gasoline direct injection, fresh air flows into the cylinder. The fuel is injected directly into the combustion chamber by high-pressure fuel injectors (8) where it forms an air-fuel mixture with the intake air. This requires a higher fuel pressure, which is generated by additional high-pressure pump (7). The pressure can be variably adjusted (up to 20 MPa) in line with the operating point by an integrated fuel-supply control valve.

2 Throttle device with potentiometric position feedback

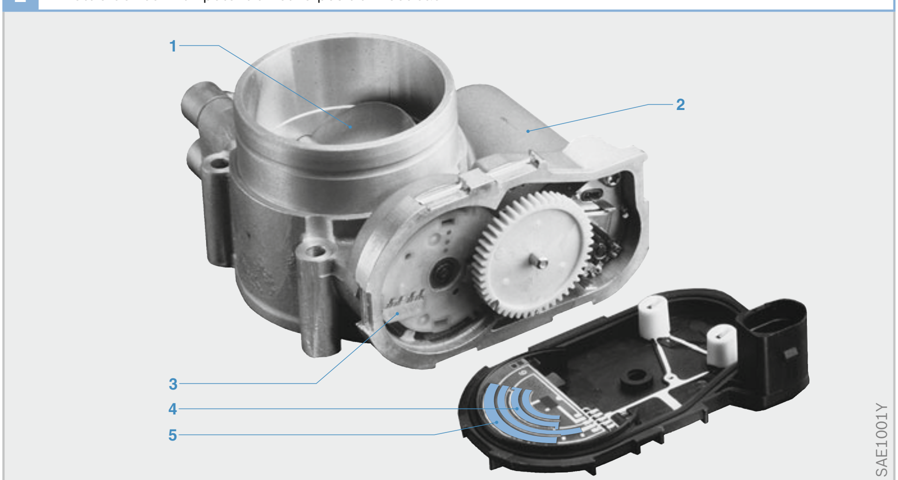


Fig. 2
1 Throttle valve
2 DC motor
3 Wiper
4 Resistance track 1
5 Resistance track 2

Fuel injector for intake-manifold injection

Function

The electromagnetic (solenoid-controlled) fuel injectors spray the fuel into the intake manifold at primary pressure. They allow fuel to be metered in the precise quantity required by the engine. They are actuated by driver stages which are integrated in the engine ECU with the signal calculated by the engine-management system.

3 EV14 electromagnetic fuel injector

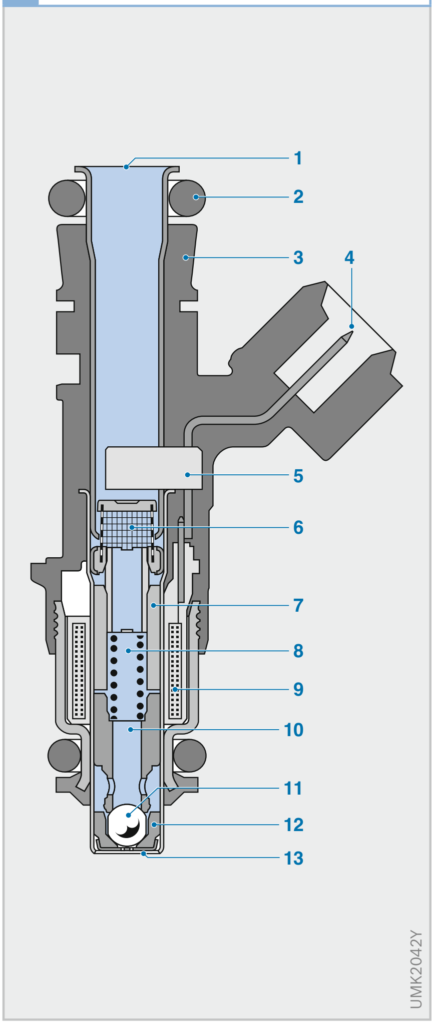


Fig. 3
1 Hydraulic port
2 O-ring
3 Valve housing
4 Electrical connection
5 Plastic clip with injected pins
6 Filter strainer
7 Internal pole
8 Valve spring
9 Solenoid coil
10 Valve needle with armature
11 Valve ball
12 Valve seat
13 Injection-orifice plate

Design and operating principle

Essentially, electromagnetic fuel injectors (Fig. 3) are comprised of the following components:

- Valve housing (3) with electrical connection (4) and hydraulic port (1)
- Solenoid coil (9)
- Moving valve needle (10) with solenoid armature and valve ball (11)
- Valve seat (12) with injection-orifice plate (13) and
- Valve spring (8)

In order to ensure trouble-free operation, stainless steel is used for the parts of the fuel injector which come into contact with fuel. The fuel injector is protected against dirt by a filter strainer (6) at the fuel inlet.

Connections

On the fuel injectors presently in use, fuel supply to the fuel injector is in the axial direction, i.e. from top to bottom ("top feed"). The fuel line is secured to the hydraulic port by means of a clamping fixture. Retaining clips ensure reliable fastening. The sealing ring (O-ring) on the hydraulic port (2) seals off the fuel injector at the fuel rail.

The fuel injector is electrically connected to the engine ECU.

Fuel injector operation

When the solenoid coil is de-energized, the valve needle and valve ball are pressed against the cone-shaped valve seat by the spring and the force exerted by the fuel pressure. The fuel-supply system is thus sealed off from the intake manifold. When the solenoid coil is energized, this generates a magnetic field which attracts the valve-needle solenoid armature. The valve ball lifts up from the valve seat and the fuel is injected. When the excitation current is switched off, the valve needle closes again due to spring force.

Fuel outlet
The fuel is atomized by means of an injection-orifice plate in which there are a number of holes. These holes (injection orifices) are stamped out of the plate and ensure that the injected fuel quantity remains highly constant. The injection-orifice plate is insensitive to fuel deposits. The spray pattern of the fuel leaving the injector is produced by the number of injection orifices and their configuration.

The injector is efficiently sealed at the valve seat by the cone/ball sealing principle. The fuel injector is inserted into the opening provided for it in the intake manifold. The lower sealing ring provides the seal between the fuel injector and the intake manifold.

Essentially, the injected fuel quantity per unit of time is determined by
- The primary pressure in the fuel-supply system
- The back pressure in the intake manifold and
- The geometry of the fuel-exit area

Electrical activation
An output module in the Motronic ECU actuates the fuel injector with a switching signal (Fig. 4a). The current in the solenoid coil rises (b) and causes the valve needle (c) to lift. The maximum valve lift is achieved after the time t_{pk} (pickup time) has elapsed. Fuel is sprayed as soon as the valve ball lifts off its seat. The total quantity of fuel injected during an injection pulse is shown in Figure 4d.

Current flow ceases when activation is switched off. Mass inertia causes the valve to close, but only slowly. The valve is fully closed again after the time t_{dr} (dropout time) has elapsed.

When the valve is fully open, the injected fuel quantity is proportional to the time. The non-linearity during the valve pickup and dropout phases must be compensated for throughout the period that the injector is activated (injection duration). The speed at which the valve needle lifts off its seat is also dependent on the battery voltage. Battery-voltage-dependent injection-duration extension (Fig. 5) corrects these influences.

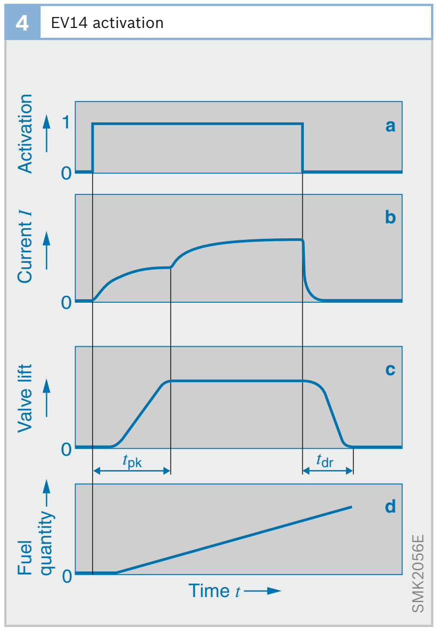
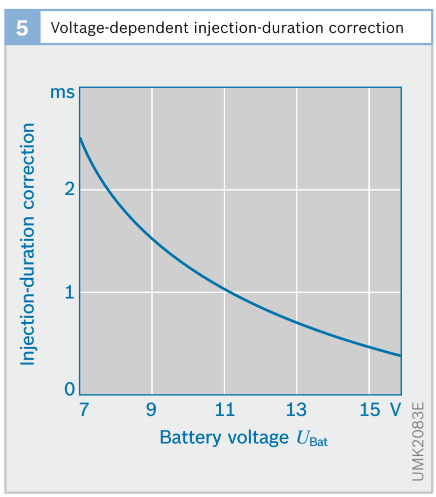


Fig. 4
a Activation signal
b Current curve
c Valve lift
d Injected fuel quantity



High-pressure fuel injector for gasoline direct injection

Function

It is the function of the high-pressure fuel injector (HDEV) on the one hand to meter the fuel and on the other hand by means of its atomization to achieve controlled mixing of the fuel and air in a specific area of the combustion chamber. Depending on the desired operating status, the fuel is either concentrated in the vicinity of the spark plug (stratified charge) or evenly distributed throughout the combustion chamber (homogenous distribution).

Design and operating principle

The high-pressure fuel injector (Fig. 6) comprises the following components:

- Inlet with filter (1)
- Electrical connection (2)
- Spring (3)
- Coil (4)
- Valve sleeve (5)
- Nozzle needle with solenoid armature (6) and
- Valve seat (7)

A magnetic field is generated when current passes through the coil. This lifts the valve needle off the valve seat against the force of the spring and opens the injector outlet bores (8). The primary pressure now forces the fuel into the combustion chamber. The injected fuel quantity is essentially dependent on the opening duration of the fuel injector and the fuel pressure.

When the energizing current is switched off, the valve needle is pressed by spring force back down against its valve seat and interrupts the flow of fuel.

Excellent fuel atomization is achieved thanks to the suitable nozzle geometry at the injector tip.

Requirements

Compared with manifold injection, gasoline direct injection differs mainly in its higher fuel pressure and the far shorter time which is available for directly injecting the fuel into the combustion chamber.

6 Design of HDEV5 high-pressure fuel injector

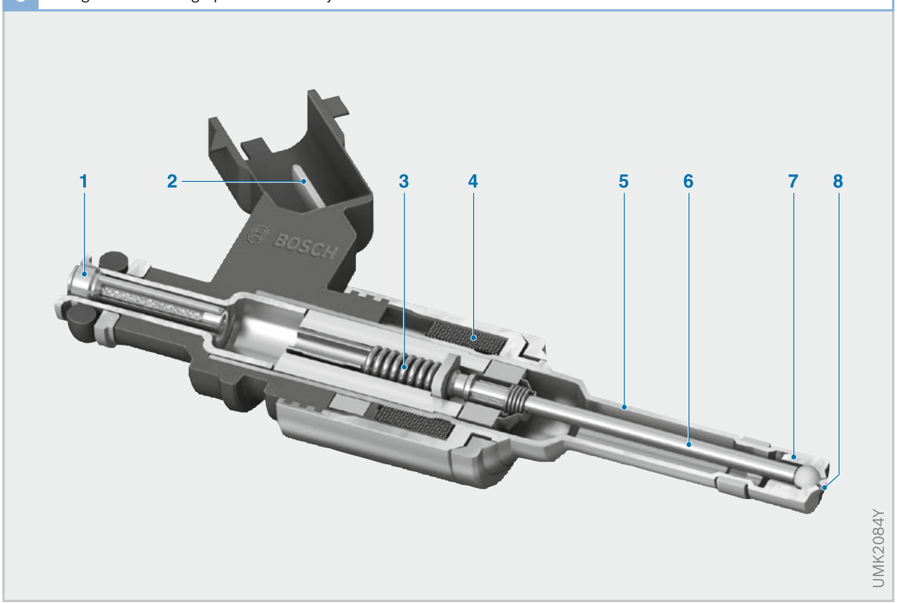


Fig. 6
1 Fuel inlet with filter
2 Electrical connection
3 Spring
4 Coil
5 Valve sleeve
6 Nozzle needle with solenoid armature
7 Valve seat
8 Injector outlet bores

Figure 7 underlines the technical demands made on the fuel injector. In the case of manifold injection, two revolutions of the crankshaft are available for injecting the fuel into the intake manifold. This corresponds to an injection duration of 20 ms at an engine speed of 6,000 rpm.

In the case of gasoline direct injection, however, considerably less time is available. In homogeneous operation, the fuel must be injected during the induction stroke. In other words, only a half crankshaft rotation is available for the injection process. At 6,000 rpm, this corresponds to an injection duration of 5 ms.

With gasoline direct injection, the fuel requirement at idle in relation to that at full load is far lower than with manifold injection (factor 1:12). At idle, this results in an injection duration of approx. 0.4 ms.

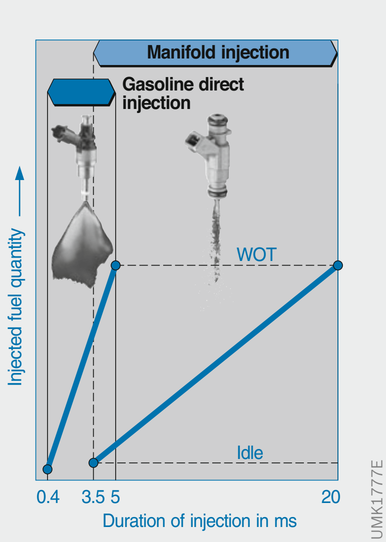
Actuation of HDEV high-pressure fuel injector

The high-pressure fuel injector must be actuated with a highly complex current curve in order to comply with the requirements for defined, reproducible fuel-injection processes (Fig. 8). The microcontroller in the engine ECU only delivers a digital triggering signal (a). An output module (ASIC) uses this signal to generate the triggering signal (b) for the fuel injector.

A DC/DC converter in the engine ECU generates the booster voltage of 65 V. This voltage is required in order to bring the current up to a high value as quickly as possible in the booster phase. This is necessary in order to accelerate the injector needle as quickly as possible. In the pickup phase (t_{pk}), the valve needle then achieves the maximum opening lift (c). When the fuel injector is open, a small control current (holding current) is sufficient to keep the fuel injector open.

With a constant valve-needle displacement, the injected fuel quantity is proportional to the injection duration (d).

7 Comparison between gasoline direct injection and manifold injection



8 Actuation of HDEV high-pressure fuel injector

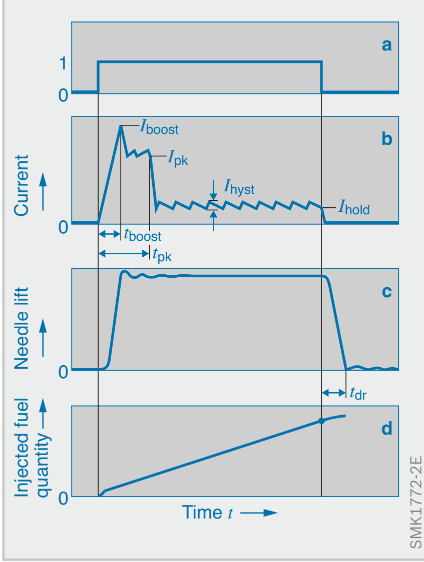


Fig. 7
Injected fuel quantity as a function of injection duration

Fig. 8
a Triggering signal
b Current curve in injector
c Needle lift
d Injected fuel quantity

Inductive ignition System

Ignition of the air-fuel mixture in the gasoline engine is electric; it is produced by generating a flashover between the electrodes on a spark plug. The ignition-coil energy converted in the spark ignites the compressed air-fuel mixture immediately adjacent to the spark plug, creating a flame front which then spreads to ignite the air-fuel mixture in the entire combustion chamber. The inductive ignition system generates in each power stroke the high voltage required for flashover and the spark duration required for ignition. The electrical energy drawn from the vehicle electrical system is temporarily stored in the ignition coil.

Design

Figure 9 shows the principle layout of the ignition circuit of an inductive ignition system. It comprises the following components:

- Ignition driver stage (4), which is integrated in the Motronic ECU or in the ignition coil
- Ignition coils (3)
- Spark plugs (5) and
- Connecting devices and interference suppressors

Generating the ignition spark

A magnetic field is built up in the ignition coil when a current flows in the primary circuit. The ignition energy required for ignition is stored in this magnetic field.

The current in the primary winding only gradually attains its setpoint value because of the induced countervoltage. Because the energy stored in the ignition coil is dependent on the current ($E = {}^1/_2 LI^2$), a certain amount of time (dwell period) is required in order to store the energy necessary for ignition. This dwell period is dependent on, among others, the vehicle system voltage. The ECU program calculates from the dwell period and the moment of ignition the cut-in point, and cuts the ignition coil in via the ignition driver stage and out again at the moment of ignition.

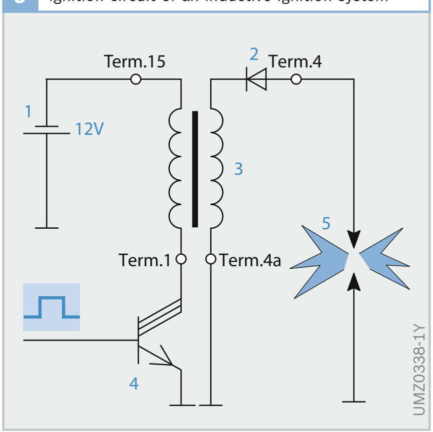
Interrupting the coil current at the moment of ignition causes the magnetic field to collapse. This rapid magnetic-field change induces a high voltage (Fig. 10) on the secondary side of the ignition coil as a result of the large number of turns (turns ratio approx. 1:100). When the ignition voltage is reached, flashover occurs at the spark plug and the compressed air-fuel mixture is ignited.

Fig. 9
1 Battery
2 AAS diode (integrated in ignition coil)
3 Ignition coil with iron core and primary and secondary windings
4 Ignition driver stage (integrated either in Motronic ECU or in ignition coil)
5 Spark plug

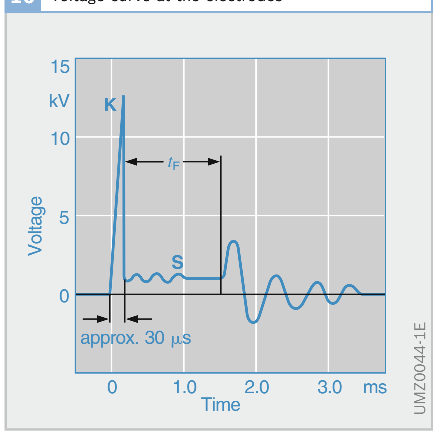
Term. 1, Term. 4, Term. 4a, Term. 15 Terminal designations

Fig. 10
K Spark head
S Spark tail
t_F Spark duration

9 Ignition circuit of an inductive ignition system



10 Voltage curve at the electrodes



Flame-front propagation

After the flashover, the voltage at the spark plug drops to the spark voltage (Fig. 10). The spark voltage is dependent on the length of the spark plasma (electrode gap and deflection due to flow) and ranges between a few hundred volts and well over 1 kV. The ignition-coil energy is converted in the ignition spark during the combustion time; this ignition spark duration lasts from as little as 100 µs to over 2 ms. Following the breakaway of the spark, the damped voltage decays.

The electrical spark between the spark-plug electrodes generates a high-temperature plasma. When the air-fuel mixture at the spark plug is ignitable and sufficient energy input is supplied by the ignition system, the arc that is created develops into a self-propagating flame front.

Moment of ignition

The instant at which the ignition spark ignites the air-fuel mixture within the combustion chamber must be selected with extreme precision. This variable has a decisive influence on engine operation and determines the output torque, exhaust-gas emissions and fuel consumption.

The influencing variables that determine the moment of ignition are engine speed and engine load, or torque. Additional variables, such as, for example, engine temperature, are also used to determine the optimal moment of ignition. These variables are recorded by sensors and then relayed to the engine ECU (Motronic). The moment of ignition is calculated and the triggering signal for the ignition driver stage is generated from program maps and characteristic curves.

Combustion knocks occur if the moment of ignition is too advanced. Permanent knocking may result in engine damage. For this reason, knock sensors are used to monitor combustion noise. After a combustion knock, the moment of ignition is delayed to too late and then slowly moved back to the pilot control value. This helps to counteract permanent knocking.

Voltage distribution

Voltage distribution takes place on the primary side of the ignition coils, which are directly connected to the spark plugs (static voltage distribution).

System with single-spark ignition coils
Each cylinder is allocated an ignition driver stage and an ignition coil (Figs. 11a and 11b). The engine ECU actuates the ignition driver stages in specified firing order. However, the system does also have to be synchronized with the camshaft by means of a camshaft sensor.

System with dual-spark ignition coils
One ignition driver stage and one ignition coil are allocated to every two cylinders (Fig. 11c). The ends of the secondary winding are each connected to a spark plug in different cylinders. The cylinders have been chosen so that when one cylinder is in the compression cycle, the other is in the exhaust cycle (only possible with engines with an even number of cylinders). It does not therefore need to be synchronized with the camshaft. Flashover occurs at both spark plugs at the moment of ignition.

11 Schematic representation of ignition coils

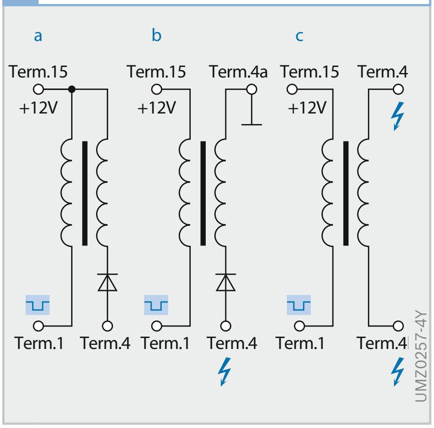


Fig. 11
a Single-spark ignition coil in economy circuit
b Single-spark ignition coil
c Dual-spark ignition coil

Ignition coils

Compact ignition coil

Design

The compact coil's magnetic circuit consists of the O core and the I core (Fig. 12), onto which the primary and secondary windings are plugged. This arrangement is installed in the coil housing. The primary winding (I core wound in wire) is electrically and mechanically connected to the primary plug connection. Also connected is the start of the secondary winding (coil body wound in wire). The connection on the spark-plug side of the secondary winding is also located in the housing, and electrical contacting is established when the windings are fitted.

12 Compact-coil design

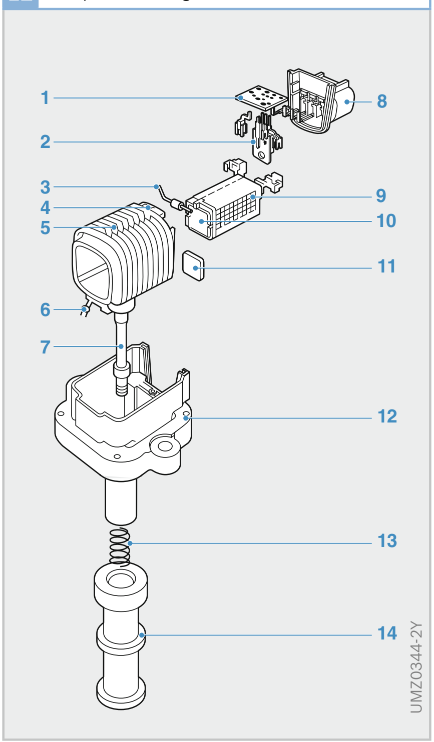


Fig. 12
1 Printed-circuit board
2 Ignition driver stage
3 AAS diode (activation arc suppression)
4 Secondary winding body
5 Secondary wire
6 Contact plate
7 High-voltage pin
8 Primary plug
9 Primary wire
10 I core
11 Permanent magnet
12 O core
13 Spring
14 Silicone jacket

Integrated within the housing is the high-voltage contact dome. This contains the contact section for spark-plug contacting, and also a silicone jacket for insulating the high voltage from external components and the spark-plug well.

Following component assembly resin is vacuum-injected into the inside of the housing, where it is allowed to harden. This produces high mechanical strength, good protection from environmental influences and outstanding insulation of the high voltage. The silicone jacket is then pushed onto the high-voltage contact dome for permanent attachment.

Remote and COP versions

The ignition coil's compact dimensions make it possible to implement the design shown in Figure 12. This version is called COP (Coil On Plug). The ignition coil is mounted directly on the spark plug, thereby rendering additional high-voltage connecting cables superfluous. This reduces the capacitive load on the ignition coil's secondary circuit. The reduction in the number of components also increases operational reliability (no rodent bites in ignition cables, etc.).

In the less common remote version, the compact coils are mounted within the engine compartment using screws. Attachment lugs or an additional bracket are provided for this purpose. The high-voltage connection is effected by means of a high-voltage ignition cable from the ignition coil to the spark plug.

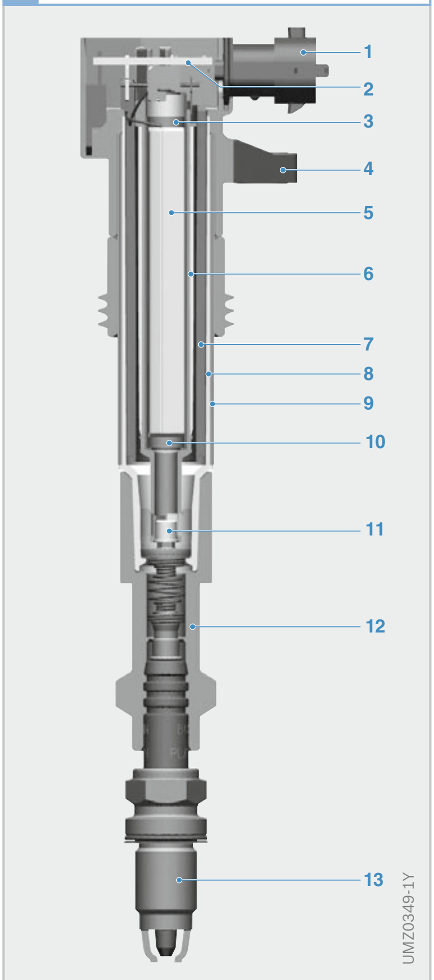
The COP and remote versions are virtually identical in design. However, the remote version (mounted on the vehicle body) is subject to fewer demands with regard to temperature and vibration conditions due to the fact that it is exposed to fewer loads and strains.

Pencil coil

The pencil coil makes optimal use of the space available within the engine compartment. Its cylindrical shape makes it possible to use the spark plug well as a supplementary installation area for ideal space utilization on the cylinder head.

Because pencil coils are always mounted directly on the spark plug, no additional high-voltage connecting cables are required.

13 Design of pencil coil



Design and magnetic circuit

Pencil coils operate like compact coils in accordance with the inductive principle. However, the rotational symmetry results in a design structure that differs considerably from that of compact coils.

Although the magnetic circuit consists of the same materials, the central rod core (Fig. 13, Item 5) consists of laminations in various widths stacked in packs that are virtually circular. The yoke plate (9) that provides the magnetic circuit is a rolled and slotted sleeve – also in electrical sheet steel, sometimes in multiple layers.

Another difference relative to compact coils is the primary winding (7), which has a larger diameter and is above the secondary winding (6), while the body of the winding also supports the rod core. This arrangement brings significant benefits in the areas of design and operation. Owing to restrictions imposed by their geometrical configuration and compact dimensions, pencil coils allow only limited scope for varying the magnetic circuit (rod core, yoke plate) and windings.

In most pencil-coil applications, the limited space available dictates that permanent magnets be used to increase the spark energy.

The arrangements for electrical contact with the spark plug and for connection to the engine wiring harness are comparable with those used for compact pencil coils.

Fig. 13
1 Plug connection
2 Printed-circuit board with ignition driver stage
3 Permanent magnet
4 Attachment arm
5 Laminated electrical-sheet-steel core (rod core)
6 Secondary winding
7 Primary winding
8 Housing
9 Yoke plate
10 Permanent magnet
11 High-voltage dome
12 Silicone jacket
13 Attached spark plug

Control of Diesel engines

System overview

Electronic control of a diesel engine enables precise and differentiated modulation of fuel-injection parameters. This is the only means by which a modern diesel engine is able to satisfy the many demands placed upon it. Electronic diesel control (EDC) is subdivided into three system blocks: sensors/setpoint generators, ECU, and actuators.

Requirements

The lowering of fuel consumption and exhaust emissions (NO_X, CO, HC, particulates) combined with simultaneous improvement of engine power output and torque are the guiding principles of current development work on diesel-engine design. Conventional indirect-injection engines (IDI) were no longer able to satisfy these requirements.

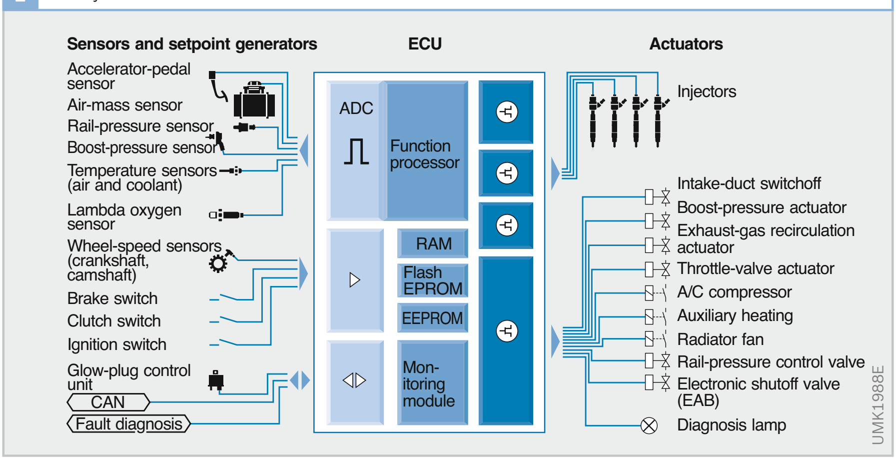
State-of-the-art technology is represented today by direct-injection diesel engines (DI) with high injection pressures for efficient mixture formation. The fuel-injection systems support several injection processes: pre-injection, main injection, and secondary injection. These injection processes are for the most part controlled electronically (pre-injection, however, is controlled mechanically on UIS for cars).

In addition, diesel-engine development has been influenced by the high levels of driving comfort and convenience demanded in modern cars. Exhaust and noise emissions are also subject to ever more stringent demands.

As a result, the performance demanded of the fuel-injection and management systems has also increased, specifically with regard to:

- High injection pressures
- Rate shaping
- Pre-injection and, if necessary, secondary injection
- Adaptation of injected fuel quantity, boost pressure and start of injection at the respective operating status
- Temperature-dependent excess-fuel quantity
- Load-independent idle speed control
- Controlled exhaust-gas recirculation
- Cruise control
- Tight tolerances for start of injection and injected-fuel quantity and maintenance of high precision over the service life of the system (long-term performance)
- Support of exhaust-gas treatment systems

1 EDC system blocks



K. Reif (Ed.), *Fundamentals of Automotive and Engine Technology*,
DOI 10.1007/978-3-658-03972-1_15, © Springer Fachmedien Wiesbaden 2014

Conventional mechanical RPM control uses a number of adjusting mechanisms to adapt to different engine operating statuses and ensures high-quality mixture formation. Nevertheless, it is restricted to a simple engine-based control loop and there are a number of important influencing variables that it cannot take account of or cannot respond quickly enough to.

As demands have increased, EDC has developed into a complex electronic engine-management system capable of processing large amounts of data in real time. In addition to its pure engine-management function, EDC supports a series of comfort and convenience functions (e.g. cruise control). It can form part of an overall electronic vehicle-speed control system ("drive-by-wire"). And as a result of the increasing integration of electronic components, complex electronics can be accommodated in a very small space.

Operating principle

Electronic diesel control (EDC) is capable of meeting the requirements listed above as a result of microcontroller performance that has improved considerably in the last few years.

In contrast to diesel-engine vehicles with conventional in-line or distributor injection pumps, the driver of an EDC-controlled vehicle has no direct influence, for instance through the accelerator pedal and Bowden cable, upon the injected fuel quantity. Instead, the injected fuel quantity is determined by a number of influencing variables. These include:

- Driver command (accelerator-pedal position)
- Operating status
- Engine temperature
- Interventions by other systems (e.g. TCS)
- Effects on exhaust emissions, etc.

The ECU calculates the injected fuel quantity on the basis of all these influencing variables. Start of injection can also be varied. This requires a comprehensive monitoring concept that detects inconsistencies and initiates appropriate actions in accordance with the effects (e.g. torque limitation or limp-home mode in the idle-speed range). EDC therefore incorporates a number of control loops.

Electronic diesel control allows data communication with other electronic systems, such as the traction-control system (TCS), electronic transmission control (ETC), or electronic stability program (ESP). As a result, the engine-management system can be integrated in the vehicle's overall control system, thereby enabling functions such as reduction of engine torque when the automatic transmission changes gear, regulation of engine torque to compensate for wheel slip, etc.

The EDC system is fully integrated in the vehicle's diagnosis system. It meets all OBD (On-Board Diagnosis) and EOBD (European OBD) requirements.

System blocks

Electronic diesel control (EDC) is divided into three system blocks (Fig. 1):

1. *Sensors and setpoint generators* detect operating conditions (e.g. engine speed) and setpoint values (e.g. switch position). They convert physical variables into electrical signals.

2. The *ECU* processes the information from the sensors and setpoint generators in mathematical computing processes (open- and closed-loop control algorithms). It controls the actuators by means of electrical output signals. In addition, the ECU acts as an interface to other systems and to the vehicle diagnosis system.

3. *Actuator*s convert the electrical output signals from the ECU into mechanical variables (e.g. solenoid-valve needle lift).

Data processing

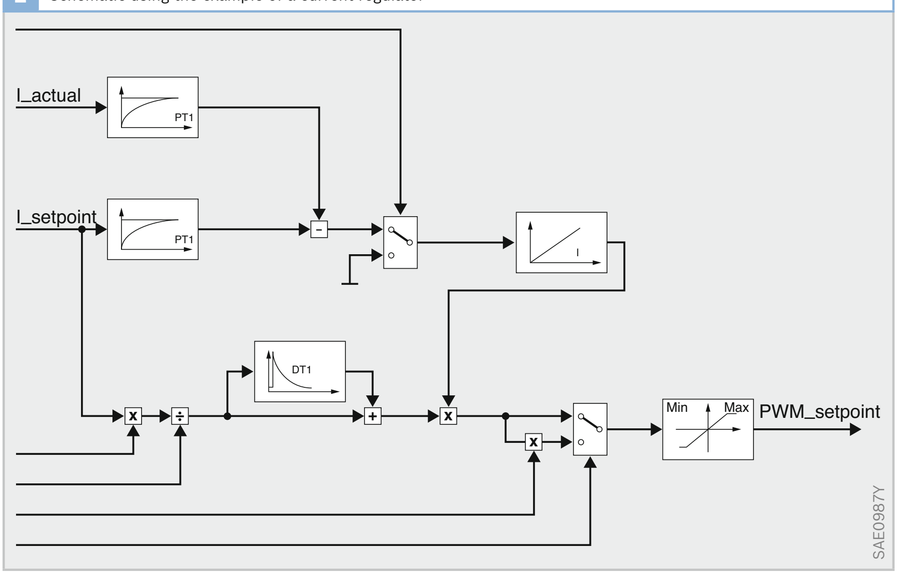
The main function of the electronic diesel control (EDC) is to control the injected fuel quantity and the injection timing. The common-rail accumulator injection system also controls injection pressure. Furthermore, on all systems, the engine ECU controls a number of actuators. The EDC functions must be matched to every vehicle and every engine. This is the only way to optimize component interaction (Fig. 3).

The control unit evaluates the signals sent by the sensors and limits them to the permitted voltage level. Some input signals are also checked for plausibility. Using this input data together with stored program maps, the microprocessor calculates the position and duration for injection timing. This information is then converted to a signal characteristic which is aligned to the engine's piston strokes. This calculation program is termed the "ECU software".

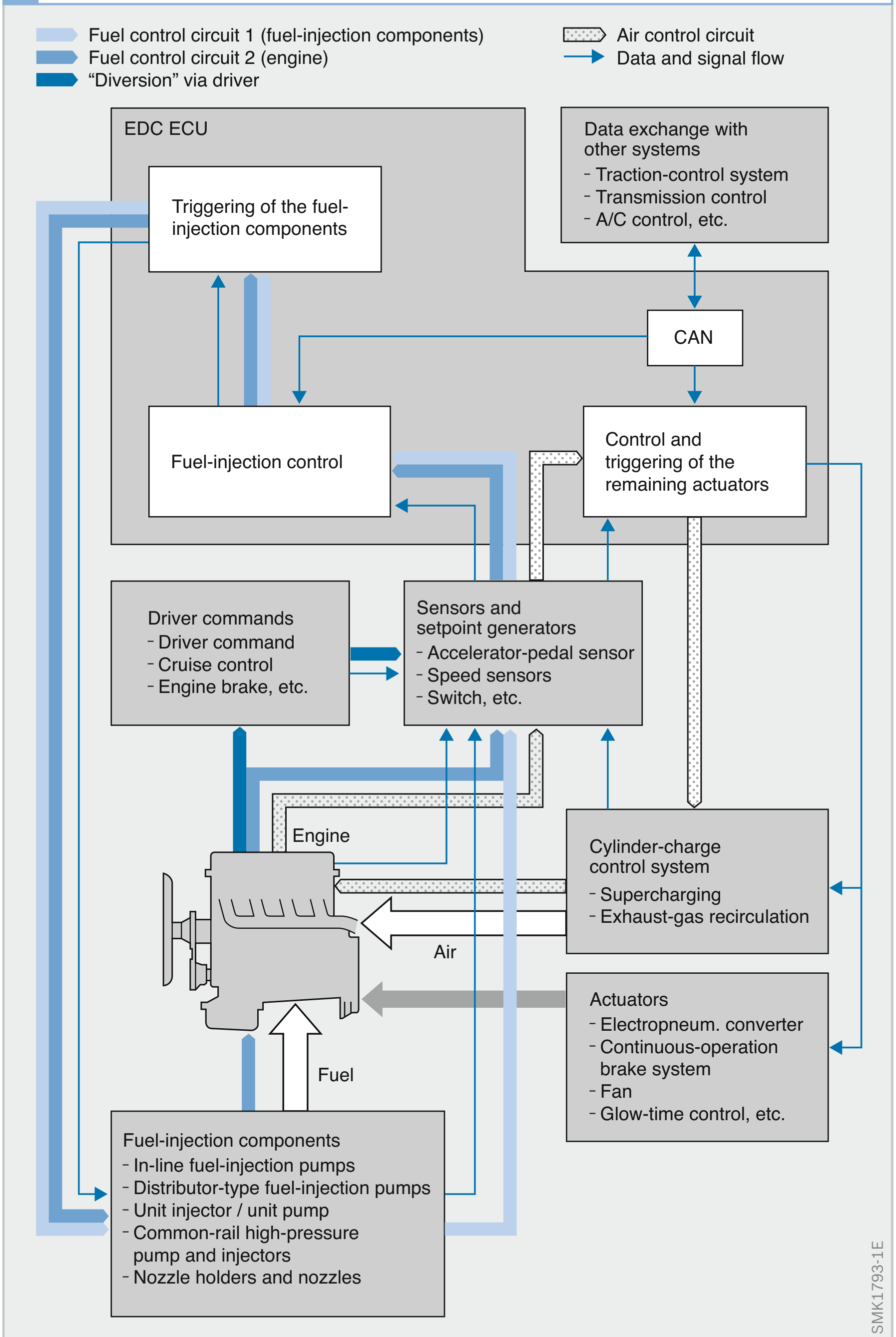
The required degree of accuracy together with the diesel engine's outstanding dynamic response requires high-level computing power. The output signals are applied to driver stages which provide adequate power for the actuators (for instance, the high-pressure solenoid valves for fuel injection, exhaust-gas recirculation positioner, or boost-pressure actuator). Apart from this, a number of other auxiliary-function components (e.g. glow relay and air-conditioning system) are triggered.

The driver-stage diagnosis functions for the solenoid valves also detect faulty signal characteristics. Furthermore, signals are exchanged with other systems in the vehicle via the interfaces. The engine ECU monitors the complete fuel-injection system as part of a safety strategy.

2 Schematic using the example of a current regulator



3 Basic sequence of electronic diesel control



Fuel-injection control

Table 1 provides an overview of the EDC functions which are implemented in the various fuel-injection systems. Figure 4 shows the sequence of fuel-injection calculations with all functions, a number of which are optional extras. These can be activated in the ECU by the after-sales service when retrofit equipment is installed.

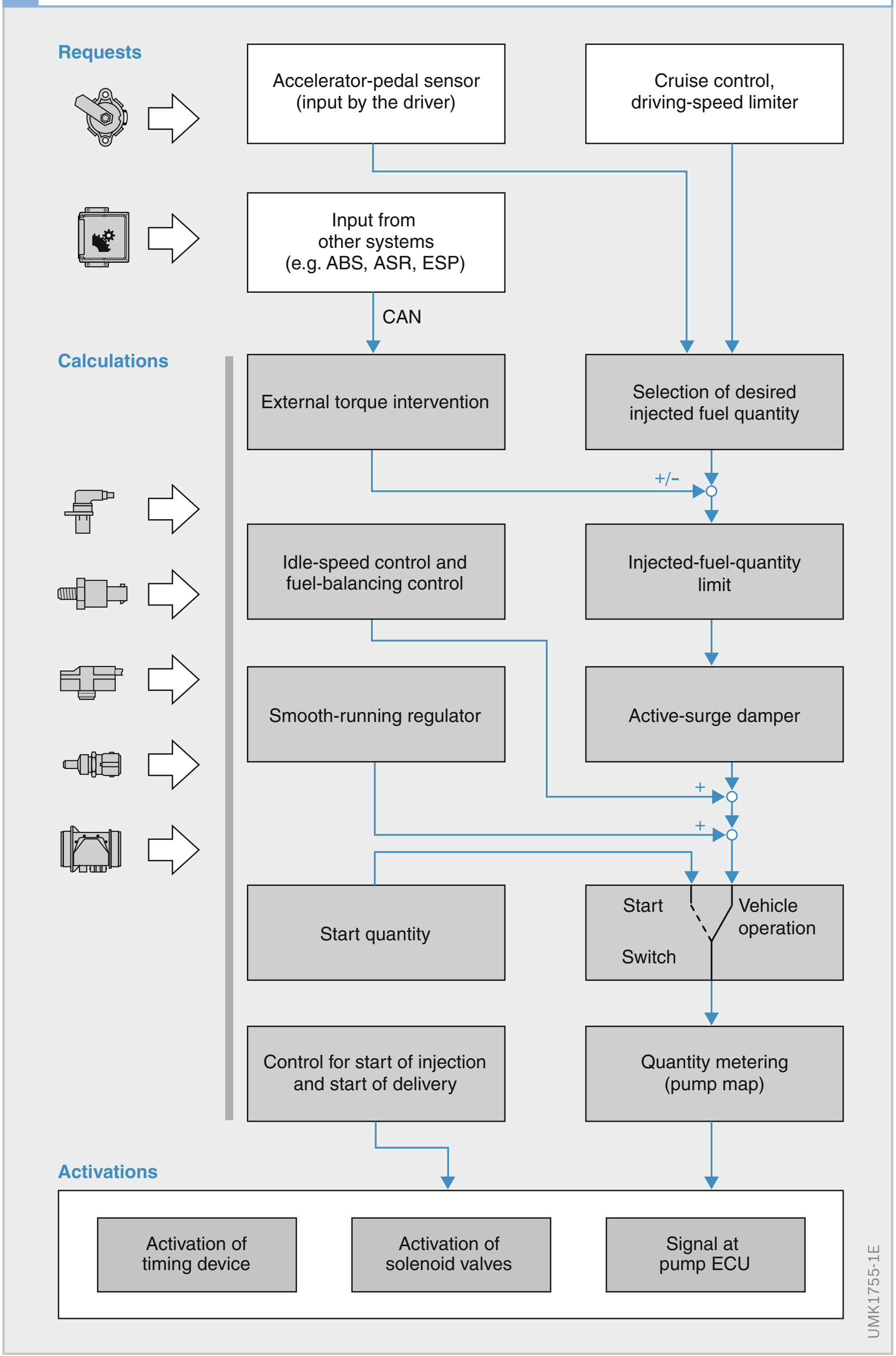
In order that the engine can run with optimal combustion under all operating conditions, the ECU calculates exactly the right injected fuel quantity for all conditions. Here, a number of parameters must be taken into account. On a number of solenoid-valve-controlled distributor-type injection pumps, the solenoid valves for injected fuel quantity and start of injection are triggered by a separate pump ECU.

1 Overview of functions of EDC variants for motor vehicles

Fuel-injection system	In-line fuel-injection pumps PE	Helix-controlled distributor-type injection pumps VE-EDC	Solenoid-valve-controlled distributor injection pumps VE-M, VR-M	Unit injector system and unit pump system UIS, UPS	Common-rail system CR
Function					
Injected-fuel-quantity limit	•	•	•	•	•
External torque intervention	• 3)	•	•	•	•
Driving-speed limitation	• 3)	•	•	•	•
Cruise control	•	•	•	•	•
Altitude correction	•	•	•	•	•
Boost-pressure control	•	•	•	•	•
Idle-speed regulation	•	•	•	•	•
Intermediate-speed regulation	• 3)	•	•	•	•
Active surge damping	• 2)	•	•	•	•
BIP control	–	–	•	•	–
Intake-port shutoff	–	–	•	• 2)	•
Electronic immobilizer	• 2)	•	•	•	•
Controlled pre-injection	–	–	•	• 2)	•
Glow control unit	• 2)	•	•	• 2)	•
A/C switch-off	• 2)	•	•	•	•
Auxiliary coolant heating	• 2)	•	•	• 2)	•
Smooth-running control	• 2)	•	•	•	•
Fuel-balancing control	• 2)	–	•	•	•
Fan activation	–	•	•	•	•
EGR control	• 2)	•	•	•	•
Start-of-injection control with sensor	• 1) 3)	•	•	•	•
Cylinder shutoff	–	–	• 3)	• 3)	• 3)
Increment-angle learning	–	–	–	•	•
Increment-angle rounding	–	–	–	• 2)	–

Table 1
1) Control-sleeve in-line fuel-injection pumps
2) Cars only
3) Commercial vehicles only

4 Calculation of fuel-injection process in ECU



Torque-controlled EDC systems

The engine-management system is continually being integrated more closely into the overall vehicle system. Vehicle-dynamics systems (e.g. TCS), comfort and convenience systems (e.g. cruise control/Tempomat), and transmission control influence electronic diesel control (EDC) via the CAN bus. Apart from this, much of the information registered or calculated in the engine-management system must be passed on to other ECUs via the CAN bus.

In order to be able to incorporate EDC even more efficiently in a functional alliance with other ECUs, and implement other changes rapidly and effectively, it was necessary to make radical changes to the newest-generation controls. These changes resulted in torque-controlled EDC, which was introduced with the EDC16. The main feature is the changeover of the module interfaces to the parameters as commonly encountered in practice in the vehicle.

Engine characteristics

Essentially, an engine's output can be defined using the three characteristics: power P, engine speed n, and torque M.

Figure 5 compares typical curves of torque and power as a function of the engine speed of two diesel engines. Basically speaking, the following formula applies:

$$P = 2 \cdot \pi \cdot n \cdot M$$

It is sufficient therefore, for example, to specify the torque as the reference variable while taking into account the engine speed. Engine power then results from the above formula. Since power output cannot be measured directly, torque has turned out to be a suitable reference variable for engine management.

Torque control

When accelerating, the driver uses the accelerator-pedal (sensor) to directly demand a given torque from the engine. Independently of the driver's requirements, other external vehicle systems submit torque demands via the interfaces resulting from the power requirements of the particular component (e.g. air-conditioning system, alternator). Using these torque-requirement inputs, the engine-management system calculates the output engine torque to be generated and controls the fuel-injection and air-system actuators accordingly. This has the following advantages:

- No system has a direct influence on engine management (boost pressure, fuel injection, preglow). The engine management system can thus also take into account other higher-level optimization criteria for the external requirements (e.g. exhaust-gas emissions, fuel consumption) and then control the engine in the best way possible.
- Many of the functions which do not directly concern the engine management system can be designed to function identically for diesel and gasoline engines.
- Expansions to the system can be implemented quickly.

5 Example of the torque and power-output curves as a function of engine speed for two car diesel engines with approx. 2.2 l engine displacement

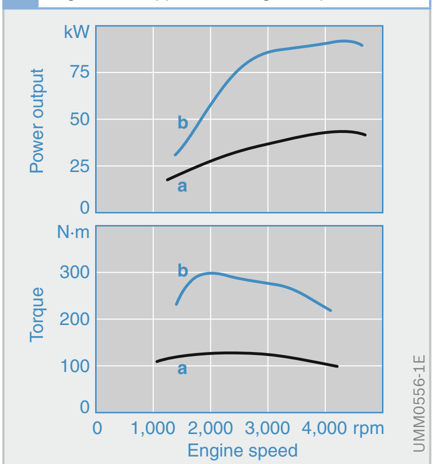


Fig. 5
a Build year 1968
b Build year 1998

Sequence of engine management

The setpoint values are processed further in the engine ECU. In order to fulfill their assignments efficiently, the engine management system's control functions all require a wide range of sensor signals and information from other ECUs in the vehicle.

Propulsion torque

The driver's input (i.e. the signal from the accelerator-pedal sensor) is interpreted by the engine management system as the request for a propulsion torque. The inputs from the cruise control and the vehicle-speed limiter are processed in exactly the same manner.

Following this selection of the desired propulsive torque, should the situation arise, the vehicle-dynamics system (TCS, ESP) increases the desired torque value when there is the danger of wheel lockup and decreases it when the wheels show a tendency to spin.

Further external torque demands

The drivetrain's torque adaptation must be taken into account (drivetrain transmission ratio). This is defined for the most part by the ratio of the particular gear, or by the torque-converter efficiency in the case of automatic transmissions. On vehicles with an automatic transmission, the transmission control stipulates the torque demand during the gearshift. This is reduced in order to produce a comfortable, smooth gearshift, thus protecting the engine. In addition, the torque required by other engine-powered auxiliary systems (e.g. air-conditioning compressor, alternator, servo pump) is determined. This torque demand is calculated either by the auxiliary systems themselves or by the engine management system. Calculation is based on the required power and engine speed, and the engine management system adds up the various torque requirements.

The vehicle's driveability remains unchanged notwithstanding varying requirements from the auxiliary systems and changes in the engine's operating states.

Internal torque demands

At this stage, the idle-speed control and the active surge damper intervene.

For instance, if demanded by the situation, in order to prevent mechanical damage, or excessive smoke due to the injection of too much fuel, the torque limitation reduces the internal torque demand. In contrast to previous engine-management systems, limitations are no longer only applied to the injected fuel quantity, but instead, depending on the required effects, also to the particular physical quantity involved.

The engine's losses are also taken into account (e.g. friction, drive for the high-pressure pump). The torque represents the engine's measurable effects to the outside. However, the engine management system can only generate these effects in conjunction with the correct fuel injection together with the correct injection point, and the necessary marginal conditions as apply to the air system (e.g. boost pressure and exhaust-gas recirculation rate). The required injected fuel quantity is determined using the current combustion efficiency. The calculated fuel quantity is limited by a protective function (e.g. protection against overheating), and if necessary can be varied by smooth-running control. During engine start, the injected fuel quantity is not determined by external inputs such as those from the driver, but rather by the separate "start quantity" control function.

Actuator triggering

The resulting setpoint value for the injected fuel quantity is used to generate the triggering data for the injection pumps and/or the fuel injectors, and for defining the optimum operating point for the intake-air system.

Lighting technology

Automotive light sources

The most important light sources for the lighting systems on the vehicle front and rear are halogen lamps, bulbs, gas-discharge lamps and LEDs.

Thermal radiators

Thermal radiators generate light from heat energy. The major liability of the thermal radiator is its low working efficiency (below 10 %) which, relative to the gas-discharge lamp, leads to very low potential for luminous efficiency.

Incandescent (vacuum) bulb

Among the thermal radiators is the bulb (Fig. 1) whose tungsten filament (2) is enclosed by glass (1). A vacuum is created inside the glass, which is why the incandescent bulb is also known as a vacuum bulb.

At 10 to 18 lm/W (lumen/Watt), the luminous efficiency of an incandescent bulb is comparatively low. During bulb operation, the tungsten particles of the filament vaporize. The glass consequently darkens over the course of the bulb's service life. The vaporization of the particles ultimately leads to the filament breaking and thus failure of the lamp. For this reason, incandescent bulbs as light sources for the headlamps have been replaced by halogen lamps. For cost reasons, however, incandescent bulbs continue to be used for other lights and as light sources in the passenger compartment. Even the lighting of passive display elements (e.g. fan, heating and air-conditioning controllers, LCD displays) is generally performed by incandescent bulbs, the color of which is changed by means of color filters for the application and design concerned.

Halogen lamp

There are two types of halogen lamp: with one or two tungsten filaments. The halogen lamps H1, H3, H7, HB3 and HB4 (see table at the end of the chapter) only have one filament. They are used as light sources for the low-beam, high-beam and fog lights.

The bulb is made of quartz glass. The quartz glass filters out the low UV content of the beam that halogen lamps emit. Unlike an incandescent bulb, the glass of a halogen lamp contains a halogen charge (iodine or bromine). This makes it possible for the filament to heat up to temperatures approaching tungsten's melting point (around 3,400 °C), thereby achieving commensurately high levels of luminous power.

1 Incandescent bulb

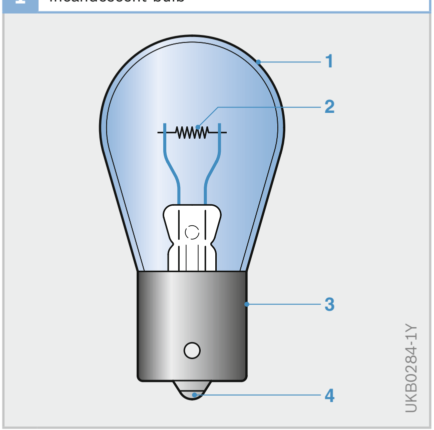


Fig. 1
1 Glass bulb
2 Filament
3 Lamp socket base
4 Electrical connection

2 H4 halogen lamp (detail from Fig. 3)

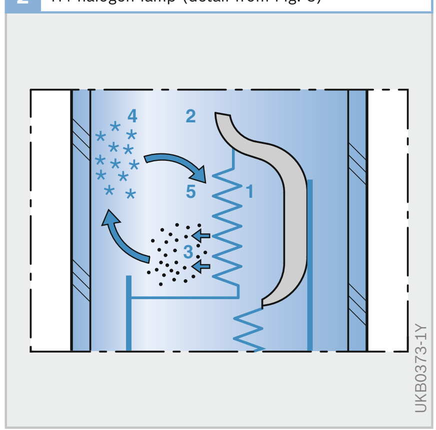


Fig. 2
1 Tungsten filament
2 Halogen charge (iodine or bromine)
3 Evaporated tungsten
4 Halogenated tungsten
5 Tungsten deposits

K. Reif (Ed.), *Fundamentals of Automotive and Engine Technology*,
DOI 10.1007/978-3-658-03972-1_16, © Springer Fachmedien Wiesbaden 2014

Close to the hot bulb wall, vaporized tungsten particles combine with the filler gas to form a transparent gas (tungsten halide). This is stable within a temperature range of approximately 200 to 1,400 °C. Tungsten particles re-approaching the filament respond to the high temperatures at the filament by dispersing to form a consistent tungsten layer. This cycle (Fig. 2) limits the wear rate of the filament. In order to maintain this cycle, an external bulb temperature of approx. 300 °C is necessary. The glass therefore encloses the filament tightly. It remains clear throughout the entire service life of the lamp.

The rate of filament wear is also limited by the high pressure that is generated in the bulb, limiting the vaporization rate of the tungsten.

The H4 halogen lamp generates the light beam in the same way but has two filaments (Fig. 3, Items 2 and 3). This means that only one lamp is required for each low-beam and high-beam headlamp.

The lower part of the low-beam filament is masked by a screen integrated in the headlamp. As a result, the light is only emitted into the upper part of the reflector (Fig. 8) and thereby prevents dazzling other road users.

Switching from low beam to high beam activates the second filament. Halogen lamps with an output of 60/55 W[1] emit around twice as much light as incandescent bulbs with an output of 45/40 W. The high luminous efficiency of around 22 to 26 lm/W is primarily the result of the high filament temperature.

Gas-discharge lamps

Gas discharge describes the electrical discharge that occurs when an electrical current flows through a gas and causes it to emit radiation (examples: sodium-vapor lamps for street lighting and fluorescent lamps for interior lighting).

The discharge chamber of the gas-discharge lamp (Fig. 4, Item 3) is filled with the inert gas xenon and a mixture of metal halides. The electrical voltage is applied between two electrodes (4) protruding into the bulb. An electronic ballast unit is required for switching on and operation. Application of an ignition voltage in the 10 to 20 kV range ionizes the gas between the electrodes, producing an electrically conductive path in the form of a luminous arc. With the alternating current (400 Hz) applied, the metallic charge is vaporized due to the temperature increase inside the bulb and light is radiated.

Under normal circumstances the lamp requires several seconds to ionize all of the particles and generate full illumination. To accelerate this process, an increased starting current flows until this point.

3 H4 halogen lamp

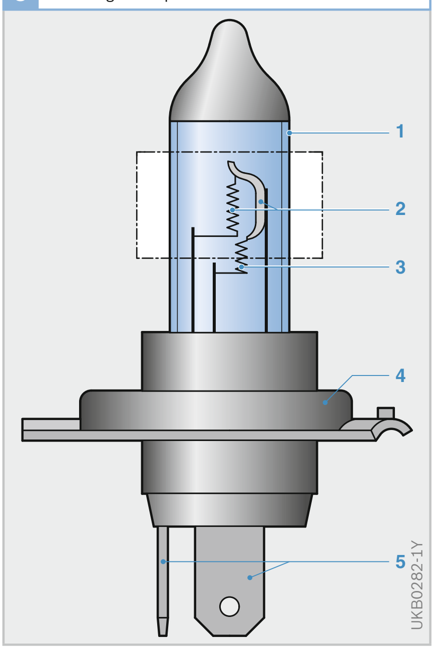


Fig. 3
1 Glass bulb
2 Low-beam filament with cap
3 High-beam filament
4 Lamp base
5 Electrical connection

[1] High beam/low beam

When maximum luminous power is achieved, limitation of lamp current commences. A sustained operating voltage of only 85 V is sufficient to maintain the arc.

Light sources relying on the gas-discharge concept acquired new significance for automotive applications with the advent of the “Litronic” electronic lighting system. This concept features several crucial benefits compared with conventional bulbs:

- Greater range of the headlamp beam
- Brighter and more even carriageway illumination
- Longer service life, as there is no mechanical wear
- High luminous efficiency (approximately 85 lm/W) due to the emission spectrum being predominately in the visible spectral range
- Improved efficiency thanks to lower thermal losses
- Compact headlamp designs for smooth front-end styling

The D2/D4-series automotive gas-discharge lamps feature high-voltage-proof sockets and UV glass shielding elements. On the D1/D3-series models, the high-voltage electronics necessary for operation are also integrated in the lamp socket. All series can be broken down into two subcategories:

- Standard lamp (S lamp) for projection headlamps (Fig. 4) and
- Reflection lamp (R lamp) for reflection headlamps (Fig. 5). They have an integrated shutter (3) to create the light-dark cutoff, comparable with the shutter in the H4 lamp.

Until now, gas-discharge lamps with the type designations D1x and D2x were used. From 2007, the D3/D4-series will also be fitted as standard. These have a lower operating voltage, a different charge gas composition, and different arc geometries.

4 D2S gas-discharge lamp

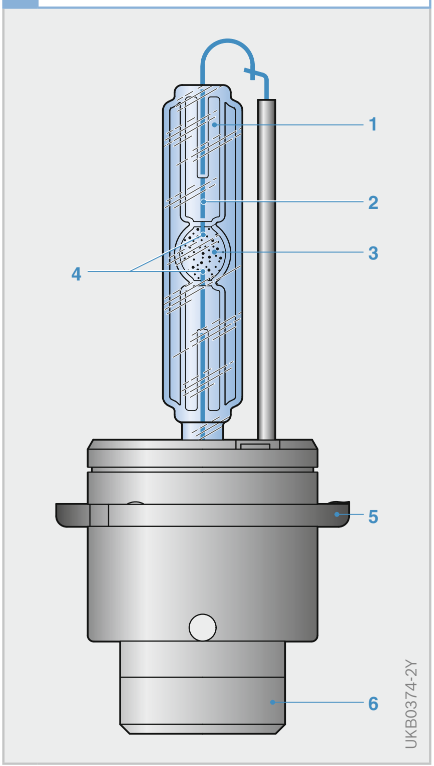


Fig. 4
Gas-discharge lamp for projection headlamps
1 Glass capsule with UV shield
2 Electrical lead
3 Discharge chamber
4 Electrodes
5 Lamp base
6 Electrical connection

5 D2R gas-discharge lamp

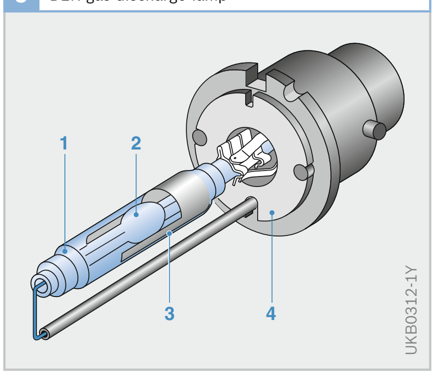


Fig. 5
Gas-discharge lamp for reflection headlamps
1 UV inert-gas bulb
2 Discharge chamber
3 Shutter
4 Lamp base

Light emitting diodes

The light emitting diode (LED) is an active light element. If an electrical voltage is applied, current flows through the chip. The electrons of the atoms of the LED chip are highly energized by the voltage. As light is emitted, they return to their initial state of low energy charge.

The 0.1 to 1 mm small semiconductor crystal is seated on a reflector that directs the light with pin-point precision.

LEDs are commonly used as light sources for lights on the rear of the vehicle, especially the additional stop lamps located in the center. They make it possible for a narrow, linear beam to be emitted.

By comparison with incandescent bulbs, LEDs are beneficial in that they emit maximum output in less than a millisecond. An incandescent bulb takes approximately 200 ms. LEDs, for example, are therefore able to emit the brake signal sooner and thus shorten the response time to the brake signal (brake pedal depressed) for drivers behind.

In the motor vehicle, LEDs are used as illuminators or in displays, in the interior they are used for lighting, in displays or display backlighting. In the lighting system, they find use as auxiliary stop lamps and tail lamps, and, increasingly in future, as daytime running lamps and in headlamps.

Technical lighting variables

Luminous intensity

The brightness of light sources can vary. Luminous intensity serves as an index for comparing them. It is the visible light radiation that a light source projects in a specific direction.

The unit for defining levels of luminous intensity is the candela (cd), roughly equivalent to the illumination emitted by one candle. The brightness of an illuminated surface varies according to its reflective properties, the luminous intensity and the distance separating it from the light source.

Examples of permissible values

Stop lamp (individual): 60 to 185 cd
Tail lamp (individual): 4 to 12 cd
Rear fog lamp (individual): 150 to 300 cd
High beam (total, maximum): 225,000 cd

Luminous flux

Luminous flux is that light emitted by a light source that falls within the visible wavelength range.
Values are expressed in lumen (lm).

Illuminance

The illuminance is the luminous flux arriving at a given surface. It increases proportionally along with the light intensity, and decreases with the square of the distance.

Illuminance is expressed in lux (lx):
$1 \text{ lx} = 1 \text{ lm/m}^2$

Range

The range is defined as the distance at which the illuminance in the light beam still has a given value (e.g. 1 lx). The geometric range is the distance at which the horizontal part of the light-dark cut-off is shown on the road surface with the headlamps on low beam.

Main headlamps (Europe)

Function

On the one hand, the main headlamps must provide maximum visual range while at the same time ensuring that the glare effect for oncoming traffic is kept to a minimum and that light distribution immediately in front of the vehicle remains in line with the requirements of safe operation. It is vital to provide the lateral illumination needed to safely negotiate bends, i.e. the light must extend outward to embrace the verge of the road. Although it is impossible to achieve absolutely consistent luminance across the entire road surface, it is possible to avoid sharp contrasts in light density.

High beam

The high beam is usually generated by a light source located at the reflector's focal point, causing the light to be reflected outward along a plane extending along the reflector's axis (Fig. 7). The maximum luminous intensity which is available during high-beam operation is largely a function of the reflector's mirrored surface area.

In four and six-headlamp systems, in particular, purely parabolic high-beam reflectors can be replaced by units with complex geometrical configurations for simultaneous use of high and low beams.

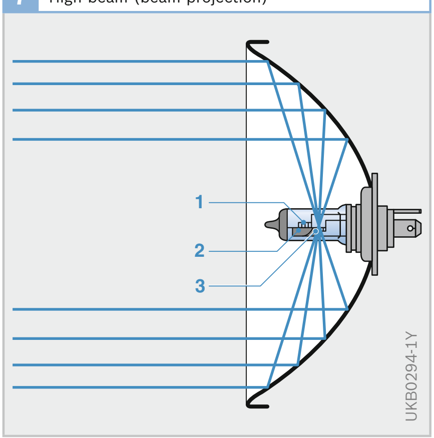
In these systems the high-beam component is designed to join with the low beam (simultaneous operation) to produce a harmonious overall high-beam distribution pattern. This strategy abolishes the annoying overlapping sector that would otherwise be present at the front of the light pattern.

Low beam (dipped beam)

The high traffic density on modern roads severely restricts the use of high-beam headlamps. The low beams serve as the primary source of light under normal conditions. Basic design modifications implemented within recent years are behind the substantial improvements in low-beam performance. Developments have included:

- Introduction of the asymmetrical low-beam pattern, characterized (RHD traffic) by an extended visual range along the right side of the road.
- Introduction of new headlamp systems featuring complex geometrical configurations (PES[1]), free-form surfaces[2]), faceted reflectors[3])) offering efficiency-level improvements of up to 50 %.
- Headlamp leveling control (also known as vertical aim control) devices adapt the attitude of the headlamps to avoid dazzling oncoming traffic when the rear

7 High beam (beam projection)



8 Low beam (beam projection)

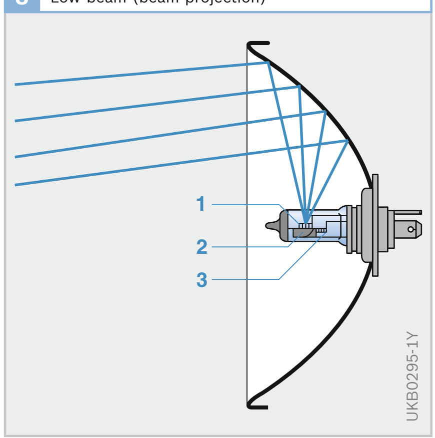


Fig. 7
1 Low-beam filament
2 Cap
3 High-beam filament at focal point

Fig. 8
1 Low-beam filament
2 Cap
3 High-beam filament

[1]) The PES (Poly-Ellipsoid System) headlamp system works with an imaging optical lens. Unlike with conventional headlamps, the light pattern generated by the reflector is reproduced on the roadway by the lens together with a screen for creating the light-dark cut-off.

[2]) Reflectors with small short focal length whose shape is calculated using special programs (CAL: Computer Aided Lighting). In this way, three separate reflectors for low beam, high beam and fog lamp can be accommodated within the same space needed by a conventional parabolic reflector, while luminous efficiency is increased at the same time.

[3]) With facetted reflectors, the surface is divided into individually optimized segments. This results in reflector surfaces with high levels of homogeneity and sideways beam spread.

of the vehicle is heavily laden. Vehicles must also be equipped with headlamp washer systems.
- "Litronic" gas-discharge lamps supply more than twice as much light as conventional halogen lamps.

Operating concept
Low-beam headlamps need a light-dark cutoff in the light pattern. In the case of H4 halogen headlamps and Litronic headlamps with D2R bulbs, this is achieved by the image from the shield (H4) or the shutter (D2R). On headlamps for all-round use (H1-, H7-, HB11 bulbs), the light-dark cutoff is achieved by the special imaging of the filament.

Headlamp systems
Dual-headlamp systems rely on a single shared reflector for low- and high-beam operation, e.g. in combination with a dual-filament H4 bulb (Fig. 9 a).

In quad headlamp systems one pair of headlamps may be switched on in both modes or during low-beam operation only, while the other pair is operated exclusively for high-beam use (Fig. 9 b).

9 Headlamp systems

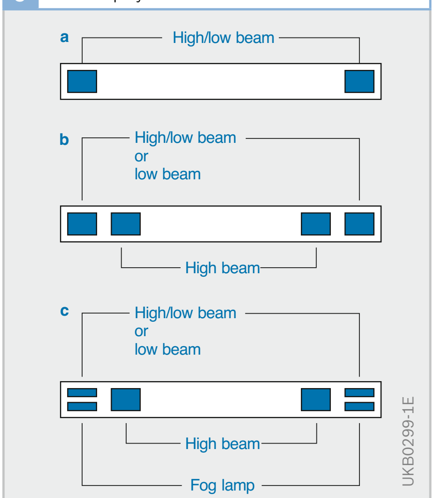


Fig. 9
a Dual-headlamp system
b Quad-headlamp system
c Six-headlamp system

Six-headlamp systems differ from the quad configuration by incorporating a supplementary fog lamp within the main headlamp assembly (Fig. 9 c).

Main headlamps (North America)

High beam
The designs for high-beam headlamps are the same as in Europe. Facetted reflectors with, for example, HB5 or H7 lamps are used.

Low beam (dipped beam)
Headlamps with a light-dark cutoff that rely on visual/optical adjustment procedures have been approved in the USA since 1 May, 1997. This has made it possible to equip vehicles for Europe and the USA with headlamps of the same type and, in some cases, even the same reflectors.

Regulations
The regulations for the attachment and wiring of main headlamps are comparable with the European regulations (Federal Motor Vehicle Safety Standard [FMVSS] No. 108 and SAE Ground Vehicle Lighting Standards Manual).

An amendment to FMVSS 108 that entered effect in 1983 made it possible to start using headlamp units of various shapes and sizes with replaceable bulbs. These were known as the RBH, or Replaceable Bulb Headlamps.

Headlamp systems
North America mirrors European practice in employing dual, quad and six-headlamp systems.

Litronic

Overview

The “Litronic” (Light-Electronics) headlamp system uses xenon gas-discharge lamps that produce a powerful lighting effect despite the low front-end surface area requirement. The illumination of the carriageway represents a substantial improvement over that provided by conventional halogen units (Fig. 10).

The light generated contains a higher proportion of green and blue and is thus more similar to the spectral distribution of sunlight. Night-time driving is therefore less exacting for the driver.

Design

The components of the Litronic headlamp system are:

- Optical unit with xenon gas-discharge lamp (S lamp, R lamp; see “Gas-discharge lamps” section)
- Electronic ballast unit with igniter and ECU

10 Light pattern on the road (comparison)

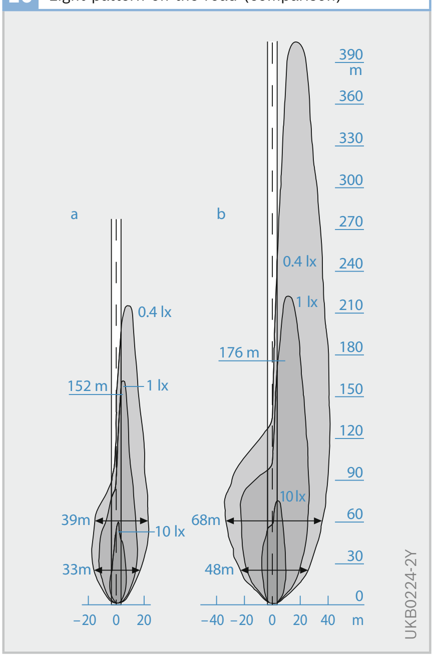


Fig. 10
a H4 lamp
b Litronic PES D2S lamp

For low beam, the headlamps with xenon gas-discharge lamps are installed in a quad system that is combined with the high-beam headlamps of the conventional design.

With the Bi-Litronic system, however, the low and high beams are generated by only one gas-discharge lamp from a dual-headlamp system.

An integral part of the headlamp is the electronic ballast unit responsible for activating and monitoring the lamp.
Its functions include:

- Ignition of the gas discharge (voltage 10 to 20 kV)
- Regulated power supply during the warm-up phase when the lamp is cold
- Demand-oriented supply in continuous operation

The control units for the individual lamp types are generally developed for a specific design type and are not universally interchangeable.

Operating principle

In the gas-discharge lamp, the arc is ignited when the light is switched on. A high voltage of 18 to 20 kV is required for this to be possible. 85 V are required to maintain the arc after ignition. The voltage is generated and regulated by an electronic

11 Electronic ballast unit

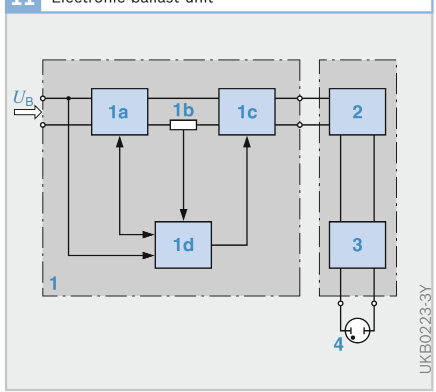


Fig. 11
Electronic ballast unit for 400 Hz alternating current supply and pulse ignition of the lamp

1 Control unit
1a DC/DC converter
1b Shunt
1c DC/AC converter
1d Microprocessor
2 Igniter
3 Lamp socket
4 D2S lamp
U_B Battery voltage

ballast unit (igniter, Fig. 11). After ignition, the gas-discharge lamp is operated for approximately 3 secs with an elevated starting current (approximately 2.6 A) so that it achieves maximum luminosity with minimal delay. The bulb's output in this period is anywhere up to 75 W. During continuous-running operation, it is 35 W.

The maximum luminous efficiency of approximately 90 lm/W is achieved once the plasma has heated the quartz glass to approximately 900 °C. Once the gas-discharge lamp has achieved maximum luminosity, the ballast unit reduces the current output to the bulb to approximately 0.4 A for continuous-running operation.

Fluctuations in the vehicle system voltage are for the most part compensated for by the ballast unit to prevent luminous flux variations. If the bulb goes out, e.g. due to an extreme voltage drop (below 9 V) or increase (above 16.5 V) in the vehicle electrical system, it is automatically reignited without delay. The reignition is limited to five attempts for safety reasons. The power supply is then interrupted by the ballast unit.

Bi-Litronic "Reflection"
The "Reflection" Bi-Litronic system makes it possible to generate the low and high beams using only one gas-discharge lamp (DR2 lamp) from a dual-headlamp system. The concept relies on an electromechanical positioner that responds to the high/low-beam switch by varying the attitude of the gas-discharge lamp within the reflector. It alternates between two different positions to generate separate projection patterns for low and high beam (Fig. 12).

This layout gives Bi-Litronic the following major advantages:

- Xenon light for high-beam operation
- Visual guidance provided by the continuous shift in light distribution from close to extended range

12 Bi-Litronic "Reflection"

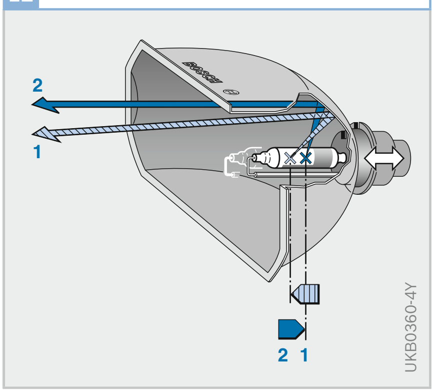


Fig. 12
1 Low beam
2 High beam

13 Litronic 4 system in reflection headlamp with integral dynamic headlamp leveling control (example)

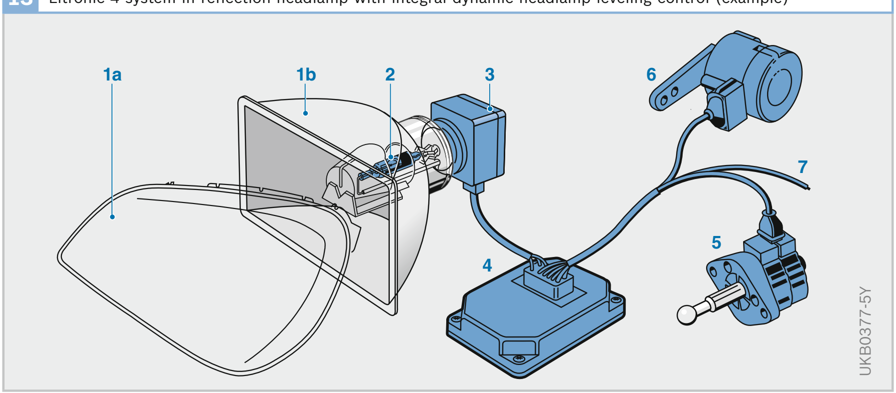


Fig. 13
1a Lens with or without scatter optics
1b Reflector
2 Gas-discharge lamp
3 Igniter
4 Control unit
5 Stepper motor
6 Axle sensor
7 To the vehicle electrical system

- Substantial reduction in space requirements as compared to a conventional quad headlamp system
- Lower costs through the use of just one gas-discharge bulb and one ballast unit per headlamp
- Greater freedom in headlamp design due to the individual reflector shape.

Special design variants of the Bi-Litronic "Reflection" lamp involve solutions in which the entire reflector is moved or individual components of the bulb cover are opened.

Bi-Litronic "Projection"
The Bi-Litronic "Projection" system is based on a PES Litronic headlamp. It shifts the position of the shutter for the light-dark cutoff to provide xenon light for high-beam operation.

With lens diameters of 60 and 70 mm, the Bi-Litronic "Projection" is the most compact combined low- and high-beam headlamp on the market, yet it still provides superb illumination.

The essential advantages of the Bi-Litronic "Projection" are:
- Xenon light for high-beam operation
- Most compact solution for high and low beams
- Modular system

Headlamp leveling control

Function

Without headlamp leveling control, the range of the headlamps would alter with a change in load or operating condition of the vehicle (constant-speed travel, stationary, acceleration, braking). The headlamp leveling control adjusts the tilt angle of the low beam to the tilt angle of the vehicle body. This results in a permanently good visual range with no dazzling of oncoming traffic under all load conditions.

14 Bi-Litronic "Projection"

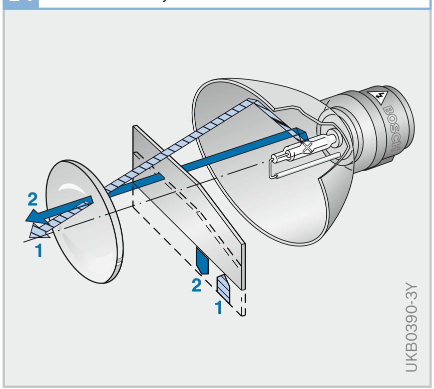


Fig. 14
1 Low beam
2 High beam

15 Litronic 2 system in projection headlamp (example)

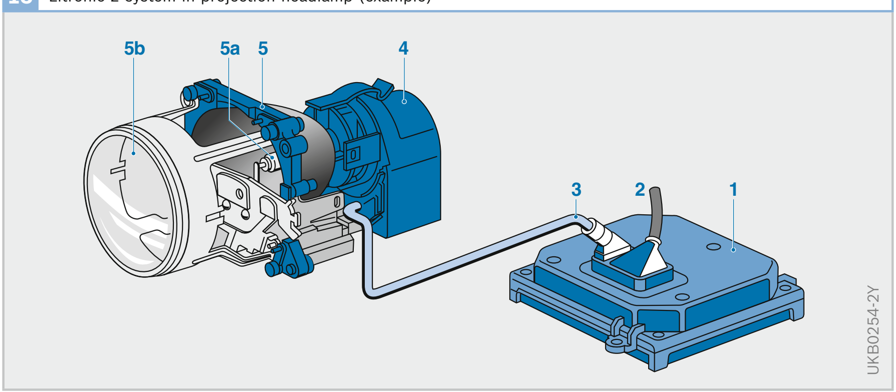


Fig. 15
1 Control unit
2 To the vehicle electrical system
3 Shielded cable
4 Igniter
5 Projection module
5a D2S lamp
5b Lens

Designs

All headlamp leveling controls feature actuators that move the headlamp reflector (housing-type design) or headlamp unit up and down. Automatic systems rely on sensors that monitor suspension travel as the basis for generating proportional signals for transmission to the aiming actuators. Manually operated units employ a switch near the driver's seat to control the setting.

Automatic headlamp leveling control

Automatic headlamp-leveling control systems fall into two categories: static and dynamic. While static systems compensate for load variations in the luggage and passenger compartments, dynamic systems also correct headlamp aim during acceleration – both from standing starts and when underway – and when braking.

The components of a typical headlamp-leveling control system include (Fig. 16):

- Sensors on the vehicle axles (Items 3 and 6) to measure the body's inclination or tilt angle.
- An ECU (5) that uses the sensor signals as the basis for calculating the vehicle's pitch angle. The ECU compares this data with the specified values and responds to deviations by transmitting appropriate triggering signals to the headlamps' servomotors.
- Servomotors (2) to adjust the headlamps to the correct angle.

Static system

In addition to the signals from the suspension sensors, the static system's control unit also receives a speed signal from the electronic speedometer. The controller relies on this signal to decide whether the vehicle is stationary, undergoing a dynamic change in speed, or proceeding at a constant speed. Automatic systems based on the static concept always feature substantial response inertia, so the system corrects only those vehicle inclinations that are consistently registered over relatively long periods.

Each time the vehicle has pulled away, it corrects the headlamp adjustment as a function of the vehicle's load. This adjustment is checked again when the vehicle steadies into constant-speed travel and is then corrected if necessary. Deviations between the target and actual position are evened out by the system.

The static system only generally requires a sensor on the rear axle of the vehicle. A DC motor is used as each headlamp's actuator.

16 Principle of an automatic headlamp-leveling control system (dynamic system)

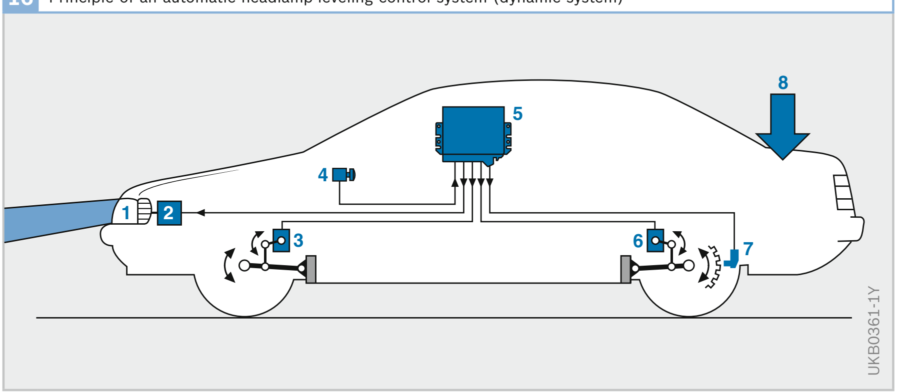


Fig. 16
1 Headlamp
2 Actuator
3 Front-suspension travel sensor
4 Light switch
5 Electronic control unit
6 Rear-suspension travel sensor
7 Speed sensor
8 Load

Dynamic system

The dynamic automatic system relies on two distinct operating modes to ensure optimal headlamp orientation under all driving conditions. Supplementary capabilities in speed-signal analysis over the static headlamp leveling control endow the system with the ability to differentiate between acceleration and braking.

With the vehicle driving at constant speed, the dynamic system, like the static system, remains in the range that features a high level of damping but as soon as the controller registers acceleration or braking, the system immediately switches to its dynamic mode. Faster signal processing and the higher servomotor adjustment speed allow the headlamp range to be re-adjusted within fractions of a second. Following acceleration or braking, the system automatically reverts to operation in its delayed-response mode.

Due to the greater dynamics requirements, the dynamic system needs one sensor per vehicle axle and rapid stepping motors to adjust the headlamps.

Adaptive lighting systems

Adaptive frontlighting system (AFS)

From 2007, function enhancements for headlamp systems based on a new EC control are permitted. The vehicle may then also have motorway lights, adverse weather lights and city lights. The optimum light pattern for each of the functions is identified and automatically selected by the vehicle electronics in response to evaluation of various vehicle sensors.

The first vehicles with AFS systems were registered back in mid-2006 thanks to an EU waiver for road traffic.

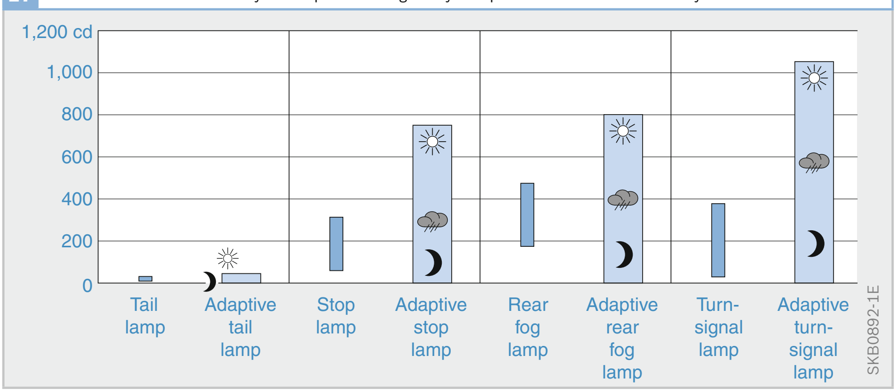
Adaptive rearlighting system (ARS)

Until now, the rear lights for vehicle perimeter lighting were equipped with single level switching. Depending on the type and design, these produced an invariable luminous intensity within the legal limit values.

Today, a multitude of sensors are used to determine environmental parameters and light conditions (brightness, dirt, visual range, wet conditions, etc.). To achieve optimum visibility (sufficient luminous intensity without excessive glare), the rear lights may in future vary luminous intensity to suit the vehicle surrounding (Fig. 17).

A stop lamp, for example, would be lit with high luminous intensity in sunlight and with low luminous intensities at night to ensure that other road users are able to recognize and draw the correct conclusion from the action of the vehicle.

17 Permitted luminous intensity of adaptive rear lights by comparison with conventional systems



Cornering lights (Europe)

The cornering lights function that has been approved for use in Germany since 2003 improves visual range on corners and in turning situations. This is made possible by a variation of the horizontal illumination of the area in front of the vehicle. With static cornering lights, this is achieved by supplementary reflectors being switched; with dynamic cornering lights, the headlamp module is pivoted laterally (Fig. 18).

During the control process, the light module or the reflector elements are pivoted by a stepping motor. The pivot angle and pivot speed are calculated by the cornering lights ECU as a function of vehicle speed and the steering angle. Sensors detect the adjustment angle of the headlamps and use failsafe algorithms to prevent dazzling of oncoming traffic in the event of a system malfunction.

18 Aiming strategy of the basic and turning modules of a static/dynamic cornering headlamp (left side)

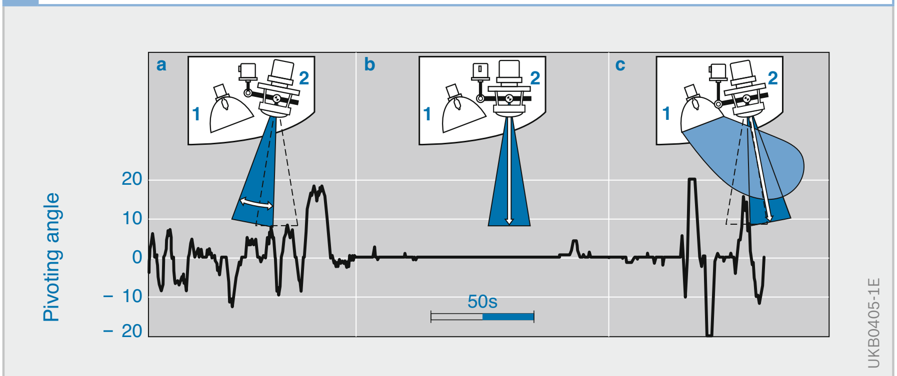


Fig. 18
- a “Country road/bends” position
- b “Motorway” position
- c “Town/turning” position
- 1 Turning module
- 2 Basic module

19 Improvement in the adaptive light pattern of the cornering headlamps

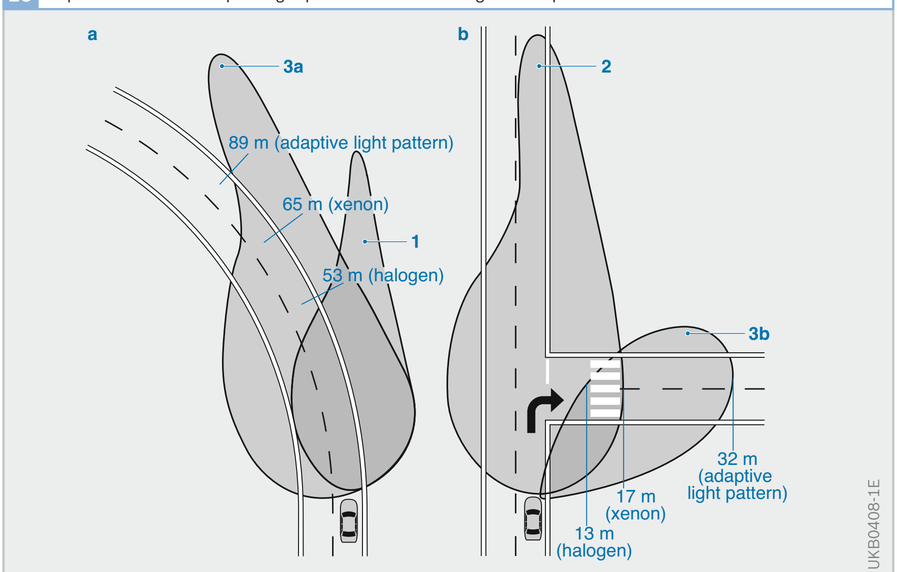
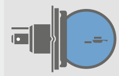
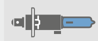
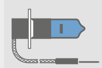
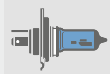
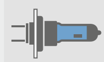
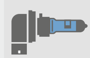
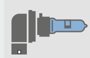
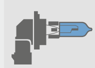
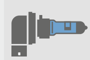
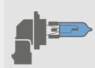
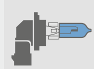
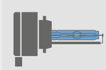
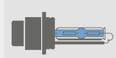
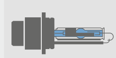
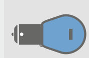


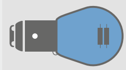
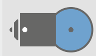
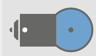
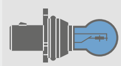
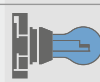
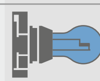
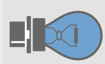
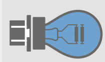
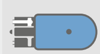
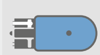
Fig. 19
- a Dynamic cornering lights, cornering to the left
- b Static cornering light, turning to the right
- 1 Light pattern of halogen headlamps
- 2 Light pattern of xenon headlamps
- 3a Adaptive light pattern: dynamic cornering lights
- 3b Adaptive light pattern: static cornering lights

Table 1

1 Specifications for motor-vehicle bulbs (2-wheeled vehicles not included)

Application	Category	Voltage rated value V	Power rated value W	Luminous flux setpoint value Lumen	IEC base type	Illustration
High beam, low beam	R2	6 12 24	45/40 1) 45/40 55/50	600 min/ 400–550 1)	P 45 t-41	
Fog lamp, high beam, low beam in 4-headlamps	H1	6 12 24	55 55 70	1,350 2) 1,550, 1,900	P14.5 e	
Fog lamp, high beam	H3	6 12 24	55 55 70	1,050 2) 1,450 1,750	PK 22 s	
High beam, low beam	H4	12 24	60/55 75/70	1,650/ 1,000 1), 2) 1,900/1,200	P 43 t-38	
High beam, low beam in 4-headlamps, fog lamp	H7	12 24	55 70	1,500 2) 1,750	PX 26 d	
Fog lamp, static cornering light	H8	12	35	800	PGJ 19-1	
High beam	H9	12	65	2,100	PGJ 19-5	
Fog lamp	H10	12	42	850	PY 20 d	
Low beam, fog lamp	H11	12 24	55 70	1,350 1,600	PGJ 19-2	
Low beam in 4-headlamps	HB4	12	55	1,100	P 22 d	
High beam in 4-headlamp	HB3	12	60	1,900	P 20 d	
Low beam, high beam	D1S	85 12 5)	35 approx. 40 5)	3,200	PK 32 d-2	
Low beam, high beam	D2S	85 12 5)	35 approx. 40 5)	3,200	P 32 d-2	
Low beam, high beam	D2R	85 12 5)	35 approx. 40 5)	2,800	P 32 d-3	
Stop, turn-signal, rear fog, revers-ing lamp	P 21 W PY 21 W 6)	6 12 24	21	460 3)	BA 15 s	

1 Specifications for motor-vehicle bulbs (continued)

Application	Category	Voltage rated value V	Power rated value W	Luminous flux setpoint value Lumen	IEC base type	Illustration
Stop lamp/ tail lamp	P 21/5 W	6 12 24	21/5 4) 21/5 21/5	440/35 3), 4) 440/35 3), 4) 440/40 3)	BAY 15 d	
Side-marker lamp, tail lamp	R 5 W	6 12 24	5	50 3)	BA 15 s	
Tail lamp	R 10 W	6 12 24	10	125 3)	BA 15 s	
Daytime running light	P 13 W	12	13	250 3)	PG 18.5 d	
Stop lamp, turn signal	P 19 W PY 19 W	12 12	19 19	350 3) 215 3)	PGU 20/1 PGU 20/2	
Rear fog lamp, reversing lamp, front turn signal	P 24 W PY 24 W	12 12	24 24	500 3) 300 3)	PGU 20/3 PGU 20/4	
Stop, turn-signal, rear fog, revers-ing lamp	P 27 W	12	27	475 3)	W 2.5 x 16 d	
Stop lamp/ tail lamp	P 27/7 W	12	27/7	475/36 3)	W 2.5 x 16 q	
License-plate lamp, tail lamp	C 5 W	6 12 24	5	45 3)	SV 8.5	
Reversing lamp	C 21 W	12	21	460 3)	SV 8.5	
Side-marker lamp	T 4 W	6 12 24	4	35 3)	BA 9 s	
Side-marker lamp, license-plate lamp	W 5 W	6 12 24	5	50 3)	W 2.1 x 9.5 d	
Side-marker lamp, license-plate lamp	W 3 W	6 12 24	3	22 3)	W 2.1 x 9.5 d	

1) High/low beam. 2) Setpoint values at test voltage of 6.3; 13.2 or 28.0 V.
3) Setpoint values at test voltage of 6.75; 13.5 or 28.0 V. 4) Main/secondary filament.
5) With ballast unit. 6) Yellow-light version.

Electronic stability program

The electronic stability program (ESP) is a closed-loop system designed to improve driveability through programmed intervention in the brake system and/or drivetrain. The integrated functionality of ABS[1] prevents the wheels from locking when the brakes are applied, while TCS[2] inhibits wheel spin during acceleration. The overall role of ESP is to prevent the vehicle's tendency to "plow" or become unstable and break away to the side, provided the vehicle remains within its physical limits.

Braking is activated on individual wheels in a targeted manner, such as the inner rear wheel to counter understeer, or the outer front wheel during oversteer, and helps to keep the vehicle's course stable under all driving conditions. ESP can also accelerate the driven wheels by specific engine-control interventions to ensure the stability of the vehicle.

[1] Antilock brake system

[2] Traction-control system

Using this *individual control* concept, the system has two options for steering the vehicle: it can brake selected wheels (selective braking) or accelerate the driven wheels.

Figure 1 shows ESP control in a schematic diagram with

- The sensors that determine the controller input parameters
- The ESP control unit with its hierarchically-structured controller, featuring a higher-level vehicle dynamics controller and the subordinate slip controllers
- The actuators used for control of braking, drive and side forces

Controller hierarchy of ESP

Higher-level vehicle dynamics controller

Function

The vehicle dynamics controller is responsible for

- Determining current vehicle status based on the yaw-rate signal and the sideslip angle estimated by the "monitor", and then

1 ESP control loop in vehicle

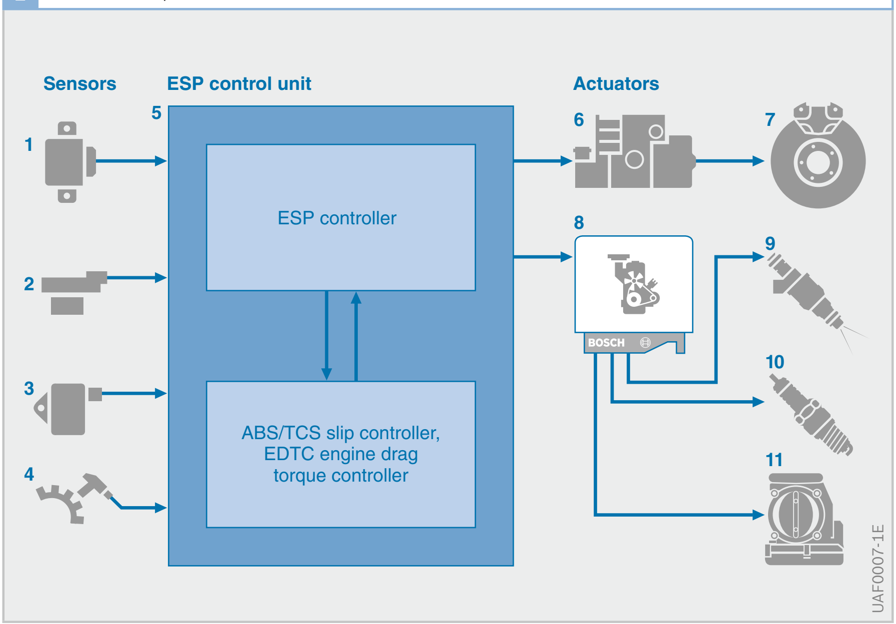


Fig. 1
1 Yaw-rate sensor with lateral-acceleration sensor
2 Steering-wheel-angle sensor
3 Brake-pressure sensor
4 Wheel-speed sensors
5 ESP control unit
6 Hydraulic modulator
7 Wheel brakes
8 Engine-management system ECU
9 Fuel injection

Only for gasoline engines:
10 Ignition-timing intervention
11 Throttle-valve intervention (ETC)

K. Reif (Ed.), *Fundamentals of Automotive and Engine Technology*,
DOI 10.1007/978-3-658-03972-1_17, © Springer Fachmedien Wiesbaden 2014

- achieving maximum possible convergence between vehicle response in the limit range and its characteristics in the normal operating range (nominal behavior).

The following components register driver commands and the system evaluates their signals as the basis for defining nominal behavior:
- Engine-management system (e.g. apply accelerator pedal)
- Brake-pressure sensor (e.g. apply brakes) or
- Steering-wheel-angle sensor (turning the steering wheel)

At this point the driver command is defined as the specified response. The coefficient of friction and the vehicle speed are also included in the calculation. The "monitor" estimates these factors based on signals transmitted by the sensors for
- Wheel speed
- Lateral acceleration
- Braking pressures and
- Yaw velocity

The desired vehicle response is brought about by generating a yaw moment acting on the vehicle. In order to generate the desired yaw moment, the system influences the tire-slip rate, and thus indirectly the longitudinal and side forces. The system influences the tire slip by varying the desired specifications for slip rate, which must then be executed by the subordinated ABS and TCS controllers.

The intervention process is designed to maintain the handling characteristics that the vehicle manufacturer intended the vehicle to have and to serve as the basis for ensuring consistently reliable control.

The vehicle dynamics controller generates the specified yaw moment by relaying corresponding slip-modulation commands to the selected wheels.

The subordinate-level ABS and TCS controllers trigger the actuators governing the brake hydraulics and the engine-management system using the data generated in the ESP controller.

Antilock brake system (ABS)

The antilock brake system (ABS) detects incipient lock on one or more wheels and makes sure that the brake pressure remains constant or is reduced. By so doing, it prevents the wheels from locking up and the vehicle remains steerable.

Wheel-speed sensors

The speed of rotation of the wheels is an important input variable for the ABS control system. Wheel-speed sensors detect the speed of rotation of the wheels and pass the electrical signals to the control unit.

A car may have three or four wheel-speed sensors depending on which version of the ABS system is fitted (ABS system versions). The speed signals are used to calculate the degree of slip between the wheels and the road surface and therefore detect incipient lock on individual wheels.

Electronic control unit

The ECU processes the information received from the sensors according to defined mathematical procedures (control algorithms). The results of these calculations form the basis for the triggering signals sent to the hydraulic modulator.

Hydraulic modulator

The hydraulic modulator incorporates a series of solenoid valves that can open or close the hydraulic circuits between the master cylinder (Fig. 2, Item 1) and the wheel-brake cylinders (4). In addition, it can connect the wheel-brake cylinders to the return pump (6). Solenoid valves with two hydraulic connections and two valve positions are used (2/2 solenoid valves).

The intake valve (7) between the master cylinder and the wheel-brake cylinder controls pressure build-up, while the exhaust valve (8) between the wheel-brake cylinder and the return pump controls pressure release. There is one such pair of solenoid valves for each wheel-brake cylinder.

Under normal conditions, the solenoid valves in the hydraulic modulator are at the "pressure build-up" setting. That means the intake valve is open. The hydraulic modulator then forms a straight-through connection between the master cylinder and the wheel-brake cylinders. Consequently, the brake pressure generated in the master cylinder when the brakes are applied is transmitted directly to the wheel-brake cylinders at each wheel.

As the degree of brake slip increases due to braking on a slippery surface or panic braking, the risk of the wheels locking up also increases. The solenoid valves are then switched to the "maintain pressure" setting. The connection between the master cylinder and the wheel-brake cylinder is shut off (intake valve is closed) so that any increase of pressure in the master cylinder does not lead to an increase in brake pressure.

If the degree of slip of any of the wheels increases further despite this action, the pressure in the wheel-brake cylinder(s) concerned must be reduced. To achieve this, the solenoid valves are switched to the "pressure release" setting. The intake valve is still closed, and in addition, the exhaust valve opens to allow the return pump integrated in the hydraulic modulator to draw brake fluid from the brake(s) concerned in a controlled manner. The brake pressure in the wheel-brake cylinder is thus reduced so that wheel lock-up does not occur.

ABS control loop

Overview

The ABS control loop (Fig. 3) consists of the following:

Fig. 2
1 Master cylinder with expansion tank
2 Brake booster
3 Brake pedal
4 Wheel brake with wheel-brake cylinder

Hydraulic modulator with
5 Damping chamber
6 Return pump
7 Intake valve
8 Exhaust valve
9 Brake-fluid accumulator

Intake valve: shown in open setting
Exhaust valve: shown in closed setting

2 Principle of hydraulic modulator with 2/2 solenoid valves (schematic)

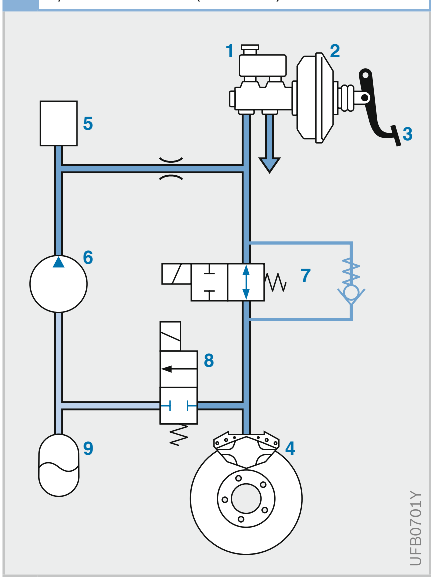
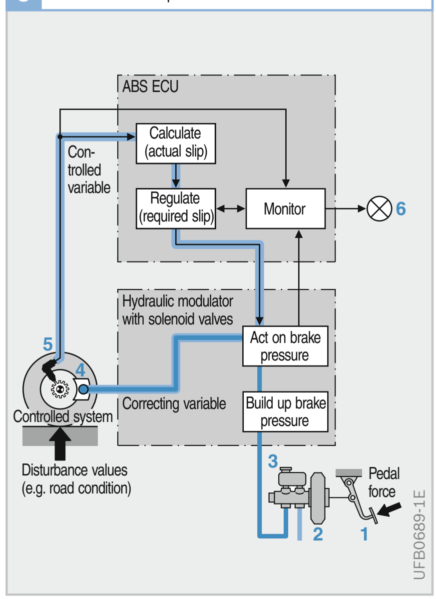


Fig. 3
1 Brake pedal
2 Brake booster
3 Master cylinder with expansion tank
4 Wheel-brake cylinder
5 Wheel-speed sensor
6 Warning lamp

3 ABS control loop



Controlled system
- The vehicle and its wheel brakes
- The wheels and the friction pairing of tire and road surface

The disturbance values affecting the control loop
- Changes in the frictional connection between the tires and the road surface caused by different types of road surface and changes in wheel load, e.g. when cornering
- Irregularities in the road surface causing the wheels and axles to vibrate
- Lack of circularity of the tires, low tire pressure, worn tire tread, differences in circumference between wheels, (e.g. spare wheel)
- Brake hysteresis and fade
- Differences in master-cylinder pressure between the two brake circuits

Controller
- The wheel-speed sensors
- The ABS control unit

Controlled variables
- Wheel speed and, derived from it, wheel deceleration
- Wheel acceleration and brake slip

The reference variable
- The foot pressure applied to the brake pedal by the driver – amplified by the brake booster – generates the brake pressure in the brake system

The correcting variable
- Brake pressure in the wheel-brake cylinder

Controlled system

The data-processing operations performed by the ABS control unit are based on the following simplified controlled system:
- A non-driven wheel
- A quarter of the vehicle's mass apportioned to that wheel
- Wheel brake
- An idealized coefficient of friction slip curve (substitute for the friction pairing of tire and road)

That curve is divided into a stable zone with a linear gradient and an unstable zone with a constant progression (μ_{HFmax}).

As an additional simplification, there is also an assumed initial straight-line braking response that is equivalent to a panic-braking reaction.

Figure 4 shows the relationships between brake torque M_B (the torque that can be generated by the brake through the tire), or road frictional torque M_R (torque that acts against the wheel through the friction pairing of tire and road surface), and time t, as well as the relationships between the wheel deceleration ($-a$) and time t, whereby the brake torque increases in linear fashion over time. The road frictional torque lags slightly behind the brake torque by the time delay T, as long as the braking sequence is within the stable zone

4 Initial braking phase, simplified

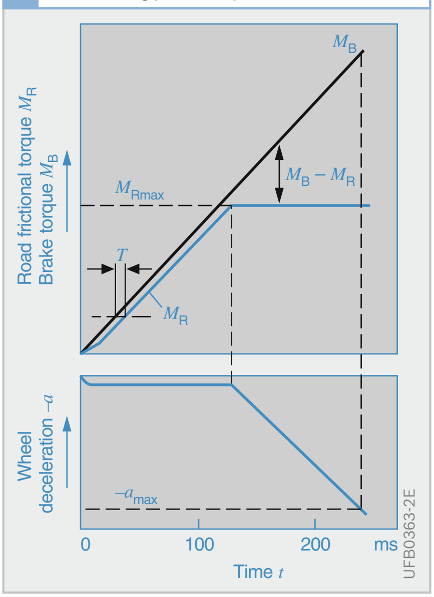


Fig. 4
($-a$) Wheel deceleration
($-a_{max}$) Maximum wheel deceleration
M_B Brake torque
M_R Road frictional torque
M_{Rmax} Maximum road frictional torque
T Time delay

of the curve for coefficient of friction versus brake slip. After about 130 ms, the maximum level (μ_{HFmax}) – and therefore the unstable zone – of the curve for coefficient of friction versus brake slip is reached. From that point on, the curve for coefficient of friction versus brake slip states that while the brake torque M_B, continues to rise at an undiminished rate, the road frictional torque M_R, cannot increase any further and remains constant. In the period between 130 and 240 ms (this is when the wheel locks up), the minimal torque difference $M_B - M_R$, that was present in the stable zone rises rapidly to a high figure. That torque difference is a precise measure of the wheel deceleration ($-a$) of the braked wheel (Fig. 4, bottom). In the stable zone, the wheel deceleration is limited to a small rate, whereas in the unstable zone it increases rapidly, according to the amount. As a consequence, the curve for coefficient of friction versus wheel slip reveals opposite characteristics in the stable and unstable zones. The ABS exploits these opposing characteristics.

Controlled variables

An essential factor in determining the effectiveness of an ABS control system is the choice of controlled variables. The basis for that choice is the wheel-speed sensor signals from which the ECU calculates the deceleration/acceleration of the wheel, brake slip, the reference speed and the vehicle deceleration. On their own, neither the wheel deceleration/acceleration nor the brake slip are suitable as controlled variables because, under braking, a driven wheel behaves entirely differently to a non-driven wheel. However, by combining those variables on the basis of appropriate logical relationships, good results can be obtained.

As brake slip is not directly measurable, the ECU calculates a quantity that approximates to it. The basis for the calculation is the reference speed, which represents the speed under ideal braking conditions (optimum degree of brake slip). So that speed can be determined, the wheel-speed sensors continuously transmit signals to the ECU for calculating the speed of the wheels. The ECU takes the signals from a pair of diagonally opposed wheels (e.g. right front and left rear) and calculates the reference speed from them. Under partial braking, the faster of the two diagonally opposite wheels generally determines the reference speed. If the ABS cuts in under emergency braking, the wheel speeds will be different from the vehicle speed and can thus not be used for calculating the reference speed without adjustment. During the ABS control sequence, the ECU provides the reference speed based on the speed at the start of the control sequence and reduces it at a linear rate. The gradient of the reference-speed graph is determined by analyzing logical signals and relationships.

If, in addition to the wheel acceleration/deceleration and the brake slip, the vehicle deceleration is brought into the equation as an additional quantity, and if the logical circuit in the ECU is modulated by computation results, then ideal brake control can be achieved. This concept has been realized in the Bosch antilock brake system (ABS).

Traction-control system (TCS)

The antilock brake system (ABS) prevents the wheel lock when the brakes are applied by lowering the wheel brake pressures. The traction-control system (TCS) prevents wheel spin by reducing the drive torque at each driven wheel.

In addition to this safety-relevant task of ensuring the stability and steerability of the vehicle when accelerating, TCS also improves the traction of the vehicle by regulating the optimum slip. The upper limit here is, of course, set by the traction requirement stipulated by the driver.

The TCS regulates the slip of the driven wheels as quickly as possible to the optimum level. To do this the system first determines a setpoint value for the slip. This value depends on a number of factors that represent the current driving situation. These factors include:

- The basic characteristic for TCS reference slip (based on the slip requirement of a tire during acceleration)
- Effective coefficient of friction
- External tractive resistance (deep snow, rough road, etc.)
- Yaw velocity, lateral acceleration, and steering angle of the vehicle

TCS interventions

The measured wheel speeds and the respective drive slip can be influenced by changing the torque balance M_{tot} at each drive wheel. The torque balance M_{tot} at each driven wheel results from the drive torque $M_{Kar}/2$ at this wheel, the respective brake torque M_{Br} and the road torque M_{Str} (Fig. 5).

$M_{tot} = M_{Kar}/2 + \mathrm{M}_{Br} + M_{Str}$
(M_{Br} and M_{Str} are negative here.)

This balance can obviously by influenced by the drive torque M_{Kar} provided by the engine as well as by the brake torque M_{Br}.

Both these parameters are therefore correcting variables of the TCS which can be used to regulate the slip at each wheel to the reference slip level.

In gasoline-engine vehicles, the drive torque M_{Kar} can be controlled using the following engine-control interventions:

- Throttle valve (throttle valve adjustment)
- Ignition system (ignition-timing advance)
- Fuel-injection system (phasing out individual injection pulses)

In diesel-engine vehicles, the drive torque M_{Kar} is influenced by the electronic diesel control system (EDC) (reduction in the injected fuel quantity).

The brake torque M_{Br} can be regulated for each wheel via the brake system. The TCS function requires the original ABS hydraulic system to be expanded because of the need for active pressure build-up.

5 Drive concept of a single-axle driven vehicle with TCS

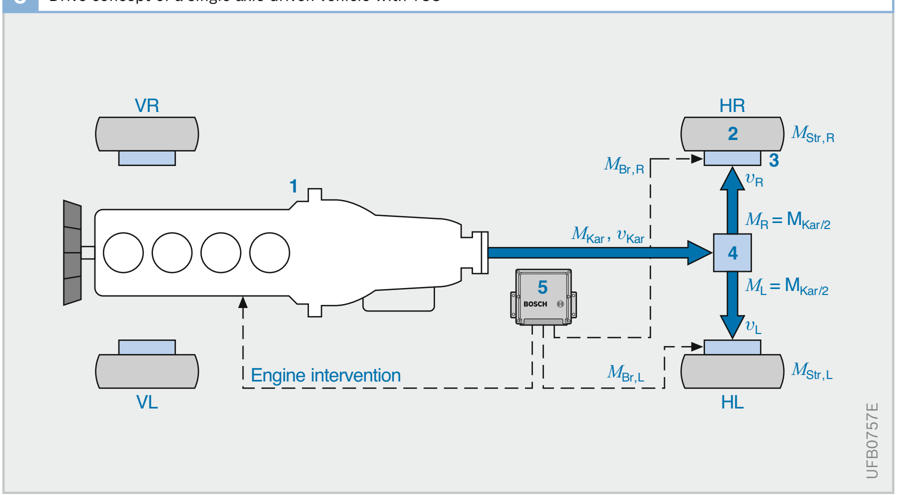


Fig. 5
1 Engine with transmission
2 Wheel
3 Wheel brake
4 Transversal differential
5 Electronic control unit with TCS functionality

Engine, transmission, gear ratio of differential and losses are combined in one unit.

M_{Kar} Drive axle torque
v_{Kar} Cardan speed
M_{Br} Brake torque
M_{Str} Torque transferred to the road
v Wheel speed
R Right
L Left
V Front
H Rear

ABS versions

Evolution of the ABS versions

Technological advances in the areas of

- Solenoid-valve design and manufacturing.
- Assembly and component integration.
- Electronic circuitry (discrete components replaced by hybrid and integrated circuits with microcontrollers).
- Testing methods and equipment (separate testing of electronic and hydraulic systems before integration in the hydraulic modulator).
- Sensor and relay technology have enabled the weight and dimensions of ABS versions to be more than halved since the first-generation ABS2 in 1978. As a result, modern systems can now be accommodated even in vehicles with the tightest space restrictions. Those advances have also lowered the cost of ABS systems to the extent that they are now fitted as standard on all types of vehicle.

1 Evolution of ABS configurations

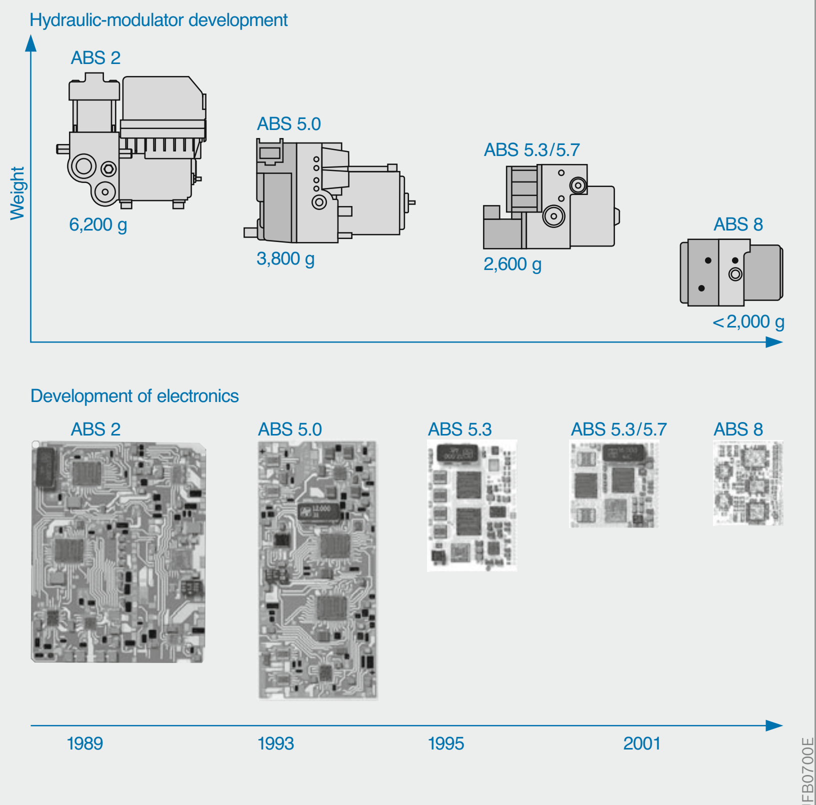


Fig. 1
Historical development of ABS showing technological advances: Decreasing weight accompanied by increasing processing power.

History of radar

Technology to mimic the animals

Radar (RAdiation Detecting And Ranging) is a wireless method for locating objects that has traditionally been used mainly in air travel and shipping. Since radar-supported air defence was introduced in World War 2, radar has also been a feature of weapons technology. More recently, radar has found application in space travel, weather forecasting and ultimately in road traffic as part of measuring distances between vehicles using ACC (Adaptive Cruise Control).

The basis for the development of radar was the sonar system (Sound Navigation and Ranging) possessed by animals for navigation and determining distances. Echolocating bats, for example, emit orientation sounds in the form of shrill whistles in the 30 to 120 kHz *ultrasound range*. They then pick up the echo from obstacles or their prey with their ears. They use the information contained in the echo to decide what to do next.

Radar functions in a similar way but uses *radio signals* not sound waves. The distance measurement method of radar is based on the measurement of the time that elapses between the emission of the *electromagnetic waves* and the reception of a signal echo reflected by an object.

While air travel and shipping radar systems, for example, operate in the 500 MHz to 40 GHz frequency range, ACC is authorized to use the 76 to 77 GHz frequency band.

Stages in the development of radar

The development of electromagnetic, long-range search facilities was a great challenge for design engineers. Only a small part of the energy originally emitted was reflected back. It is therefore necessary to emit a large amount of energy and to concentrate it in as narrow a beam bundle as possible. The only devices suitable for this are very sensitive transmitters and receivers for waves that are shorter than the dimensions of the target.

The development path that led to radar technology is punctuated by the following historical milestones and characters:

1837 **Morse:** Message transmission over long distances by means of electrical pulses sent by the telegraph was the first method to find widespread use

1861/1876 **Reis** and **Bell:** Replacement of telegraphs by the telephone makes message transfer much more direct and user-friendly

1864 **Maxwell, Hertz** and **Marconi:** Existence of “radio waves” proven theoretically and experimentally. Radio waves reflect off metallic objects in the same way as light waves reflect off a mirror

1922 **Marconi:** The pioneer of the radio proposes further investigations into earlier research approaches to radio measuring technology

1925 **Appleton** and **Barnett:** The principle of radio measuring technology is used to detect conducting strata in the atmosphere

Breit and **Tuve:** Development of pulse modulation that permits exact development measurements

1935 **Watson Watt:** Invention of the radar

1938 **Ponte:** Invention of the magnetron (velocity-modulated tube for generating high-frequency vibrations)

Adaptive cruise control

System description

Like the basic cruise-control system that has been available as a standard feature for many years, ACC (Adaptive Cruise Control) can be categorized as a driver-assistance system. Cruise control regulates driving speed to maintain the desired speed selected by the driver using the cruise-control unit. In addition to the basic cruise-control function, ACC measures the distance to the vehicle in front and its relative speed, and uses this information together with other collected data (position of other vehicles in the same or different lane; in future, even stationary objects) to regulate the time gap between the vehicles. ACC is thus able to adapt the vehicle's speed to match the speed of the vehicle traveling in front and maintain a safe distance from it. The driver is able to override or switch off the ACC function at any time (e.g. by depressing the gas or brake pedal).

Distance sensor

In the main, ACC systems currently have a radar sensor (Fig. 1, Item 1) that operates in a frequency range of between 76 and 77 GHz and emits four radar lobes. Once activated, ACC detects a range of up to approximately 200 m in front of the vehicle.

The radar beams reflected by vehicles in front are analyzed for timing, Doppler shift and amplitude ratio. These factors are used to calculate distance, relative speed and angle position relative to vehicles in front.

Network architecture

The ACC function cannot be represented independently as a stand-alone system; various subsystems (engine-management system, electronic stability program, transmission control, instrument cluster) must be networked with each other. The evaluation and control electronics (control unit) of the ACC are integrated in the sensor housing. They receive and send data on a CAN data bus from and to other electronic control units.

Course setting

To ensure reliable ACC operation no matter what the situation – e.g. also on bends – it is essential that the preceding vehicles can be allocated to the correct lane(s). For this purpose, the information from the ESP sensor system (yaw rate, steering angle, wheel speeds and lateral acceleration) is evaluated with regard to the ACC-equipped vehicle's own curve status.

1 Adaptive cruise control (ACC)

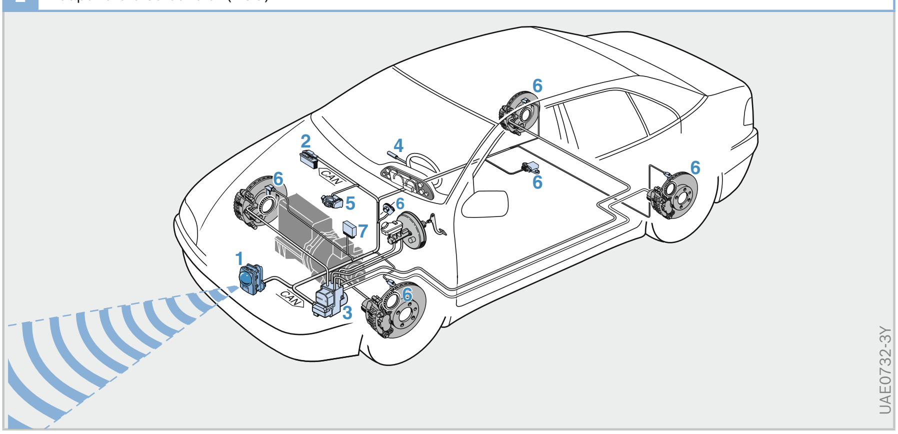


Fig. 1
1 ACC sensor and control unit
2 Engine-management system ECU (ME or DI Motronic) for gasoline engines or electronic diesel control (EDC) for diesel engines
3 Active brake intervention via ESP
4 Control and display unit
5 Engine-control intervention by means of electrically adjustable throttle valve (ME or DI Motronic)
6 Sensors
7 Transmission-shift control by means of electronic transmission control (optional)

K. Reif (Ed.), *Fundamentals of Automotive and Engine Technology*,
DOI 10.1007/978-3-658-03972-1_18, © Springer Fachmedien Wiesbaden 2014

Setting options

The driver inputs the desired speed and the desired time gap; the time gap available to the driver usually ranges from 1 to 2 s. The time gap to the vehicle in front is calculated from the radar signals and compared with the desired time gap specified by the driver. If this value is shorter than the desired value, the ACC system responds in a manner appropriate to the traffic situation by initially reducing engine torque, and only if necessary by automatically braking the vehicle. If the desired time gap is exceeded, the vehicle accelerates until either the speed of the vehicle in front or the desired speed set by the driver is reached.

Engine-control intervention

Speed control requires an electronic engine-performance control system. The ME or DI Motronic engine-management system and the electronic diesel control (EDC) are integrated with this function. This system allows the vehicle to be accelerated to the desired speed or, if an obstacle appears, to be decelerated by means of automatic throttle closing.

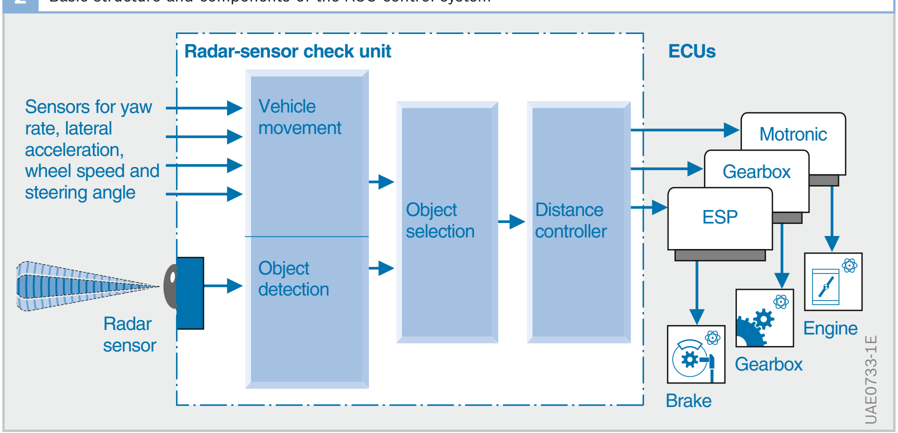
Brake intervention

If the rate of deceleration achieved by easing off the gas is not sufficient, the vehicle will have to be braked. The electronic stability program (ESP) is required here as it is able to initiate the brake intervention.

Due to the design of ACC as a comfort system, the braking deceleration calculated by the ACC controller is limited to approximately 2 to 3 m/s² with current ACC systems. If this is not sufficient for the current traffic situation (e.g. if the vehicle in front brakes sharply), the vehicle audibly requests the driver to take over responsibility for braking. The necessary braking deceleration is then to be achieved using the service brake. Safety functions, such as panic braking, are not part of the ACC system.

The other stabilizing systems of ABS, TCS or ESP may be active as normal during an active ACC control intervention as necessary. Depending on the parameter settings of ACC, stabilization interventions may result in ACC being deactivated.

2 Basic structure and components of the ACC control system



Control and display

Controls include switches, push-buttons or thumbwheels for

- Activating the function and
- Setting the desired speed and
- Desired time gap

The following information may be displayed to the driver in the instrument cluster:

- Desired speed
- Information on the activation status
- The time gap selected by the driver
- Indication of the follow-up mode, which informs the driver as to whether the system is controlling the distance to a detected target object or not

System limits

ACC does not yet permit control operations in city environments. This system can only be activated at speeds in excess of 30 km/h.

Control algorithms

The control system basically consists of three control modules:

- *Control module 1: cruise control*
  If the radar sensor has not detected any vehicles in front, the system maintains the desired speed set by the driver.
- *Control module 2: follow-up control*
  The radar sensor has detected vehicles in front. Control essentially maintains the time gap to the nearest vehicle at a constant setting.
- *Control module 3: control when cornering*
  When negotiating tight bends, the radar sensor can "lose sight" of the vehicle in front because of the limited width of its "field of vision". Until the vehicle comes in sight of the radar again, or until the system is switched to normal cruise control, special measures come into effect. Depending on the manufacturer, the speed would then be maintained, the current rate of lateral acceleration adapted or the ACC function deactivated, for example.

Object detection and lane allocation

The central task of the radar sensor and its integrated electronics is to detect objects and allocate them either to the same lane as the one on which the vehicle is traveling, or to a different lane. Firstly, lane allocation demands the precise detection of vehicles in front (high angle resolution and accuracy), and secondly, a precise knowledge of the motion of the system's own vehicle. Vehicle motion is calculated from the signals sent by sensors also used for the electronic stability program (ESP) (course prediction). These include the wheel-speed sensors and driving-dynamics sensors for the yaw rate and lateral acceleration. Optionally, information supplied by a steering-angle sensor may also be processed. The decision as to which of the detected objects is used as the reference for adaptive cruise control is essentially based on a comparison between the positions and motion of the detected objects and the motion of the system's own vehicle.

Adjustment

The radar sensor is fitted at the front end of the vehicle. Its radar lobes are aligned relative to the vehicle longitudinal axis. This is done using adjusting screws at the fastening part of the sensor. If it is moved out of alignment by physical force, e.g. deformation of the mounting due to accident damage or any other effect, realignment must be carried out. Small degrees of misalignment are automatically corrected by the permanently active alignment routines implemented in the software. If manual realignment is required, this is indicated to the driver.

Ranging radar

The radar (RAdiation Detecting And Ranging) transceiver unit transmits packets of electromagnetic waves using an antenna. These reflect off an object made of electrically conductive materials (e.g. vehicle body) and are then received. The signals received in this manner are "compared" with the transmitted signals with respect to their propagation time and/or frequency.

Measuring principles

Propagation time measurement

For all radar methods, the distance measurement is based on the direct or indirect propagation time measurement for the time between when the radar signal is transmitted and when the signal echo is received. The direct propagation time measurement is used to measure period τ. With direct reflection, this is equal to twice the distance d to the reflector divided by the speed of light c:

$$\tau = 2d/c$$

For a distance of d = 150 m and $c \approx$ 300,000 km/s, the propagation time is

$$\tau \approx 1\ \mu s.$$

Frequency modulation

Direct propagation time measurement requires much effort; an indirect propagation time measurement is simpler. The method is known as FMCW (Frequency Modulated Continuous Wave). Rather than comparing the times between the transmitted signal and received echo, the FMCW radar compares the frequencies of the transmitted signal and received echo. The prerequisite for a meaningful measurement is a modulated transmit frequency.

With the FMCW method, radar waves linearly modulated in their frequency are transmitted for a duration of typically a few milliseconds and in a cycle of a few hundred MHz (f_s, continuous curve in Fig. 3). The signal reflected off a vehicle in front is delayed in accordance with the signal propagation time (f_e, dashed line in Fig. 3). In the rising ramp, the frequency is lower; in the falling ramp, the frequency is higher by the same amount. The difference in frequency Δf is a direct measure for the distance.

If there is additionally a relative speed between the vehicles, the receive frequency f_e is increased (f_e', dotted line in Fig. 3) by a specific amount Δf_d in both the rising and falling ramp due to the Doppler effect. This produces two different frequency differences Δf_1 and Δf_2. Their addition produces the distance between the vehicles, and their subtraction the relative speed of the vehicles.

The signal processing in the frequency range therefore delivers a frequency for each object that as a linear combination produces a term for distance and relative speed. From the measured frequencies of two ramps with a different gradient, it is possible to determine the distance to and the relative speed of an object. For situations involving several targets, several ramps with a different gradient are required.

3 FMCW method

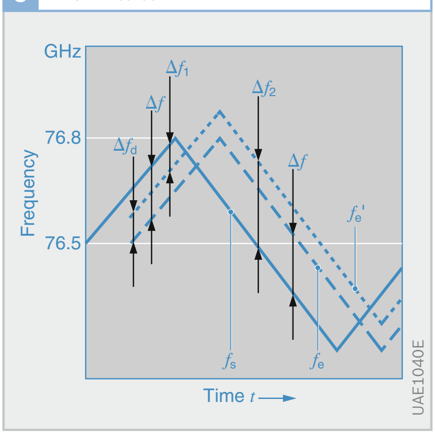


Fig. 3
f_s Transmission signal
f_e Return signal at same road speed
f_e' Return signal with relative road speed

Doppler effect
Although the relative speed of the other vehicle can be measured using a number of subsequent distance measurements, it is calculated more quickly, more reliably and more accurately when the Doppler effect is utilized in the measurement.

For an object moving relative to the radar sensor with a relative speed (relative speed v_{rel}), the signal echo undergoes a frequency shift f_D compared to the emitted signal. At the relevant differential speeds, this is represented as:

$$f_D = -2 f_C \cdot v_{rel}/c$$

f_C is the carrier frequency of the signal. At the radar frequencies commonly used for ACC, $f_C = 76.5$ GHz, there is a frequency shift of $f_D \approx -510 \cdot v_{rel}$/m, and thus 510 Hz at a relative speed of -1 m/s (approaching).

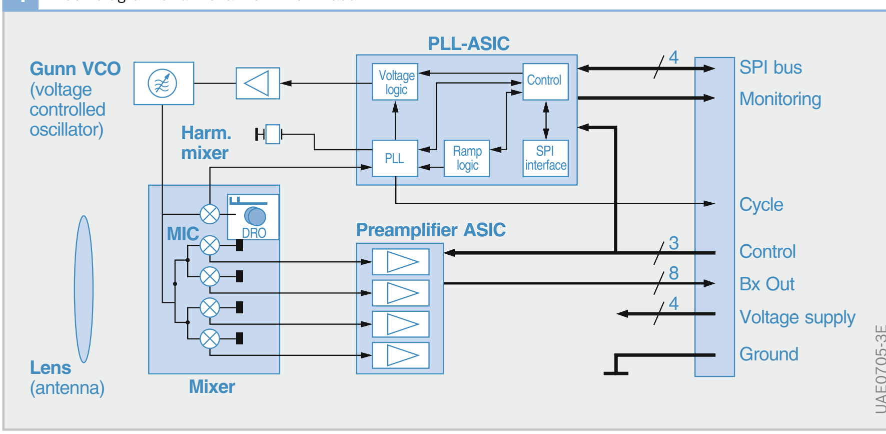
Angle measurement
The third basic dimension which is needed is the side offset (angle) of the preceding vehicle. The only way this can be measured is by radiating the radar beam in a number of different directions. The (reflected) signals are then applied to determine from which direction the strongest reflection came. This method needs either high-speed back-and-forth movement of the beam (scanning), or the installation of a multi-beam antenna array.

High-frequency part of the ACC sensor
The high-frequency part can be broken down into four functional groups.

HF generation
The HF generation and control section makes the high frequency available for transmission (Fig. 4). The HF output of between 76 and 77 GHz is generated using a voltage-controlled oscillator (VCO) comprising a Gunn diode in a mechanical resonator. A small part of the output generated is downmixed into an intermediate frequency band using a dielectrical resonance oscillator (DRO) with harmonic mixer and supplied to the control electronics (PLL-ASIC, PLL = Phase Locked Loop). The latter controls the VCO by means of an output driver and provides frequency stabilization and modulation.

Transmission and reception circuit
In the transmission and reception circuit, the HF output is divided between four transmission/reception channels by three Wilkinson splitters. Using bandpass mixers, this output is supplied to the antenna while the return signal is downmixed to the basic band.

4 Block diagram of a 4-channel FMCW radar



Amplification
Signals in the basic band are amplified in an ASIC. It has four channels, switching amplification, and a special characteristic curve. To some extent, this compensates for the large signal dynamics in that high frequencies (correspondingly high distances) are more strongly amplified. In addition, the characteristic is integrated with a low-pass anti-aliasing filter for subsequent scanning.

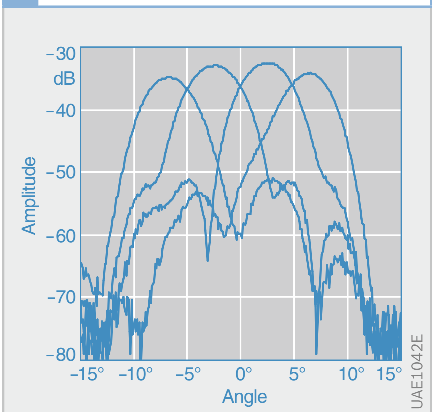
Antenna system
The antenna system is a monostatic system. It comprises four combined transmitter and receiver patches on the HF substrate, four polyrods (plastic cones) for prefocusing and a plastic lens for beam concentration. As part of the housing, the lens also acts as a radar-optical window and shield. The radar waves are emitted simultaneously and coherently by the four antenna patches to produce a single transmission wave. The beams are only actually split into four separate beams on the receiver side. Four separate receiver channels are used here.

The transmit frequency is modulated by the voltage-controlled oscillator (VCO) in a linearly ramped fashion (Fig. 3) with a gradient of $m = \mathrm{d}f/\mathrm{d}t$. While the received signal returns after the propagation time $\tau = 2d/c$ the transmit frequency has changed in the meantime by the differential frequency $f_\mathrm{D} = \tau \cdot m$. Therefore, the propagation time, and thus the distance, can be measured indirectly by ascertaining the differential frequency between the transmitted and received signals. The differential frequency, in turn, can be ascertained using a mixer, followed by low-pass filtering. To determine the frequency, the signal is digitized and converted into a frequency spectrum using an FFT (Fast Fourier Transformation).

However, the information about the difference frequency does not only contain information for the propagation time but also for the Doppler shift. This situation means that there is at first a certain ambiguity in the evaluation. It can be resolved by applying multiple FMCW modulation cycles with different gradients.

To determine the angle at which the radar locates an object, multiple radar lobes are transmitted and evaluated. At least two overlapping radar beams are required to measure the angle. No conclusions on the angle of sight can be drawn from the relationships between the amplitudes that are measured for an object in adjacent beams. If, for example, four radar beams are used, the horizontal angular dependence of which is shown in Figure 5 in the form of a two-way antenna diagram as an example, it is possible to determine the horizontal angle of sight by comparing amplitude and phase of the measured radar signals using the antenna diagram.

5 Two-way antenna diagram



Radar signal processing

The low-frequency part of the FMCW radar comprises several components (Fig. 6).

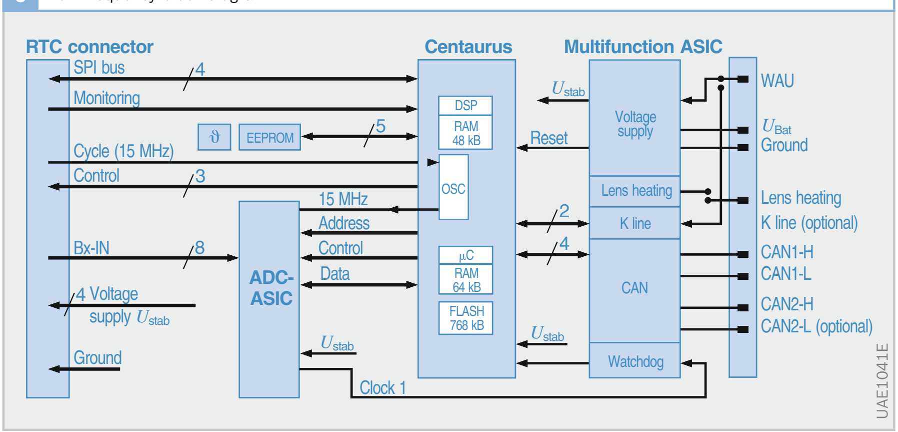
A dual-core processor is used for digital data processing. The digital signal processor (DSP) contained in this module is used for data acquisition, calculating the fast fourier transform (FFT) and for other basic signal processing. The processor also contains a microcontroller in which additional signal processing, the application software and control unit functions are executed. Furthermore, various peripherals are integrated in the dual processor: serial interfaces, two CAN controllers (Controller Area Network), an analog/digital converter and various digital ports.

The processor is surrounded by various peripheral modules. The analog radar signals from the HF PCB are converted into digital sample values in an analog/digital converter (ADC). This takes place simultaneously for four channels. A digital low-pass filter is also integrated in this module to ensure limitation to the Nyquist bandwidth. An EEPROM is used as an external, non-volatile memory. Application parameters and, if applicable, fault codes are stored here. A multifunction ASIC is used to generate the supply voltages (different DC voltages) and as an output driver (K line, CAN, lens heating to prevent icing). In addition, a watchdog is also integrated. Using a temperature sensor, it is possible to measure the internal temperature of the system.

The unit is connected to the vehicle by an eight-pin connector. The connector supplies battery voltage (approximately 12 V), ground (GND), two CAN buses, or alternatively a wakeup or K line, or a radome heater, or a time gap signal.

The low-frequency circuit can be designed using standard printed-circuit board technology. Figure 7 provides a look inside the unit.

6 Low-frequency block diagram



7 Exploded view of an FMCW radar with integrated signal processing

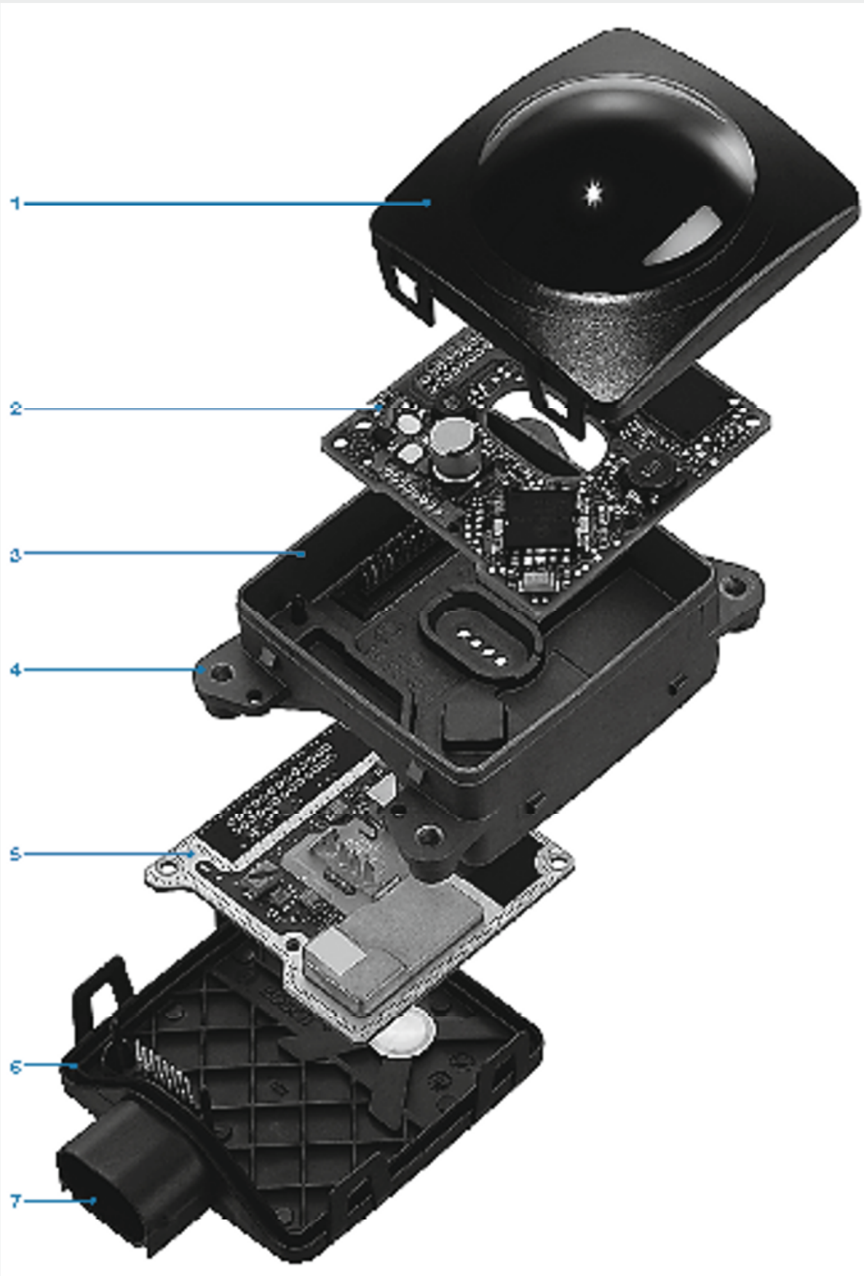


Fig. 7
1 Upper housing section with lens
2 HF PCB
3 Bearing points for alignment
4 Intermediate carrier
5 LF PCB
6 Housing base
7 Plug

Occupant-protection systems

In the event of an accident, occupant-protection systems are intended to keep the accelerations and forces that act on the passengers low and lessen the consequences of the accident. These passive vehicle safety systems include:

- Seat belts with seat-belt pretensioners
- Airbags and
- Rollover protection systems (on cabriolets)

Seat belts and seat-belt pretensioners provide the greater part of the protective effect since they absorb 50 to 60 % of impact energy. With front airbags, the energy absorption is about 70 % if deployment timing is properly synchronized.

In order to achieve optimum protection, the response of all components of the complete occupant-protection system must be adapted to one another.

Seat belts and seat-belt pretensioners

Function

The function of seat belts is to restrain the occupants of a vehicle in their seats when the vehicle impacts against an obstacle. In the event of a frontal impact, seat-belt pretensioners pull the seat belts tighter against the body and hold the upper body as closely as possible against the seat backrest. This prevents unimpeded forward displacement of the occupants caused by inertia. Seat-belt pretensioners thus improve the restraining characteristics of a three-point inertia-reel belt and increase protection against injury.

Operating principle

In an impact, the shoulder-belt tightener eliminates the seat belt slack and the “film-reel effect” by rolling up and tightening the belt webbing. On activation, the system electrically fires a pyrotechnic propellant charge. The rising pressure acts on a piston,

1 Electronic impact protection system

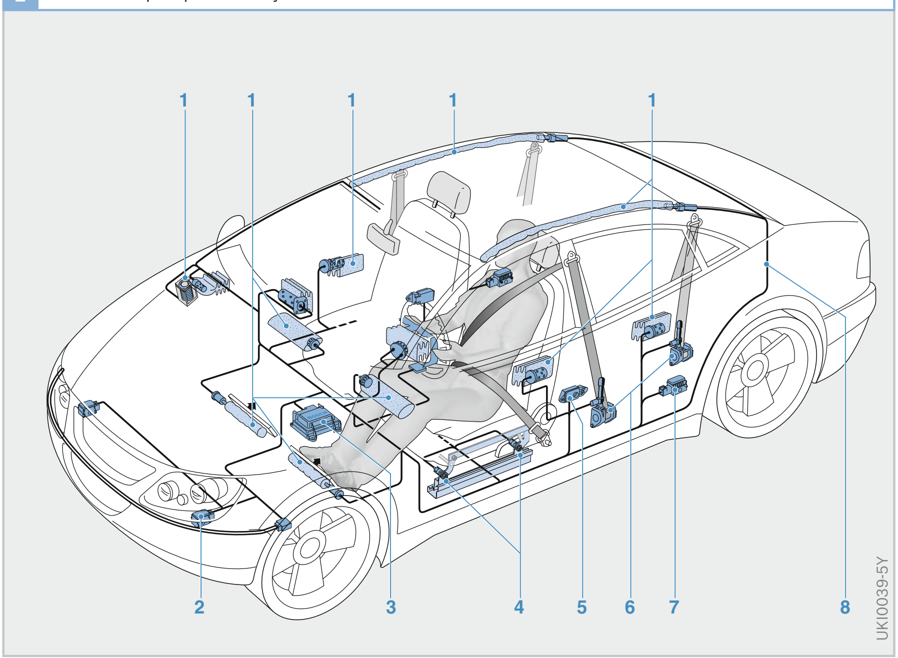


Fig. 1
1 Airbag with gas inflator
2 Upfront sensor
3 Central electronic control unit with integrated rollover sensor
4 iBolt™
5 Peripheral pressure sensor (PPS)
6 Seat-belt pretensioner with propellant charge
7 Peripheral acceleration sensor (PAS)
8 Bus architecture (CAN)

K. Reif (Ed.), *Fundamentals of Automotive and Engine Technology*,
DOI 10.1007/978-3-658-03972-1_19, © Springer Fachmedien Wiesbaden 2014

which turns the belt reel via a steel cable in such a way that the belt is held tightly against the body (Fig. 2).

Variants

In addition to the shoulder-belt tighteners, there are variants which pull the seat-belt buckle back (buckle tighteners) and simultaneously tighten the shoulder and lap belts. Buckle tighteners further improve the restraining effect and the protection to prevent occupants from sliding forward under the lap belt ("submarining effect").

A further improvement is achieved by the use of belt-force limiters. In this case, the seat belt tighteners initially tighten fully (using the maximum force of approx. 4 kN, for example) and restrain the occupants. If a certain belt tension is exceeded, the belt gives thereby extending the degree of forward movement. The kinetic energy is converted into deformation energy, which prevents the occurrence of acceleration peaks. Deformation elements include a torsion rod in the inertia reel shaft. However, there is also an electronically controlled single-stage belt-force limiter, which reduces the belt tension to 1 to 2 kN by firing a detonator a specific period after deployment of the second front airbag stage and after a specific extent of forward movement is reached.

2 Shoulder-belt tightener

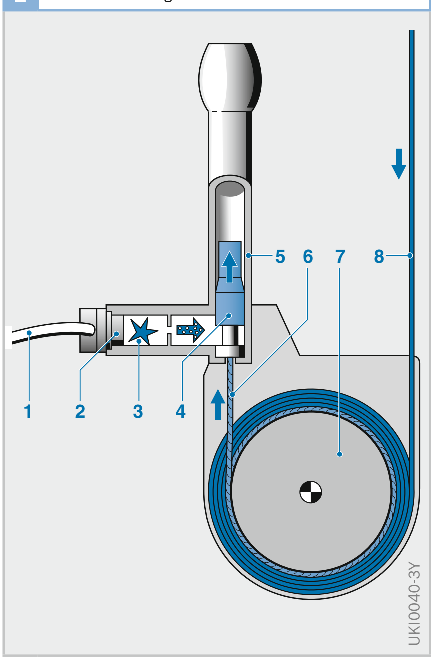


Fig. 2
1 Firing cable
2 Firing element
3 Propellant charge
4 Piston
5 Cylinder
6 Wire rope
7 Belt reel
8 Belt webbing

Front airbag

Function

The function of front airbags is to protect the driver and front passenger against head and chest injuries in a vehicle impact with an obstacle. In a serious accident, a seat-belt pretensioner cannot keep the head from striking the steering wheel.

Operating principle

To protect driver and front passenger, pyrotechnical gas inflators inflate the driver and passenger airbags highly dynamically after a vehicle impact detected by sensors. In order to provide maximum protection, the airbag must be fully inflated before the occupant plunges into it. Once the occupant makes contact with it, the airbag partly deflates in order to "gently" absorb impact energy acting on the occupant with noncritical (in terms of injury) surface pressures and declaration forces.

The maximum permissible forward displacement of the driver before the airbag on the driver's side has inflated is approximately 12.5 cm. This equates to a time of approximately 40 ms from the start of impact (in the case of an impact with a hard obstacle at 50 km/h). 10 ms elapse before the electronics detect the impact and trigger the electronic ignition system. A further 30 ms is required for the airbag to inflate. The airbag is deflated through the outlet openings after another 80 to 100 ms. The entire process takes little more than a tenth of a second.

Impact detection

The deceleration arising from the impact is detected by one (or two) longitudinal acceleration sensor(s) and the change in speed is calculated from it. In order to be able to better detect oblique and offset impacts, the deployment algorithm can also evaluate the signal from the lateral acceleration sensor.

The impact must also be analyzed in addition to the crash sensing. The airbag should not trigger from a hammer blow in the workshop, gentle impacts, bottoming out, driving over curbstones or potholes. With this goal in mind, the sensor signals are processed in digital analysis algorithms whose sensitivity parameters have been optimized with the aid of crash data simulations. Depending on the vehicle manufacturer's production concept, the deployment parameters and the vehicle's equipment level can also be programmed into the ECU at the end of the assembly line ("end-of-line programming").

In order to prevent airbag-related injuries to "out-of-position" occupants (e.g. leaning too far forward) or to small children in reboard (rearward-facing) child seats, it is essential that the front airbags be triggered and inflated in accordance with the particular situations. The following measures are implemented for this purpose:

Deactivation switch
This can be used to disable the passenger airbag. The status of the airbag function is indicated by lamps.

Intelligent airbag systems
The introduction of improved sensing functions and control options for the airbag inflation process, with the accompanying improvement in protective effect, is intended to result in a gradual reduction in the risk of injury. Such functional improvements are:

- Impact severity detection through further optimization of the deployment algorithm or the use of one or two upfront sensors (Fig. 4). The latter are acceleration sensors installed in the vehicle's crumple zone (e.g. on the radiator crossmember) which facilitate early detection of and distinction between different types of impact, such as ODB (Offset Deformable Barrier) crashes, pole or underride impacts. They also allow an assessment of the impact energy.
- Seltbelt usage detection.
- Occupant presence, position and weight detection.
- Seat position and backrest inclination detection.
- Use of front airbags with two-stage gas inflators or with single-stage gas inflators and pyrotechnically triggered gas-discharge valves.
- Use of seat-belt pretensioners with occupant-weight-dependent belt-force limiters.
- Through data exchange with other systems, e.g. ESP (Electronic Stability Program), and environment sensors, it is possible to use information from the phase shortly before the impact to further optimize the deployment of the restraints.

Side airbag

Function

Side airbags, which inflate along the length of the roof lining for head protection (inflatable tubular systems, window bags, inflatable curtains) or from the door or seat backrest (thorax bags, upper body protection) are designed to cushion the occupants and protect them from injury in the event of a side impact.

Operating principle

Due to the lack of a crumple zone, and the minimum distance between the occupants and the vehicle's side structural components, it is particularly difficult for side airbags to inflate in time. For this reason, the time required for crash sensing and activating of the side airbags is approximately 5 to 10 ms for hard side impacts. The inflation time of the approximately 12 l capacity thorax bag is not permitted to be more than 10 ms.

These requirements can be fulfilled through evaluation of peripheral (at suitable points on the body, e.g. b-pillar or door), lateral (sideways) acceleration and pressure sensors.

Rollover protection systems

Function

In the event of an accident where the vehicle rolls over, open-top vehicles such as convertibles lack the protecting and supporting roof structure of closed-top vehicles. Extendable rollover bars or the extendable head restraints provide protection against injury for occupants.

Operating principle

Current sensing concepts no longer trigger the system at a fixed threshold but rather at a threshold that conforms to a situation and only for the most common rollover situation, i.e. about the longitudinal axis. The Bosch sensing concept involves a surface micromechanical yaw-rate sensor and high-resolution acceleration sensors in the vehicle's transverse and vertical axes (y and z axes).

The yaw-rate sensor is the main sensor, while the y and z-axis acceleration sensors are used both to check plausibility and to detect the type of rollover (slope, gradient, curb impact or "soil-trip" rollover). On Bosch systems, these sensors are incorporated in the airbag triggering unit.

Deployment of occupant-protection systems is adapted to the situation according to the type of rollover, the yaw rate and the lateral acceleration, i.e. systems are triggered after between 30 and 3,000 ms by automatic selection and use of the algorithm module appropriate to the type of rollover.

Combined ECUs for seat-belt pretensioners, front and side airbags and rollover protection equipment

Optimum occupant protection against the effects of frontal, offset, oblique or pole impact is obtained through the precisely coordinated interaction of electronically detonated pyrotechnical front airbags and seat-belt pretensioners. To maximize the effect of both protective devices, they are activated with optimized time response by a common ECU installed in the passenger cell.

The following functions are currently incorporated in the central ECU, also referred to as the trigger unit:

- Crash sensing by acceleration sensor and safety switch or by two acceleration sensors without safety switch (redundant, fully electronic sensing).
- Rollover detection by yaw rate and acceleration sensors that record y and z axis acceleration in the low-*g* range (up to approximately 5 *g*).
- Prompt activation of front airbags and seat-belt pretensioners in response to different types of impact in the vehicle longitudinal direction (e.g. frontal, oblique, offset, pole, rear-end).
- Control of rollover protection equipment.
- For the side airbags, the ECU operates in conjunction with a central lateral sensor and two or four peripheral acceleration sensors. The peripheral acceleration sensors (PAS) transmit the triggering command to the central ECU via a digital interface. The central ECU triggers the

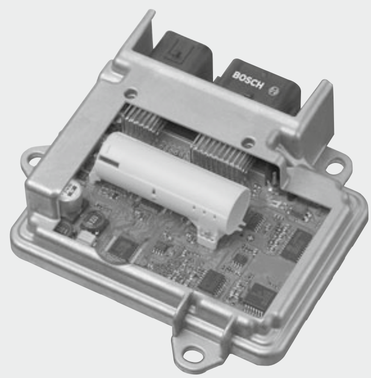
4 Central combined airbag 9 ECU (block diagram)

Terminal designations:

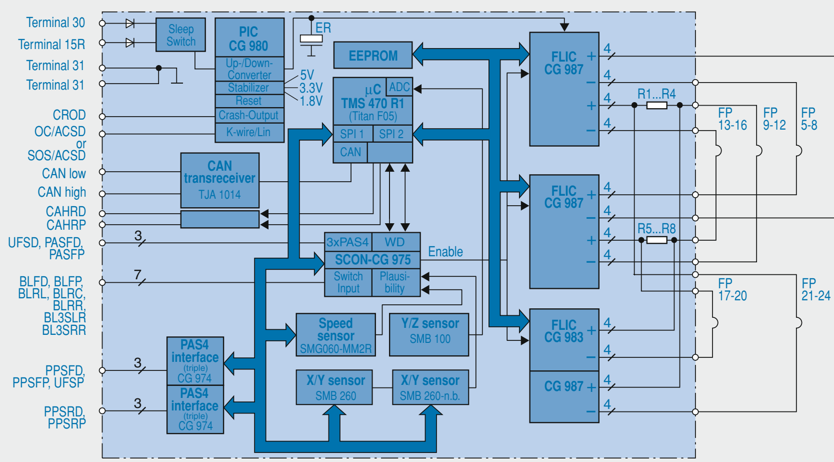
Terminal 30	Direct battery positive, not fed through ignition lock
Terminal 15R	Switched battery positive when ignition lock in “radio”, “ignition on” or “starter” position
Terminal 31	Body ground (at one of the device mounting points)

Abbreviations:

CROD	CRash Output Digital
OC/ACSD	Occupant Classification/ Automatic Child Seat Detection
SOS/ACSD	Seat-Occupancy Sensing/ Automatic Child Seat Detection
CAN low	Controller Area Network, low level
CAN high	Controller Area Network, high level
CAHRD	Crash Active Head Restraint Driver
CAHRP	Crash Active Head Restraint Passenger
UFSD	UpFront Sensor Driver
PASFD	Peripheral Acceleration Sensor Front Driver
PASFP	Peripheral Acceleration Sensor Front Passenger
BLFD	Belt Lock (switch) Front Driver
BLFP	Belt Lock (switch) Front Passenger
BLRL	Belt Lock (switch) Rear Left
BLRC	Belt Lock (switch) Rear Center
BLRR	Belt Lock (switch) Rear Right
BL3SRL	Belt Lock (switch) 3rd Seat Row Left
BL3SRR	Belt Lock (switch) 3rd Seat Row Right
PPSFD	Peripheral Pressure Sensor Front Driver
PPSFP	Peripheral Pressure Sensor Front Passenger
UFSP	UpFront Sensor Passenger
PPSRD	Peripheral Pressure Sensor Rear Driver
PPSRP	Peripheral Pressure Sensor Rear Passenger
ZP	Firing pellets 1...4 or 21...24
FLIC	Firing Loop Integrated Circuit
PIC	Periphery Integrated Circuit
SCON	Safety CONtroller
µC	Microcontroller



UKI00350Y



UKI0036-4E

side airbags provided the internal lateral sensor has confirmed a side impact by means of a plausibility check. Since the central plausibility confirmation arrives too late in the case of impacts into the door or above the sill, peripheral pressure sensors (PPS) inside the door cavity are used to measure the adiabatic pressure changes caused by deformation of the door. This will result in rapid detection of door impacts. Confirmation of "plausibility" is now provided by PAS mounted on supporting peripheral structural components. This is now unquestionably faster than the central lateral-acceleration sensors.

- Voltage transformer and energy accumulator in case the supply of power from the vehicle battery is interrupted.
- Selective triggering of the seat-belt pretensioners, depending on monitored seat-belt buckle status: firing of the airbag only takes place if the belt is engaged in the belt buckle. At present, proximity-type seat-belt buckle switches are used, i.e. Hall IC switches which detect the change in the magnetic field when the buckle is fastened.
- Setting of multiple trigger thresholds for two-stage seat-belt pretensioners and two-stage front airbags depending on the status of belt use and seat occupation.
- Reading of signals from the interior sensors and appropriate triggering of restraints.
- Watchdog (WD): airbag triggering units must meet high safety standards with regard to false activation in non-crash situations and correct activation when needed (crashes). For this reason, ninth-generation (AB 9) airbags launched in 2003 incorporate three independent hardware watchdogs (WDs): WD1 monitors the 2 MHz system eClock using a dedicated, independent oscillator. WD2 monitors the realtime processes (time base 500 µs) for correct and complete sequence. For this reason, the safety controller (SCON, see Fig. 4) sends the microcomputer eight digital messages to which it must respond by sending eight correct replies to the SCON within a time window of 1 ± 0.3 ms. WD3 monitors the background processes such as the built-in self-test routines of the ARM core for correct operation. The microcomputer's response to the SCON in this case must be provided within a period of 100 ms.
- With AB 9, sensors, analyzer modules and driver stages are linked by two SPIs (Serial Peripheral Interfaces). The sensors have digital outputs and their signals can be transmitted directly via SPIs. Shunts therefore remain on the printed-circuit board without effect, unlike with analog sinusoidal transfers, and a high level of functional reliability is achieved. Deployment is only permitted if an independent hardware plausibility channel has also detected the impact and enables the driver stages for a limited period.
- Diagnosis of internal and external functions and of system components.
- Storage of failure modes and durations with crash recorder; readout via the diagnosis or CAN-bus interface.
- Warning-lamp activation.

Acceleration sensors

Acceleration sensors for crash sensing can be located at the following points in the vehicle:

- Directly integrated in the ECU (seat-belt pretensioners, front airbag)
- At selected points on the right and left-hand side of the vehicle on supporting structural parts such as seat crossmembers, door sills, b and c-pillar (side airbag) or
- In the deformation zone at the front end of the vehicle (upfront sensors for "intelligent airbag systems")

These sensors are surface micromechanical sensors consisting of fixed and moving finger structures and spring pins. Since the sensors only have low working capacitances (approximately 1 pF), it is necessary to accommodate the evaluation electronics in the same housing in the immediate proximity of the sensor element so as to avoid stray capacitance and other forms of interference.

Gas inflators

The pyrotechnical propellant charges of the gas inflators for generating the airbag inflation gas and for actuating seat-belt pretensioners are activated by an electrical firing element. The gas inflator in question inflates the airbag with charge gas. The driver airbag built in the steering-wheel hub (volume approx. 60 l) or, as the case may be, the passenger airbag fitted in the glove compartment space (approx. 120 l) is inflated in approx. 30 ms from firing.

AC firing

In order to prevent inadvertent triggering through contact between the firing element and the vehicle system voltage (e.g. faulty insulation in the wiring harness), the firing element is fired by alternating-current pulses with a frequency of approx. 80 kHz (AC firing). A small ignition capacitor with a capacitance of 470 nF incorporated in the firing circuit in the firing element plug galvanically isolates the firing element from the DC current. This isolation from the vehicle system voltage prevents inadvertent triggering, even after an accident when the airbag remains untriggered and the occupants have to be freed from the deformed passenger cell by emergency services. It may even be necessary to cut through the (permanent +) firing circuit wires in the steering column wiring harness and short-circuit them according to positive and ground.

Passenger-compartment sensing

For passenger classification, an absolute weight measuring method is available with the iBolt (intelligent bolt). These force-measuring iBolts (Fig. 1) secure the seat frame (seat link) to the sliding rail and replace the four mounting screws otherwise used. They measure the weight-dependent change in the gap between the bolt sleeve and the internal bolt with integral Hall-element IC connected to the sliding base.

▶ Micromechanics

Micromechanics is defined as the application of semiconductor technology in the production of mechanical components from semiconductor materials (usually silicon). Not only silicon's semiconductor properties are used but also its mechanical characteristics. This enables sensor functions to be implemented in the smallest-possible space. The following techniques are used:

Bulk micromechanics

The silicon wafer material is processed using anisotropic (alkaline) etching and, where needed, an electrochemical etching stop. The material is etched away from the reverse side of the silicon layer (Fig. 1, Item 2) in those areas where it is not protected by the etching mask (1). This method can be used to create very small diaphragms (a) with typical thicknesses of between 5 and 50 µm, holes (b) and bars and ridges (c), e.g. for pressure or acceleration sensors.

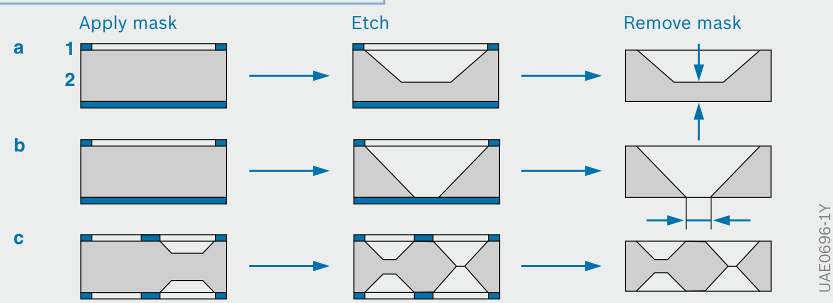
Surface micromechanics

The substrate material here is a silicon wafer on whose surface very small mechanical structures are formed (Fig. 2). First of all, a "sacrificial layer" is applied and shaped (A) using semiconductor production processes (e.g. etching). An approx. 10 µm polysilicon layer is then deposited on top of this (b) and structured vertically using a mask and etching (c). In the final processing step, the "sacrificial" oxide layer underneath the polysilicon layer is removed by means of gaseous hydrogen fluoride (d). In this way, structures such as flexible electrodes (Fig. 3) for acceleration sensors can be created.

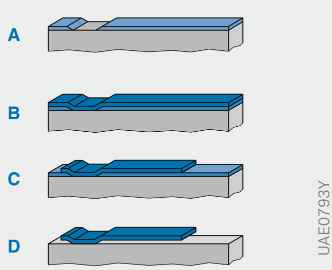
Wafer bonding

Anodic and seal glass bonding are methods used to join wafers together by the action of electricity and heat or heat and pressure in order, for instance, to hermetically seal a reference-vacuum chamber or to protect sensitive structures by placing a cap over them.

1 Structures produced by bulk micromechanics



2 Surface micromechanics (processing steps)



3 Surface micromechanics (structure details)

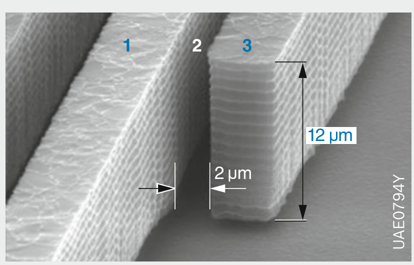


Fig. 1
a Production of diaphragms
b Production of holes
c Production of bars and ridges

1 Etching mask
2 Silicon

Fig. 2
A Cutting and structuring the sacrificial layer
B Cutting the polysilicon
C Structuring the polysilicon
D Removing the sacrificial layer

Fig. 3
1 Fixed electrode
2 Gap
3 Spring electrode

Hybrid drives

A concept for economizing on fuel, for reducing CO_2 and pollutant emissions, and at the same time for increasing driving pleasure and driving comfort is provided by hybrid electric vehicles (Hybrid Electric Vehicle, HEV). For drive purposes these vehicles use both an internal-combustion engine and at least one electric motor (electrical machine). There are in this respect a multitude of HEV configurations which partly pursue different optimization objectives and which utilize to differing extents electrical energy to drive the vehicle.

Principle

There are essentially three objectives being pursued when hybrid electric drives (Fig. 1) are used: fuel economy, reduced emissions and increased torque and power ("driving pleasure"). Different hybrid concepts are used, depending on the objective being pursued. A distinction is basically made between *mild-hybrid* and *full-hybrid* vehicles, depending on their ability also to be able to run by pure electrical means.

In a *mild hybrid* the internal-combustion engine is assisted by an electric motor, which delivers additional drive power and braking power in different operating states. In a *full hybrid* as well the internal-combustion engine is combined with one (or two) electric motor(s). In additional to running on the internal-combustion engine and being assisted by the electric motor, this latter type also allows for purely electric driving.

Both hybrid concepts have a start/stop function, as is familiar from conventional start/stop systems. When the vehicle is stationary, e.g., when stopped at traffic lights, the internal-combustion engine is switched off. The avoidance of idling phases helps to save fuel. An automatic start/stop system can, depending on the level of hybridization, naturally also be used in vehicles with conventional drives.

Both mild-hybrid and full-hybrid systems require an electric energy accumulator, which powers the driving electric motor. This energy accumulator is usually a traction battery at a comparatively high voltage level.

The combination of electric and combustion-engine drive sources in the mild hybrid and full hybrid has various advantages over conventional drivetrains:

- The electric motor offers constantly high torques at low rotational speeds. In this

1 Principle of the hybrid drive using the example of a parallel hybrid

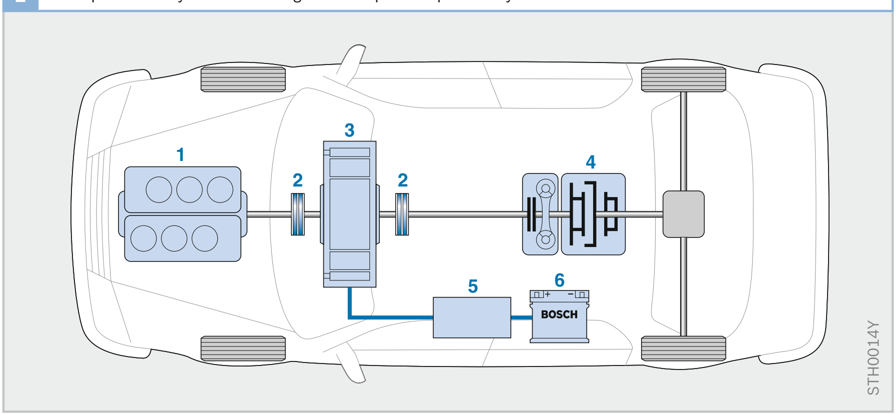


Fig. 1
1 Internal-combustion engine
2 Clutch
3 Electric motor
4 Transmission
5 Inverter
6 Battery

K. Reif (Ed.), *Fundamentals of Automotive and Engine Technology*,
DOI 10.1007/978-3-658-03972-1_20, © Springer Fachmedien Wiesbaden 2014

way it ideally supplements the internal-combustion engine, whose torque only starts to increase at mid-range rotational speeds. The electric motor and internal-combustion engine together are thus able to deliver a high dynamic response from every driving situation (Fig. 2).

- The assistance provided by the electric motor makes it possible to operate the internal-combustion engine predominantly in the range of its best efficiency or in ranges in which only low pollutant emissions occur (operating-point optimization).
- The combination with an electric motor facilitates if necessary the use of a smaller internal-combustion engine while retaining the same overall power output (power-neutral downsizing).
- The combination with an electric motor facilitates if necessary the use of a higher-geared transmission while retaining the same levels of driving performance (downspeeding).

The hybrid systems also offer the possibility of fuel economy through the recovery of braking energy. Through generator operation of the electric motor (or if necessary by means of an additional generator), it is possible when braking to convert part of the vehicle's kinetic energy into electrical energy. The electrical energy is stored in the energy accumulator and can be used to drive the vehicle.

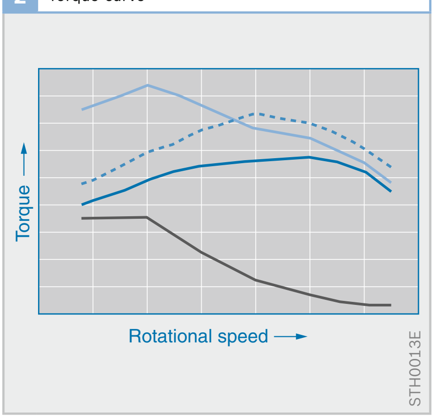
Operating modes

Depending on the operating state and required torque, the internal-combustion engine and the electric motor contribute to the drive power to different extents. The hybrid control system determines the torque distribution between the two drives (see section *Operating strategy*). The way in which the internal-combustion engine, electric motor(s) and energy accumulator interact defines the different operating modes: hybrid and electric driving, boosting, generator operation and recuperative braking.

Hybrid driving

Hybrid driving refers to all those states in which both the internal-combustion engine and the electric motor generate drive torque (Fig. 3). When distributing the drive torque, the hybrid control system takes into account – in addition to the optimization objective (fuel consumption, emissions) – in particular the state of charge of the energy accumulator.

2 Torque curve



3 Hybrid driving

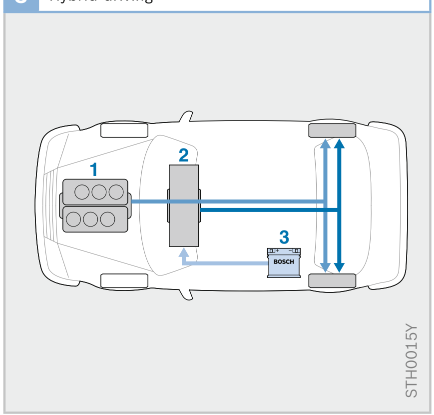


Fig. 2
— Resulting hybrid
-- Standard engine, 1.6 *l* displacement
— Engine, downsized, 1.2 *l* displacement
— Electric motor, 15 kW

Fig. 3
1 Internal-combustion engine
2 Electric motor
3 Battery

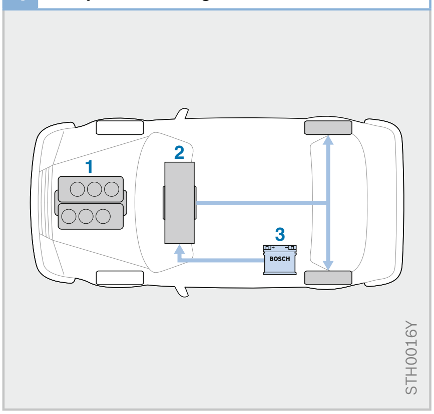
Purely electric driving
Purely electric driving, in which the vehicle is driven over longer distances by the electric motor alone, is only possible with the full hybrid. The internal-combustion engine is decoupled from the electric motor for this purpose (Fig. 4). In this operating mode the vehicle can run virtually noiselessly and locally without emissions.

Boosting
In boosting mode the internal-combustion engine and the electric motor deliver positive drive torque. Both deliver their maximum torque for the vehicle's maximum propulsion torque (Fig. 6).

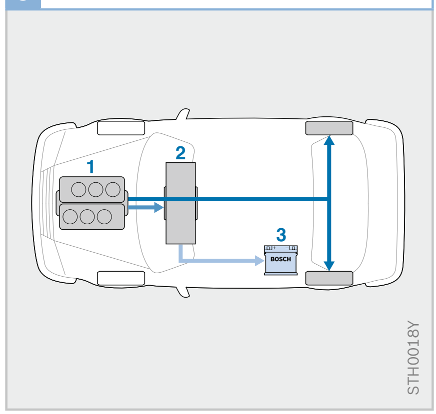
Generator mode
The electric energy accumulator is charged in generator mode. For this purpose the internal-combustion engine is operated in such a way as to deliver a greater amount of power than is needed for the desired propulsion of the vehicle. The excess amount of power is fed to the generator and converted into electrical energy, which is stored in the energy accumulator (Fig. 5).

The energy accumulator is also charged via the generator in overrun mode provided this is permitted by the battery state of charge.

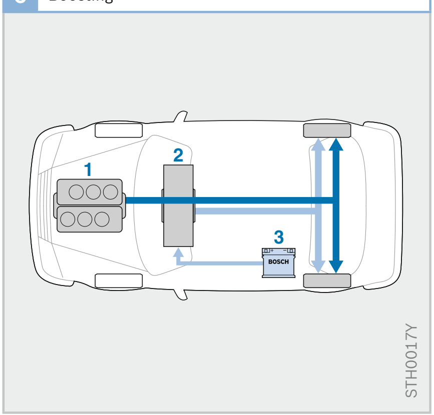
4 Purely electric driving



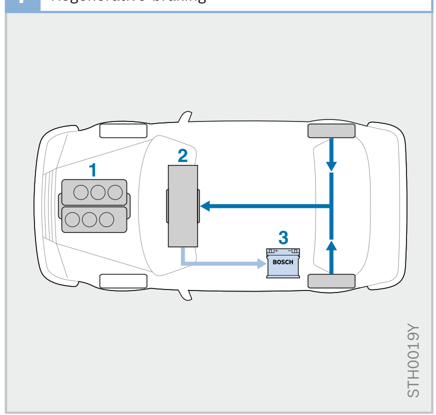
5 Generator mode



6 Boosting



7 Regenerative braking



Figs. 4 – 7
1 Internal-combustion engine
2 Electric motor
3 Battery

Regenerative braking

During regenerative braking the vehicle is not – or not only – braked by the service brake's friction torque, but instead by a generator braking torque of the electric motor. The electric motor is therefore operated like a generator and converts the vehicle's kinetic energy into electrical energy, which is stored in the energy accumulator (Fig. 7).

Regenerative braking is also known as recuperative braking or as recuperation.

Start/stop function

Both mild hybrid and full hybrid have a start/stop function (Fig. 8). But even vehicles with conventional drives can be equipped with a start/stop system.

Function

When the vehicle is stopped, the engine ECU checks whether

- no gear is engaged,
- the speed sensor of the antilock braking system indicates zero,
- the electronic battery sensor is signaling sufficient energy for a starting process.

When these vehicles are satisfied, the engine is automatically switched off.

As soon as the clutch is actuated the starter receives the signal to restart the engine. The engine is started quickly and quietly and is immediately ready for operation again.

Components

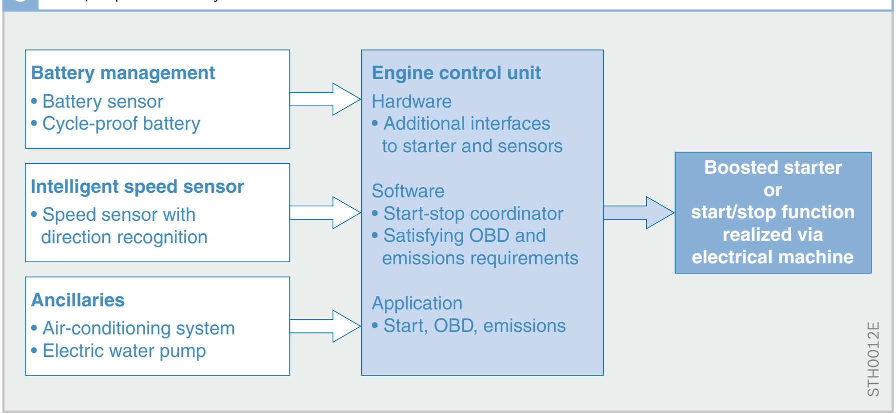
In the start/stop system a reinforced starter (Fig. 9, no. 1) replaces the conventional starter.

The start/stop system requires an adapted engine management system (4), which has additional interfaces to the starter and sensors. Since the start/stop system is an emission-relevant system, it must satisfy the requirements of OBD (on-board diagnosis), i.e., it must be monitored in driving mode and exhaust-gas-relevant faults must be stored in the ECU's fault memory.

Because of the many starting processes it has to manage, the battery (2) must be cycle-proof. It is monitored by a battery sensor, which before the internal-combustion engine is automatically switched off checks the battery state of charge and signals this to the engine ECU.

Ancillaries such as, for example, the A/C compressor, which are normally driven via the internal-combustion engine and are also required during the standstill phases, must be electrically driven or replaced by

8 Start/stop functionality



9 Components of the start/stop system

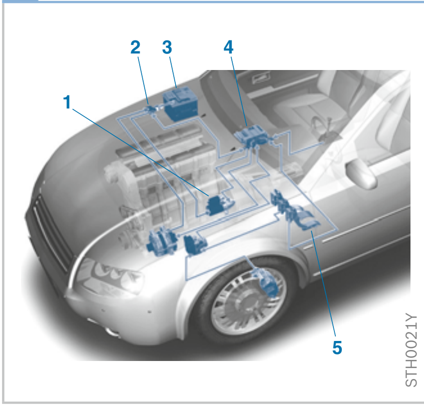


Fig. 9
1 Starter
2 Battery sensor
3 Battery
4 Engine ECU with start/stop function
5 Pedals and sensors

other solutions. This also applies to the mild hybrid and the full hybrid, in which the start/stop function can be realized by means of the electric motor.

Fuel economy

Fuel savings of 3.5 % to 4.5 % can be achieved by the start/stop system in the New European Driving Cycle.

Degrees of hybridization

The degree of hybridization indicates the extent to which distribution of the drive power can be varied between the internal-combustion engine and the electric motor. A distinction is made between mild hybrid and full hybrid, depending on the degree of hybridization. They differ essentially in the power output of the electric motor or with regard to the amount which the electric drive contributes to the overall drive power. They also differ with regard to the energy content of the electric accumulator.

Mild hybrid

Function

The mild hybrid (Fig. 10) offers in addition to the start/stop function the possibility of recuperative braking (1) and of torque assistance by the electric motor (2). The electric motor delivers an additional torque, which is added to the internal-combustion engine's torque. The energy accumulator (4) makes available for this purpose an electrical power output of normally up to 20 kW. This output is mainly used for starting and accelerating at low engine speeds.

Purely electric driving is only possible when the internal-combustion engine is under coupled motion, since it cannot be decoupled from the electric motor. Such an operating state is useful in energy terms only if the drag torque of the internal-combustion engine is not too great. Mild hybrids are therefore often combined with internal-combustion engines which demonstrate the possibility of cylinder shutoff.

Design

The mild hybrid is realized as a parallel hybrid, i.e., internal-combustion engine and electric motor are positioned on the same shaft (crankshaft).

In addition to the conventional low-voltage electrical system (14 V) for supplying the loads/consumers, a traction electrical system with a clearly higher voltage is provided to feed the electric drive.

For a detailed design, see section *Parallel hybrid*.

Fuel economy

The fuel savings of a mild hybrid compared with a conventional vehicle can be up to 15 % in the New European Driving Cycle (NEDC).

Full hybrid

Function

The full hybrid (Fig. 10) can, in contrast to the mild hybrid, be driven over longer distances with the electric drive alone. The internal-combustion engine does not rotate during electric driving. The voltage of the traction electrical system or the battery usually ranges between 200 and 350 V.

Design

The full hybrid can be realized with a parallel or serial flow of energy or can be a combination of parallel and serial flows of energy. The parallel flow of energy can be represented by one electric drive. To realize a serial flow of power, there must be two electric drives in the drivetrain.

In a parallel hybrid with two clutches (P2-HEV) an interrupting clutch is fitted between the internal-combustion engine and the electric motor. This enables the internal-combustion engine to be decoupled from the electric motor for purely electric driving.

For a detailed design, see section *Parallel hybrid*.

A full hybrid with combined serial and parallel power flows is realized by a power-branching system, in which the central transmission element is a planetary-gear set.

For a detailed design, see section *Power-branching hybrid*.

Fuel economy

The fuel savings of a full hybrid can be up to 30 % in the New European Driving Cycle.

Plug-in hybrid

Full hybrids can alternatively also be designed as plug-in hybrids. These offer the possibility of charging the traction battery externally (e.g., from the power socket) via a corresponding charger. Here it is advisable to use a larger battery in the vehicle so as to be able to cover shorter distances by purely electrical means and to use the hybrid drive for longer journeys only.

The greatest challenge currently facing plug-in hybrids is posed by disadvantages with regard to costs and weight of the larger battery. Furthermore the limited charging power of the domestic power sockets results in long charging times.

10 Components of the hybrid system (mild and full hybrid)

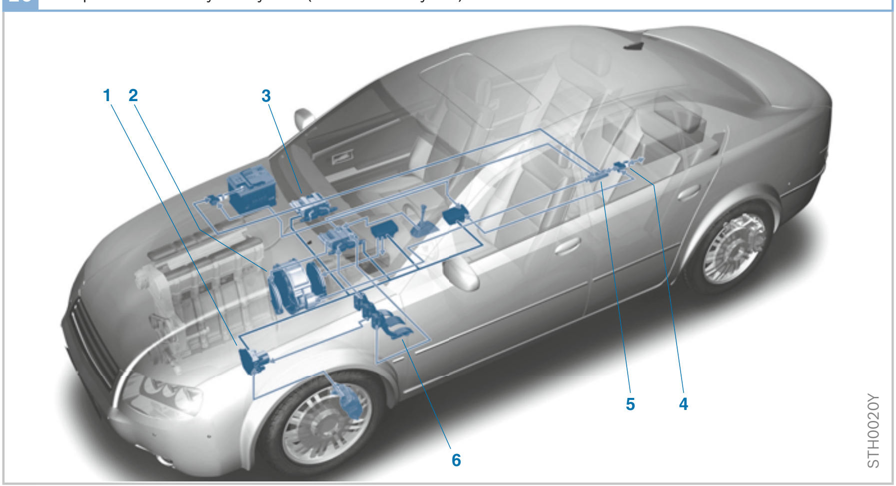


Fig. 10
1 Regenerative braking system
2 Electric motor (IMG)
3 Hybrid and engine ECU
4 High-voltage battery and battery-management system
5 Inverter
6 Pedals and sensors

Drive configurations

Series hybrid drive

The series hybrid drive (S-HEV) is characterized by the series connection of the energy converters (electric motors and internal-combustion engine) (Fig. 11). The series arrangement requires, in addition to the internal-combustion engine, two electric motors, where one operates as a generator and the other as a motor. The internal-combustion engine is not connected to the powered axle.

First the kinetic energy of the internal-combustion engine is converted by a generator (3) into electrical energy. The pulse-controlled inverter (5) converts the power output based on the driver command and supplies the second electric motor (4), which is responsible for driving the wheels. This means that the power output necessary to move the vehicle is transferred exclusively from the electric motor (4) to the drive shaft.

The advantage of this drivetrain arrangement is that the operating point of the internal-combustion engine can be freely selected as long as the requested electrical energy is made available. Depending on the operating strategy, the internal-combustion engine with its power output can follow the current demand or it can operate uniformly at the most efficient operating point and deliver excess energy to the battery. Operation at the most efficient operating point provides for particularly low pollutant emissions – with the exception of NO_X emissions.

It must be borne in mind that both electric motors must be of sufficient size to be able to consume or deliver the power of the internal-combustion engine. The high power capability of the electric motors also has the advantage that even marked vehicle decelerations can be recuperated.

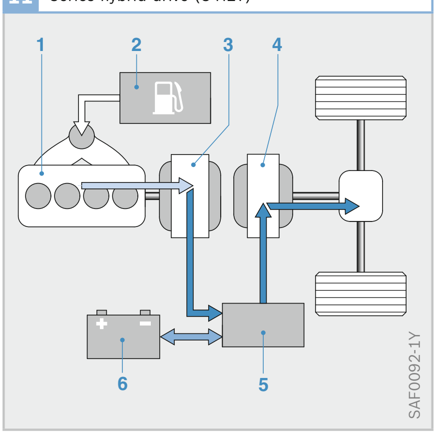
A disadvantage of this arrangement lies in the multiple energy conversion and the associated losses of efficiency. Starting out from the customary mid-range losses of the individual components, there is a total loss of approximately 30 %. Further disadvantages are high costs, component size and high excess weight. The use of this arrangement in passenger cars is therefore very limited.

The series hybrid drive is used in heavy commercial vehicles, such as, for example, diesel-electric drives in locomotives, and in buses which are driven in urban traffic with high levels of stop-and-go operation.

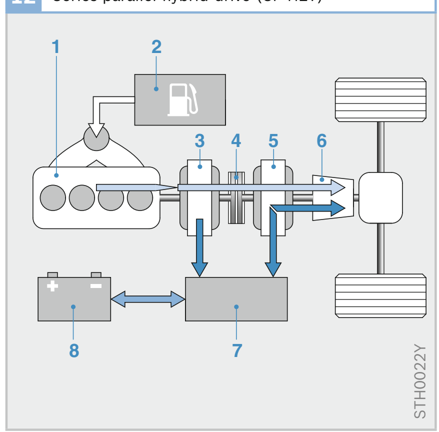
Fig. 11
1 Internal-combustion engine
2 Tank
3 Generator
4 Electric motor
5 Inverter
6 Battery

Fig. 12
1 Internal-combustion engine
2 Tank
3 Generator
4 Clutch
5 Electric motor
6 Transmission
7 Inverter
8 Battery

11 Series hybrid drive (S-HEV)



12 Series-parallel hybrid drive (SP-HEV)



The series-parallel hybrid (SP-HEV) represents a special form of the series concept (Fig. 12). It differs from the series drivetrain arrangement in that it has a clutch which connects the two electric motors. When the clutch is open, the system behaves like the above-mentioned S-HEV. When the clutch is closed, the internal-combustion engine can deliver its power directly to the powered axle, which corresponds to a parallel drivetrain topology. The disadvantages of the S-HEV with regard to costs, space and excess weight remain, but the electric motors can be of smaller design in that the transmittable power in series operation does not have to cover the vehicle's full drive power aimed for. The series operation range can be limited to smaller power outputs, since at higher speeds and power requirements parallel operation is to be preferred, also because if a better overall drive efficiency.

Parallel hybrid drive

Unlike the series and power-branching concepts, parallel drivetrain topologies require only one electric motor (Fig. 13). This can be operated like both a generator and a motor, and is mechanically connected to the internal-combustion engine's crankshaft. This involves a torque addition, where the torques of the drives (internal-combustion engine and electric motor) can be freely varied while the rotational speeds are in fixed proportion to each other. In addition, when the clutch is closed, a purely mechanical power transmission from the internal-combustion engine to the powered axle is possible, irrespective of the state of the electric motor. The overall efficiency is thus higher than in the other hybrid topologies.

Direct connection of the electric motor to the internal-combustion engine has however a disadvantageous effect on the capacity to freely select the operating point, since the rotational speeds of both assemblies is determined by the transmission ratio and the driving speed. This can be altered by a transmission shift, but only for both assemblies in the same way. When a range transmission is used, the rotational speed of the drive combination of electric motor and internal-combustion engine therefore cannot be continuously freely selected.

A fundamental advantage of the parallel hybrid is the possibility of maintaining the conventional drivetrain in wide ranges. This has a positive effect both on space and vehicle production, and also on the usual driveability and customer acceptance. The development and implementation expenditure of the parallel drivetrain topology for passenger cars is, when compared with series and power-branching concepts, low since lower electrical power outputs are required and the necessary adaptations when converting a conventional drivetrain are fewer.

The parallel hybrid drive is further subdivided according to the number of clutches and the positioning of the electric motor. The most frequently encountered design variations are explained in the following.

Parallel hybrid with one clutch

In the parallel hybrid with only one clutch (P1-HEV; Fig. 13) the electric motor is rigidly connected to the internal-combustion engine's crankshaft such that the electric motor cannot be operated independently of the engine. For this reason, during regenerative braking, the engine must be under coupled motion, i.e., the engine's drag torque is lost as recuperation potential. Purely electric driving is indeed theoretically possible, but the engine must also be under coupled motion here. The resulting losses and noise and vibration problems prohibit this vehicle operation. Merely purely electric gliding is representable from a certain speed. Here the electric motor delivers the propulsion torque for maintaining the speed and the drag power of the engine.

In the simplest version of the P1-HEV a crankshaft starter-generator is used, where the electric motor is only responsible for starting the internal-combustion engine and for supplying the vehicle electrical system. Thanks to an additional electrical accumulator and higher electric-motor power capability, it is possible to design a full mild hybrid which additionally facilitates support of the engine by the electric motor and recovery of the braking energy.

Parallel hybrid with two clutches

To facilitate purely electric driving and regenerative braking to the full extent (without drag losses), an additional clutch is required between the internal-combustion engine and the electric motor (Fig. 14). Based on the number of clutches, this topology is known as P2-HEV. In recuperation phases or for electric driving the engine is disconnected from the drivetrain and switched off when the second clutch is opened. In this way the vehicle's deceleration energy can be recovered without drag losses and stored in the battery. Recuperation is merely limited by the power limits of the electric motor.

Even for electric driving the engine does not have to be under coupled motion such that slow creeping becomes comfortably possible. It is also possible to use the electric motor's full power output for electric driving without power losses to place the engine under coupled motion. However, it must be possible for the engine to be restarted by the electric motor at any time, and thus some of the electric motor's power capability must be reserved for this purpose.

The greatest challenges faced by the P2-HEV concept lie in accommodating the second clutch in the smallest space available and in restarting the engine from electric driving without compromising on comfort.

Axle-split parallel hybrid (AS-HEV)

In the P1-HEV and P2-HEV the electric motor and internal-combustion engine are arranged on a common powered axle in front of the transmission. Both drive assemblies thus always operate at the same rotational speed. One way of eliminating this uniformity of speed is to split the drive assemblies between the two vehicle axles. This topology is know as axle-split hybrid (AS-HEV).

13 Parallel hybrid with one clutch (P1-HEV)

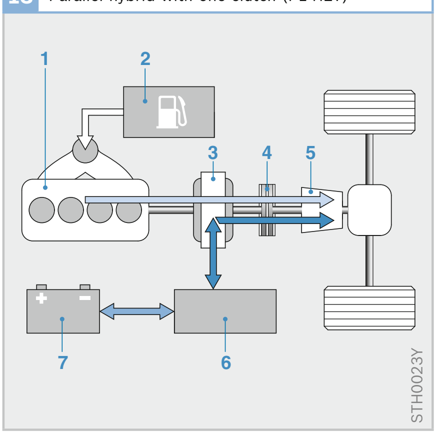


Fig. 13
1 Internal-combustion engine
2 Tank
3 Electric motor (IMG)
4 Clutch
5 Transmission
6 Inverter
7 Battery

14 Parallel hybrid with two clutches (P2-HEV)

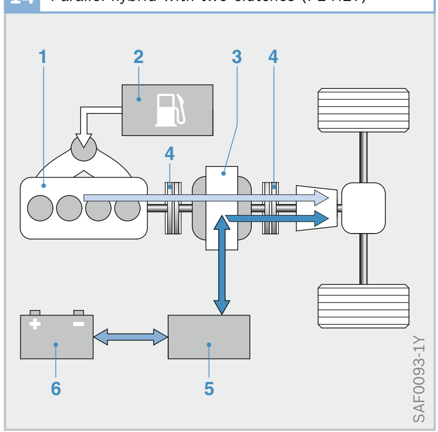
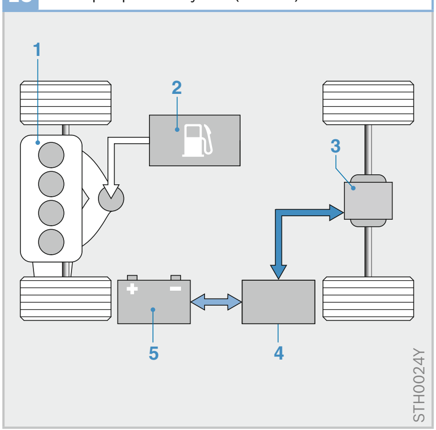


Fig. 14
1 Internal-combustion engine
2 Tank
3 Electric motor/generator
4 Clutch
5 Inverter
6 Battery

In the AS-HEV the internal-combustion engine and the electric motor are not directly connected to each other mechanically, but instead act on different vehicle axles (Fig. 15). Traction-force addition is thus realized via the road. Regenerative braking and electric driving are effected on front-wheel-drive vehicles via the electric rear axle, while the unaltered conventional drivetrain drives the front axle. When both assemblies are active as an engine/motor, this gives rise to a four-wheel drive. The torques between front and rear axles can be freely varied here between the respective power limits.

It becomes clear that an essential difference exists between the AS-HEV and the other parallel hybrids when the vehicle is stationary. When the axle is stationary, the electric motor in the AS-HEV cannot generate electrical power. Thus the vehicle electrical system must be supplied and air conditioning effected when stationary by other means. This is possible, for example, using a powerful generator on the engine. With the aid of a DC/DC converter the generator can charge the high-voltage battery even when the vehicle is stationary and supply the high-voltage loads/consumers.

15 Axle-split parallel hybrid (AS-HEV)



There are various advantages to connecting the electric motor to its own vehicle axle:

- Package: The conventional drivetrain does not have to be altered.
- The engine and electric motor can be run at different rotational speeds, thus allowing high-speed concept to be used for the electric motor as well.
- High levels of efficiency are achieved in recuperation and electric driving.
- There is no need for the engine to be started by the electric motor (but a separate starter is required).

Disadvantageous aspects of the AS-HEV are:

- A separate starter is needed for the engine.
- A configuration of the torque and speed ranges of the electric motor without transmission to the vehicle's entire driving range is required. (Alternative: an additional simple transmission for the electric motor, e.g., 2-speed.)
- When stationary the high-voltage battery cannot be charged (only with additional measures, e.g., DC/DC converter).
- Supply of the 12V vehicle electrical system when stationary must be ensured (e.g,. 12V generator).
- Monitoring of driving dynamics (ESP) is required for both axles.

Electric 4WD functionality

In the AS-HEV a four-wheel drive (4WD) is realized by the combination of a conventional drive and an electrically driven axle. An electric final drive can also be combined with any other hybrid configuration in order thereby to realize electric 4WD functionality.

Fig. 15
1 Internal-combustion engine
2 Tank
3 Electric motor
4 Inverter
5 Battery

Parallel hybrid with different transmissions

The parallel hybrid can basically be realized with all transmission types, where a combination with particular transmissions produces special advantages. It is particularly worth highlighting in this respect the dual-clutch transmission (DCT). This consists of two sub-transmissions, which can select different gears independently of each other. This creates the possibility of connecting the electric motor to one of these sub-transmissions and running it in a different gear from the engine (Fig. 16). In this way it is possible to optimize the electric motor's operating point in some ranges independently of the engine's operating point, which opens up an additional efficiency potential.

Power-branching hybrid drive

Principle

The core element of the power-branching hybrid topology is the planetary-gear set (Fig. 18). In this gear set the power output of the internal-combustion engine is split into two paths. This involves a mechanical path, where power can be transmitted directly to the wheels by gear teeth, and an electrical path. As well as the engine and output, an electric motor (Fig. 17, no. 7) acts on the third shaft of the planetary-gear set. The load point of this electric motor serves to transfer the engine's rotational speed and load according to the drive requirements into wheel speed and output torque.

In a planetary-gear set the rotational speeds of two shafts always determine the rotational speed of the third shaft. The torque ratios between the three shafts are similarly determined in this way. The result of this is that a transmission of power in the mechanical path is only possible where the electric motor draws power and converts it into electrical power. Because electrical power is constantly generated in this way, it is neither possible nor for efficiency reasons sensible to store this power in a battery. For this reason, a second electric motor (4), which is mounted directly on the output shaft, is used to close an electrical path and directly convert the arising electrical power back into mechanical power. Thus a drive request, which is made up of a wheel speed and a desired wheel torque, gives rise to a pre-

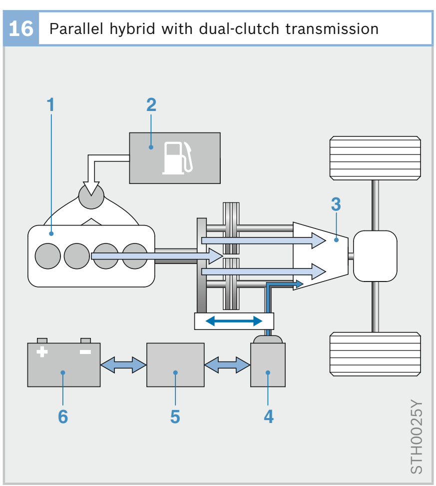


Fig. 16
1 Internal-combustion engine
2 Tank
3 Transmission
4 Electric motor (SMG)
5 Inverter
6 Battery

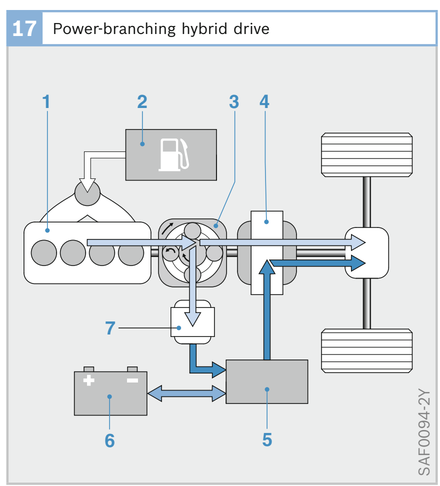


Fig. 17
1 Internal-combustion engine
2 Tank
3 Planetary-gear set
4 Electric motor
5 Inverter
6 Battery
7 Generator

ferred engine speed, which is set using the speed of the first electric motor (7). The desired wheel torque is generated by the engine and transmitted partly via the mechanical path and partly via the electrical path to the wheels.

As in all hybrid vehicles the battery (6) serves to influence specifically the drivetrain operating state. The desired wheel torque can with the aid of the battery result in either a higher or a lower load state. By using the energy stored in the battery, it is possible to avoid very poor engine efficiency ranges, whereby the electric motor (4) alone provides for propulsion of the vehicle and the internal-combustion engine is switched off.

The PS-HEV, as is mass-produced by Toyota in the Prius model, has the arrangement described. The two paths combine the fundamental principles of the series and parallel hybrid drives, which is why the power-branching drive is also referred to as a series-parallel topology.

18 Planetary-gear set (load distributor)

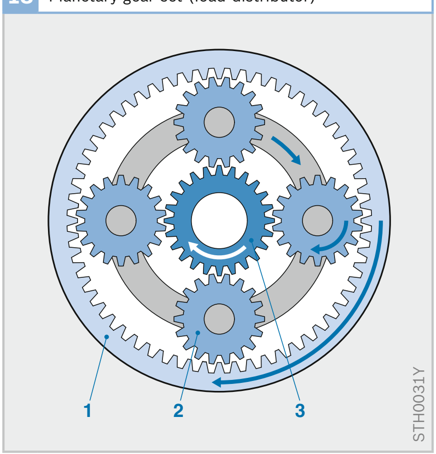


Fig. 18
1 Internal gear: drives the vehicle's powered axle
2 Planetary gears: drive the internal gear
3 Sun gear: drives the generator

Continuously variable transmission

A significant advantage of the power-branching concept lies in the Continuously Variable Transmission [CVT] behavior and thus with the associated free selection of the internal-combustion engine's operating point. Moreover, the drivetrain can be realized without a conventional transmission and in particular without gearshift and clutch elements, which results in high driving comfort without traction-force interruption and in reduced mechanical components.

On the other hand, decoupling the engine speed from the driving speed can give rise to a more unusual driving feel – especially for European car drivers. In this respect it is comparable with the driveability of vehicles with conventional CVT transmissions.

System limits

The previously discussed limitations of a series hybrid with regard to the dimensioning of the electric motors and the efficiency chain are lessened in the power-branching concept. Because a significant amount of drive energy is transported via the electrical path, powerful electric motors are required – depending on the layout of the drivetrain. The necessary energy-conversion processes have an effect on the overall efficiency of the drive – particularly if the vehicle is to be used over a wide driving-speed range. The upshot of this is that the great savings potential that the vehicle demonstrates in urban traffic is not manifested to this extent in long-distance or interstate driving.

In order to obtain an improvement in this field, vehicles are currently being developed which have two driving modes and are therefore known as two-mode hybrids.

Two-mode hybrid

Fig. 19 shows a possible design for a two-mode hybrid. In this example the two-mode hybrid has two electrical CVT drive positions and one purely mechanical transmission. It is possible through the possibilities of combining the input and output shafts of the planetary-gear set to achieve improved efficiency over a wide spread of driving speeds.

The direct mechanical gear step is made possible by the use of two clutches. The excellent overall efficiency and the many degrees of freedom of this concept are offset by the system's high complexity and relatively high costs.

19 Two-mode hybrid

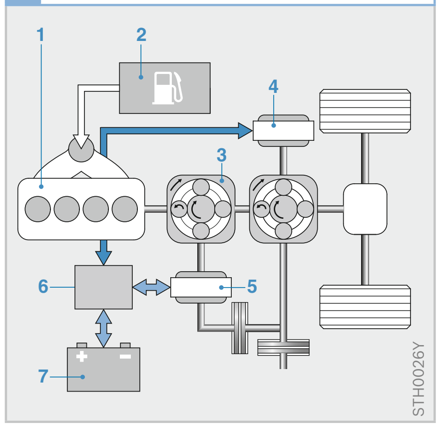


Fig. 19
1 Internal-combustion engine
2 Tank
3 Planetary-gear set
4 Electric motor (SMG)
5 Electric motor (SMG)
6 Inverter
7 Battery

Operation of hybrid vehicles

Operation of a hybrid electric vehicle is essentially determined by the operating strategy. Depending on the higher optimization objective (reduced emissions, fuel economy), the operating strategy establishes at every moment the distribution of the requested drive torque to the internal-combustion engine and the electric motor so that the engine operates at the most favorable operating points possible. The operating strategy also controls the generation of electrical energy for charging the traction battery.

Hybrid control

The efficiency which can be achieved with the relevant hybrid drive is dependent not only on the hybrid topology but also crucially on the higher-level hybrid control. Fig. 20 uses the example of a vehicle with a parallel hybrid drive to show the networking of the individual components and control systems in the drivetrain. The higher-level hybrid control coordinates the entire system, the subsystems of which have their own control functions. These control functions are battery management, engine management, management of the electric drive, transmission management and management of the braking system. In addition to pure control of the subsystems, the hybrid control also includes an operating strategy which optimizes the way in which the drivetrain is operated. The operating strategy brings influence to bear on the consumption- and emission-reducing functions of the HEV, i.e., on start-stop operation of the engine, regenerative braking and operating-point optimization. These include the decisions on a driving state such as electric driving or recuperation and distribution of the driver-command torque to the engine and electric motor.

An important integral part of operating-point optimization is the electric-driving function. It is possible through boosting the electric drive to achieve a higher torque and thus a better acceleration capability particularly at low engine speeds. This requires a holistic consideration of design and operating-strategy optimization to exploit the maximum potential.

20 Networking of control systems in the drivetrain

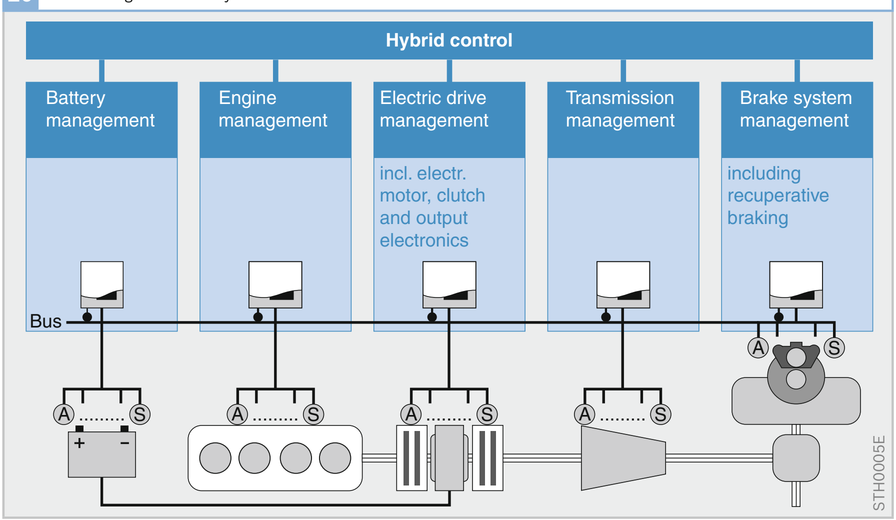


Fig. 20
A Actuator
S Sensor

K. Reif (Ed.), *Fundamentals of Automotive and Engine Technology*,
DOI 10.1007/978-3-658-03972-1_21, © Springer Fachmedien Wiesbaden 2014

Operating strategy here means a driving-situation-dependent distribution of torque between the two drive sources of engine and electric motor.

Operating strategies for hybrid vehicles

Further steps for reducing CO_2 are currently required for all internal-combustion engine concepts. Furthermore, vehicles with diesel engines show a reduction potential with regard to untreated NO_X emissions. Improvements can be achieved here by moving the engine operating points into the ranges of lower emissions.

Operating strategy for reducing NO_X

Vehicles with lean-running internal-combustion engines already achieve relatively low consumption values in part-load operation. At low part load, however, the friction loss increases such that the specific fuel consumption is also high. In addition, low combustion temperatures and a local oxygen deficiency in the low part-load range result in high carbon-monoxide and hydrocarbon emissions.

Already a relatively weak electric assembly can replace the internal-combustion engine in the low load range. If the required electrical energy can be recovered through regeneration, this simple strategy can deliver a huge benefit with regard to fuel consumption and emissions.

It is to be expected that in future lower emission limits will be increased for nitrogen oxides. By avoiding unfavorable engine operating points, hybridization of a diesel vehicle offers the possibility of significantly influencing exhaust-gas emissions. With low engine emissions the measures for exhaust-gas treatment could be partially reduced.

Fig. 21a shows the ranges in which the internal-combustion engine is primarily operated in the New European Driving Cycle (NEDC). The passenger-car diesel engine is operated both at low part load (i.e., with poor efficiency levels and high HC and CO emissions) and at medium/higher load (i.e., in the range of high NO_X emissions).

21 Shifting of the operating range with strategy for reducing NO_X

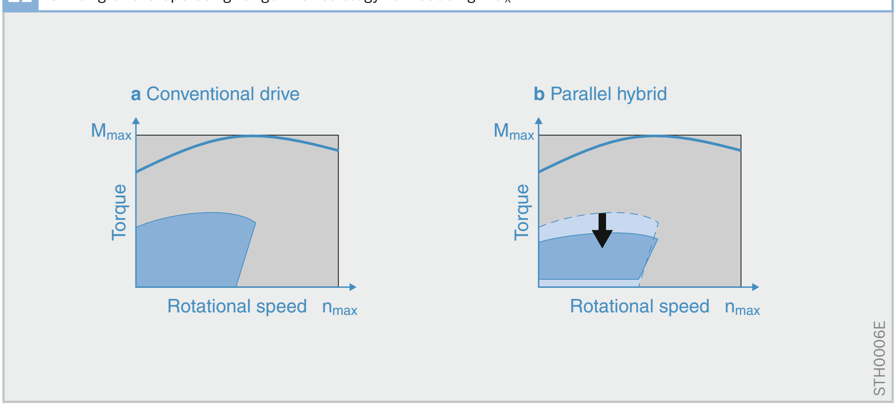


Fig. 21
a Range of operating points in the driving cycle
b Boost: joint operation of engine and electric motor

Fig. 21b shows by way of example the range of operating points for a parallel hybrid which bypasses low internal-combustion engine loads through purely electric driving and/or load-point increase. This on the one hand reduces the fuel consumption, but on the other hand reduces the CO, HC and NO_X emissions – which are high in this range. To achieve a further lowering of NO_X emissions, it is possible to lower load points in the medium load range by simultaneously operating electric motor and engine (boosting).

Operating strategy for reducing CO_2

With vehicles with stoichiometrically running gasoline engines extremely low emission values can be realized on account of the three-way catalytic converter used.
In the hybrid vehicle extremely low emissions are also possible with large-capacity internal-combustion engines by means of appropriate warm-up strategies. Under circumstances the demands place on the exhaust-gas treatment system can even be reduced. The objectives for both gasoline hybrid vehicles and diesel hybrid vehicles are thus fuel economy and increased power.

Fig. 22 shows for the different HEV topologies a possible optimization of the operating range of the internal-combustion engine with regard to minimum CO_2 emissions (i.e., reduced consumption).

In the New European Driving Cycle (NEDC) internal-combustion engines in conventional vehicles are operated at low part load and thus at suboptimum efficiency.
In vehicles with parallel hybrid drives low engine loads can be avoided by means of purely electric driving (Fig. 22b).

Since the required electrical energy as a rule cannot be recovered exclusively through recuperation, the electric motor is then operated as a generator. This results, when compared with a conventional vehicle, in a shift in engine operation to higher loads and thus better levels of efficiency.
In this way more electrical energy can be made available than in the previously described NO_X strategy for diesel, and as a result electric driving is possible to a greater extent. But, here too, on account of the service-life requirements of the traction battery, a compromise must be found between CO_2 emissions and the energy throughput, since a high energy throughput has a negative influence of the service life of the traction battery.

22 Shifting of the operating range with strategy for reducing CO_2

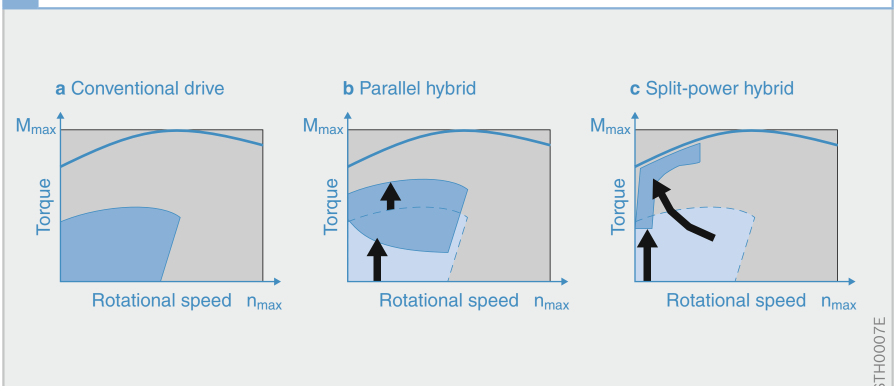


Fig. 22
a Range of operating points in the driving cycle
b Avoidance of lower engine loads through purely electric driving with subsequent charging
c eCVT effect: movement of operating points to the optimum energy range of the drivetrain

In the case of the power-branching hybrid vehicle (Fig. 22 c), the operating range of the internal-combustion engine is subject to greater limitations than the parallel hybrid vehicle. As a rule it is operated as a function of engine speed at the load at which the entire drivetrain operates under optimum energy conditions. Here, too, it is possible, because of the series operating mode, on the electrical path (simultaneous operation of the two electric motors as generators and motors) to keep energy throughput and cycling of the traction battery lower than in a parallel hybrid.

Operating-point optimization

Distribution of drive torque

Different configurations of hybrid vehicle control or operating-strategy optimization have significant effects on fuel consumption, emissions, available torque, and the layout of the components (e.g., operating range of the electric motor and of the engine, energy throughput and cyclization of the electrical accumulator), since their operating points are directly dependent on the operating strategy. It has already become clear that cross-system hybrid control is of crucial importance. There is a multitude of possibilities and degrees of freedom for optimizing operation.

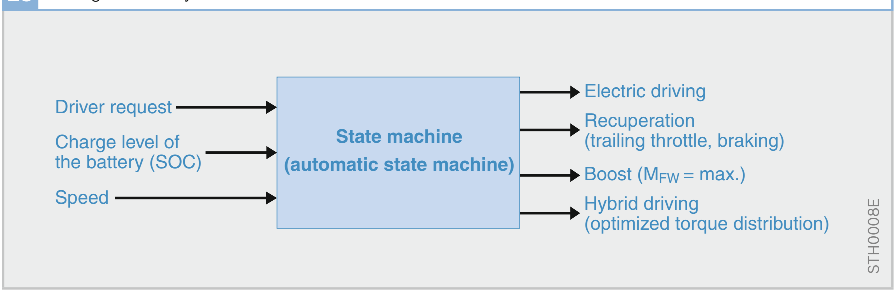
To exploit the fuel-economy potential, in particular distributing the requested drive torque to the drive source of engine and electric motor is hugely important.

Determination in the automatic state machine

Torque distribution is however not necessary in all driving states. Fig. 23 shows the different driving states of a hybrid vehicle which a determined in an automatic state machine by the driver command, the state of the electrical accumulator and the vehicle speed.

In the case of purely electric driving and recuperation the internal-combustion engine is shut down and in boost mode the maximum available torque is requested from both drive sources. Purely electric driving is limited to low vehicle speeds and low accelerations. Recuperation occurs only when the vehicle is decelerating. Boost mode is then used above all when maximum propulsion is requested by the driver (kickdown).

23 Driving states of hybrid vehicles



Distribution by operating strategy

The wide range of hybrid driving in which distribution of the drive torque is to be specified lies between the operating states which are determined by the automatic state machine. On account of the many degrees of freedom and dependencies an optimization is required which can be realized most effectively with the aid of model-based procedures.

Fig. 24 shows the dependencies of the operating strategy. The hybrid control distributes the desired drive torque to the drive sources of engine and electric motor, including in so doing among others the vehicle speed and the state of the electrical accumulator. In addition the operating strategy still requires an equivalence value of the stored electrical energy, which contains how much fuel has been consumed in order to generate this electrical energy.

The different types of electrical energy generation (recuperation and combustion-engine charging) in order to assign to the battery's energy content an equivalence value of the optimization variable (e.g., fuel consumption). This equivalence value provides the basis for deciding which energy is used.

Determining the equivalence value

Determining and optimizing this equivalence value can be done in different ways. The optimum can only be found when the entire driving cycle is known, which amounts to a look into the future (a priori knowledge). However, this look into the future is only possible during specified driving cycles or during simulation. In real driving operation only present and past driving states can be used to determine the equivalence value (a posteriori knowledge). The different optimization horizon for the equivalence value is shown in Fig. 25.

The diagram shows by way of example the speed characteristic of the NEDC driving cycle. Time t = 625 s is taken as the present. Without knowing the full distance, the last long braking from 120 km/h to a stop, which contains a large recuperation potential, cannot be used to optimize the equivalence value.

24 Operating strategy

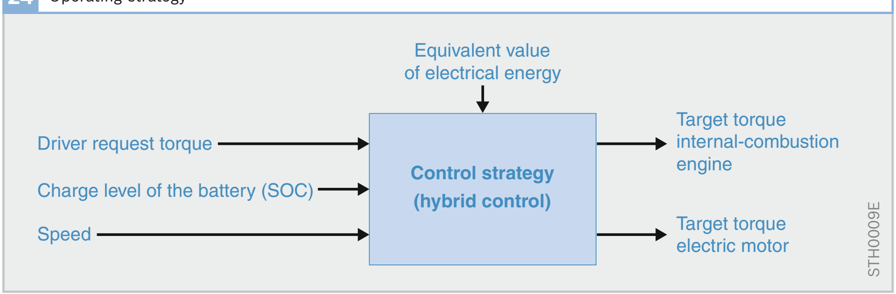
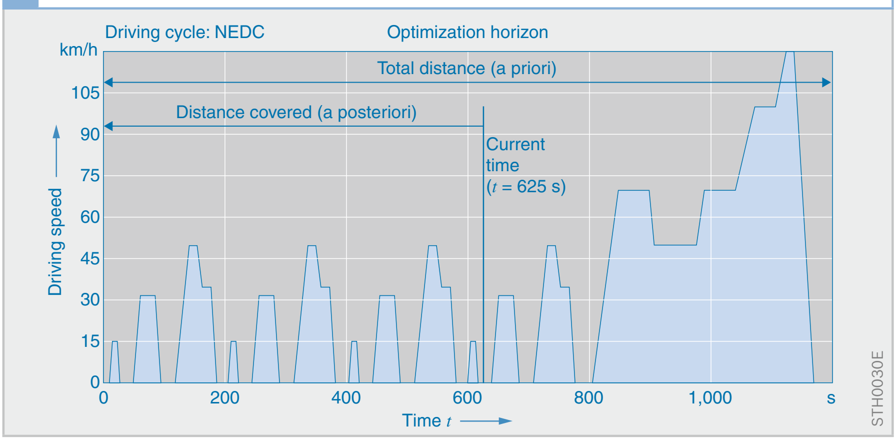


Fig. 26 uses the cumulative fuel consumption to show the different optimization horizons. The consumption of a comparable conventional vehicle is also shown. It can be recognized that an a priori optimization exploits additional potential, since it can among other things utilize the recuperation phase at the end of the cycle.

If the operating-strategy optimization is networked with driver-assistance systems, e.g., with a navigation system, the future driving profile (especially the speed profile) can be estimated down to a certain level.

25 Horizon of different equivalence values



26 Cumulative fuel consumption with different operating strategies

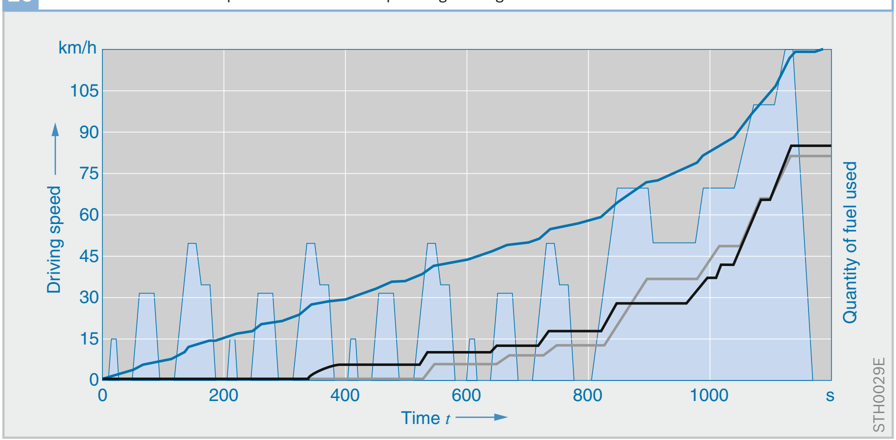


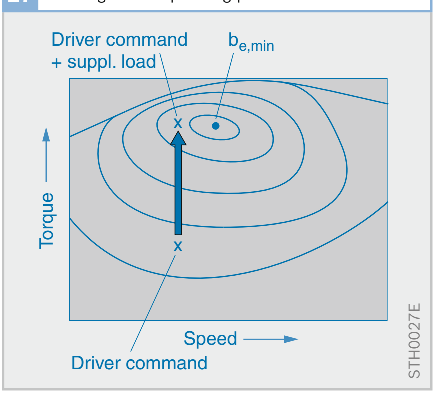
Fig. 26
— Conventional vehicle
— HEV: a priori strategy
— HEV: a posteriori strategy

Strategy of electrical energy generation

In the hybrid vehicle electrical energy can be generated by means of battery charging with the engine and by means of recuperation (recovery of braking energy). Whereas energy is recovered without additional fuel expenditure during recuperation, fuel must be expended to charge the battery with the engine. Here the efficiency of this charging process is dependent on the engine's current operating point.

Because for the most part not enough energy can be generated from recuperation alone and furthermore the storage capability of the battery is limited, charging with the engine cannot be avoided. To keep the amount of fuel to be expended for this purpose as low as possible, this type of power generation is performed then if possible when the engine is operated in operating ranges with poor efficiency and as large an efficiency increase as possible can be achieved by means of the additional load (Fig. 27). Optimum utilization of efficiency improvement when charging with the engine is the function of the operating strategy in that this also involves a distribution of torque between the engine and the electric motor.

27 Shifting of the operating point



Design of the internal-combustion engine

Use of suitable internal-combustion engines

It is basically possible to use any internal-combustion engine from vehicles with conventional drivetrains in a hybrid vehicle. Gasoline, natural-gas and diesel engines can be combined with an electric drive, but with different optimization objectives (see sections *Operating strategy for diesel hybrid vehicles / for gasoline hybrid vehicles*).

Thanks to the additional possibilities provided by the HEV interconnection for example with regard to operating-point shifting, it is also possible if necessary to pursue for hybrid vehicles internal-combustion engine concepts which are different from those for conventionally driven vehicles. Because of the reduced size of the operating range, the necessary efficiency optimization can be limited to this range and high costs for additional components can be avoided. For example, the second turbocharger in modern twin turbocharging concepts can be dispensed with, since its functions (providing for a fast response and for a higher torque at low engine speeds) are covered by the electric motor.

If sufficient power is made available by the electrical accumulator, the electric drive can compensate for torque deficits and a slower response by certain engine concepts.

Dynamic demands placed on the drivetrain are implemented by the combination of an electric drive and an internal-combustion engine. The engine can be relieved of load during dynamic processes thanks to the favorable torque characteristic above all at low engine speeds and the fast response of the electric drive. Load peaks on the engine can therefore be avoided.

Fig. 27
$b_{e,min}$: minimum effective fuel consumption

Atkinson cycle

The previously described altered requirements with regard to maximum power and dynamic response make it possible to use the Atkinson cycle, which cannot be used with a conventional drive because of the lower specific power (due to poor full-load charge) and weakness in dynamic processes. The Atkinson cycle (Fig. 28) necessitates a different stroke/bore ratio of compression and expansion strokes, which geometrically can only be realized with difficulty, but can be represented with the aid of variable valve timing. It offers a better utilization of the expansion phase and thereby increased efficiency.

This concept is implemented, for example, in the Toyota Prius. In addition, the maximum rotational speed is limited here in order to reduce the basic friction of the overall engine through a weaker design of the valve gear. The required maximum rotational speed of the generator can be kept small here.

Downsizing

In addition to using simple or cost-effective internal-combustion engines, specific optimization of the internal-combustion engine in combination with an electric drive delivers advantages. One possibility for improvement is offered by downsizing, which anticipates reducing engine displacement while retaining the power output with the aid of turbocharging. Here,

28 Atkinson cycle

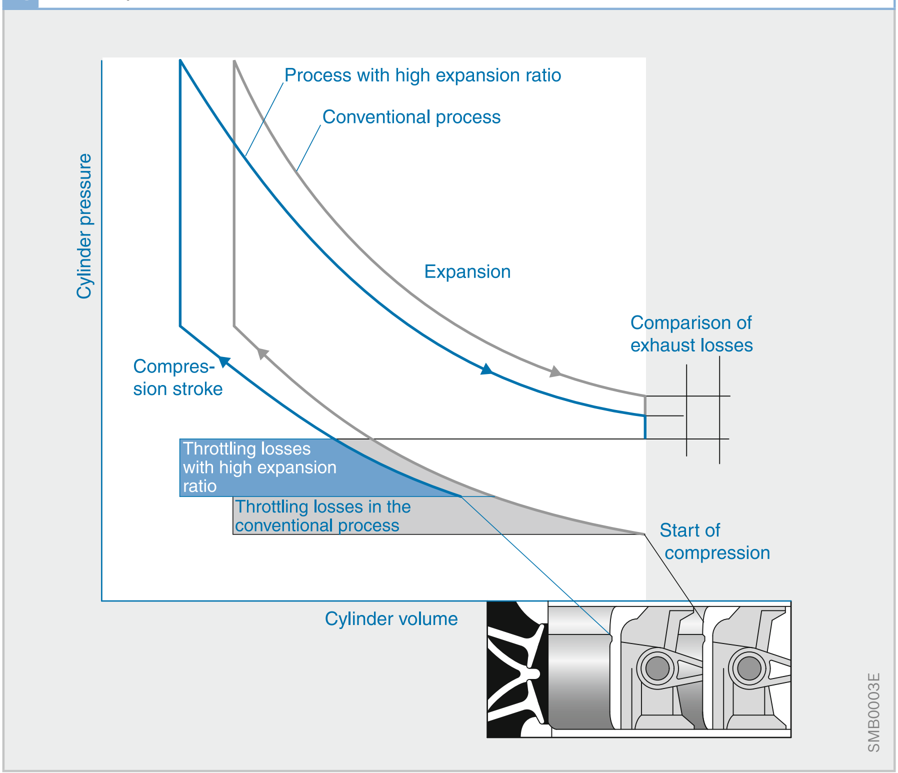


Fig. 28
Source: Toyota

through the use of the electric motor's drive power, unfavorable operating ranges can be avoided and dynamic torque weaknesses compensated for.

In the case of the hybrid vehicle, however, it is also possible with downsizing to tolerate a reduction in engine power, since this can be compensated for with the aid of the electric drive. The overall drive power remains the same here. However, in this case, the maximum power is only available for a limited time (necessitated by the battery charge), which gives rise to a reduced sustained vehicle top speed.

Optimization with regard to emissions and fuel consumption

To achieve the objectives of reduced emissions and consumption, it is possible to exploit degrees of freedom for engine operation which are dependent on the topology of the drivetrain.

An important strategy is to avoid operating points at which the engine demonstrates unfavorable efficiency or high emissions. The underlying operating strategy must be optimized with regard to the improvement objective (e.g., reduced consumption or reduced CO_2 or NO_X). The changed operating conditions can be utilized to optimize the engine concept and exhaust-gas treatment. From the changed requirements there follow changes in functions and in the application of engine management, which are not considered further here.

Friction optimization of the internal-combustion engine

Some primary energy is saved in a hybrid vehicle through the recovery of braking energy (recuperation). This can be performed during both active braking and overrunning, e.g., when driving downhill. To exploit the savings potential as fully as possible, the internal-combustion engine must be shut down in these operating ranges. If this is not possible, the engine must be under coupled motion and its drag friction limits the recuperation potential. In this case, friction optimization of the engine represents an important requirement of the engine.

Regenerative braking system

During regenerative braking kinetic energy of the drive wheels is converted by the electric motor – which is operated as a generator for this purpose – into electrical energy. In this way some of the energy which is normally lost as frictional heat during braking is fed in the form of electric energy to the battery and then utilized. At the same time the generator operation of the electric motor brings about a vehicle-braking effect.

To use a hybrid-drive system to better effect, it is necessary to be able to charge the electrical energy accumulator efficiently. Sufficient electrical energy must be made available for

- the under certain circumstances frequently occurring restarts of the combustion engine at the start/stop system,
- electric torque assistance or electric driving operation in the case of mild- and full-hybrid systems.

On the one hand, the electrical energy for charging the battery can be produced by increasing the engine load and by operating the electric motor as a generator. On the other hand, it is sensible to use the vehicle's kinetic energy during deceleration processes. In conventional vehicles this energy is converted into heat either by the engine drag torque or when the brake pedal is operated by the vehicle service brake.

Through generator use of the electric motor, hybrid vehicles open up the possibility of recovering at least some of the energy and supplying it either to the electrical loads/consumers or to the vehicle's electric drive. This process is known as *regenerative* or *recuperative* braking.

Strategies of regenerative braking

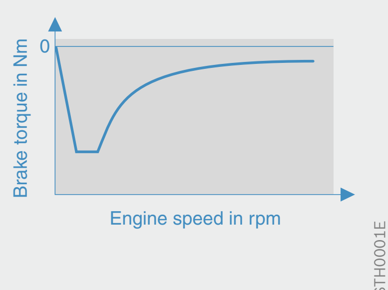
Principle

In the case of a full hybrid, for the purpose of regenerative braking, the internal-combustion engine is decoupled and the drag torque is replaced by an equivalent generator torque of the electric motor (drag-torque simulation). The energy released is stored.

If the engine cannot be decoupled (as in the case of a mild hybrid), alternatively a lower generator torque can be impressed on the drivetrain in addition to the engine drag torque (drag-torque increase).

However, when driveability is taken into consideration, no great decelerations can be implemented through drag-torque simulation or increase. What is problematic is the different regenerative braking torque during the individual braking processes and the resulting different braking power. This must be adapted to the battery state of charge and the thermal load of the electric drive. If, for example, the battery temperature increases significantly after a few brakings, it is necessary under certain circumstances to recover the regenerative power in order to avoid thermal overloading of the system.

1 Maximum torque characteristic during generator operation of the electric motor



K. Reif (Ed.), *Fundamentals of Automotive and Engine Technology*,
DOI 10.1007/978-3-658-03972-1_22, © Springer Fachmedien Wiesbaden 2014

Cooperatively regenerative braking system

The service-brake system must be modified in the case of higher decelerations in order to further exploit the kinetic energy. To this end all or some of the service-brake friction torque must be replaced with regenerative braking torque without the vehicle deceleration changing when the brake-pedal position and force are kept constant. This is realized in a cooperatively regenerative braking system, where vehicle control and braking system interact in such a way that constantly as much friction braking torque is recovered as generator braking torque can be replaced by the electric motor.

Requirements

A cooperatively regenerative braking system must satisfy the following requirements:

- determination of the driver's deceleration command,
- maintenance of the power capability usual for conventional vehicles and operation of the service brake,
- determination of a suitable distribution of braking torque between friction brake and regenerative brake, taking into account stability, comfort and efficiency criteria,
- determination of a suitable distribution of braking torque to the vehicle axles,
- adjustment of the friction braking torque.

Replacing the regenerative braking torque with the friction braking torque calls for a suitable data interface between the corresponding drivetrain ECUs and the brake control unit.

Cooperative braking maneuvers

During a braking operation the maximum generator torque that can be achieved by the electric motor changes over a wide speed range (Fig. 1). This results from the fact that the power output of the electric motor (i.e., torque * speed) is constant in this range. Only at low rotational speeds is there a range of a constant maximum electric-motor torque. If the rotational speed decreases, the achievable regenerative braking torque drops back to zero.

A constant deceleration of the vehicle requires a constant torque at the wheels. If the electric motor operated as a generator is exploited to its limit torque, the friction braking torque must be continuously reduced as vehicle speed decreases, because the generator torque increases. A braking operation of a power-branching hybrid is shown by way of example in Fig. 2. At the beginning of the maneuver the generator torque is increased until it reaches its maximum (if this corresponds to the requested overall braking torque). Towards the end of the braking operation the generator torque is reduced and completely replaced by friction braking torque, because the electric motor cannot deliver any more generator torque at very low rotational speeds (Fig. 1).

2 Cooperative braking maneuver for a power-branching hybrid

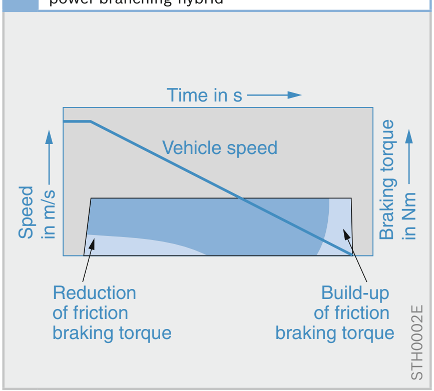


Fig. 2
Reduction and build-up of friction braking torque and replacement by generator braking torque

During a braking operation from high vehicle speed to a stop distribution between friction braking torque and regenerative braking torque is therefore continuously adapted with the pedal constantly actuated.

Braking-force distribution

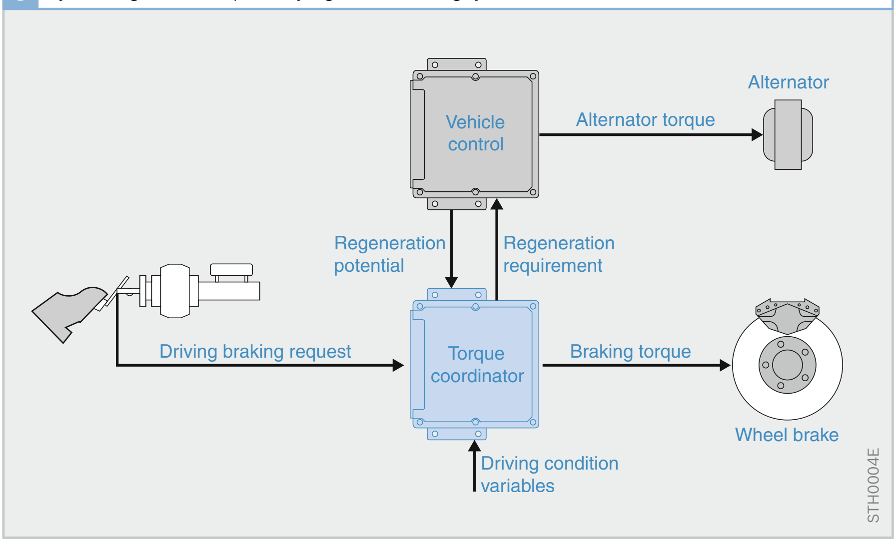
As is the case with the design of conventional braking systems, the braking-force distribution between front and rear axles is of crucial importance to the vehicle's directional stability also in the case of the design of a regenerative braking system. As deceleration increases, the normal force on the front wheels increases while the normal force on the rear wheels decreases.

If the electric motor is connected to the front wheels, a greater wheel torque can be transmitted as deceleration increases and thus normal force increases. For this reason, in order to maintain directional stability, friction-coefficient utilization on the front axle should not exceed friction-coefficient utilization on the rear axle.

If the vehicle is a rear-wheel drive or has an electrified rear axle (i.e., engine on the front axle and electric motor on the rear axle), the offsettable regenerative torque decreases as deceleration increases.

Even the electrical power of the battery has a major influence on the utilization of recuperation, since the limiting element for absorbing electrical energy is from the vehicle motion. As the power of the energy accumulator increases, so the maximum possible purely recuperative deceleration increases. For reasons of vehicle stability, however, a large generator braking torque can only be transmitted to the front axle. High purely recuperative decelerations can therefore only be achieved with a vehicle with front-wheel drive or four-wheel drive. In the case of the latter, the braking torque is distributed to both axles, depending on the configuration of the center differential, in such a way that it corresponds at least approximately to the ideal braking-force distribution of the friction brake.

3 System diagram of a cooperatively regenerative braking system



Influences on stability control

Because the regenerative braking system influences braking stability, ABS and ESP control operations must be adapted to the altered driving physics.

It is advisable to suppress the regenerative component of braking when unstable driving states or excessively high brake slip are detected and to represent deceleration and stabilization interventions alone by the friction-brake system. Otherwise instabilities and drivetrain oscillations could disrupt an optimum wheel-slip control operation.

Interventions of the vehicle stabilization system over the vehicle's life are so rare in proportion to the partial braking operations that suppressing recuperative braking in these situations does not have any noticeable influence on the vehicle's average consumption.

Implementation of the service brake

The friction brake of the cooperatively regenerative braking system can be represented in different configurations. It is primarily mechatronic braking systems which decouple the brake pedal and wheel brake and represent the brake-pedal characteristic by adding a pedal simulator. Here the energy for brake boosting can be stored hydraulically, pneumatically or electrically.

Common to all implementations is the block diagram of the cooperatively regenerative braking system (Fig. 3). The vehicle control continuously monitors all the relevant parameters of the hybrid drive and determines which torque the electric motor can make available for braking. When the brake pedal is actuated the torque coordinator of the brake control unit calculates a distribution of the braking torque to the friction brake and the recuperative braking system.

The recuperative torque component is signaled back to the vehicle control and from there forwarded to the electric-motor actuator. The residual braking torque is adjusted by the friction brake while vehicle stability and wheel slip are monitored.

Workshop technology

More than 30,000 garages/workshops around the world are equipped with workshop technology, i.e. test technology and workshop software from Bosch. Workshop technology is becoming increasingly important as it provides guidance and assistance in all matters relating to diagnosis and troubleshooting.

Workshop business

Trends

Many factors influence workshop business. Current trends are, for example:

- The proportion of diesel passenger cars is rising
- Longer service intervals and longer service lives of automotive parts mean that vehicles are being checked into workshops less frequently
- Workshop capacity utilization in the overall market will continue to decline in the next few years
- The amount of electronic components in vehicles is increasing – vehicles are becoming "mobile computers"
- Internetworking of electronic systems is increasing, diagnostic and repair work covers systems which are installed and networked in the entire vehicle
- Only the use of the latest test technology, computers and diagnostic software will safeguard business in the future

Consequences

Requirements

Workshops must adapt to the trends in order to be able to offer their services successfully on the market in the future. The consequences can be derived directly from the trends:

- Professional fault diagnosis is the key to professional repairs
- Technical information is becoming the crucial requirement for vehicle repairs
- Rapid availability of comprehensive technical information safeguards profitability

1 Diagnosis on a vehicle with a diagnostic tester

K. Reif (Ed.), *Fundamentals of Automotive and Engine Technology*,
DOI 10.1007/978-3-658-03972-1_23, © Springer Fachmedien Wiesbaden 2014

- The need for workshop personnel to be properly qualified is increasing dramatically
- Investment by workshops in diagnosis, technical information and training is essential

Measurement and test technology

The crucial step for workshops to take is to invest in the right test technology, diagnostic software, technical information, and technical training in order to receive the best possible support and assistance for all the jobs and tasks in the workshop process.

Workshop processes

The essential tasks which come up in the workshop can be portrayed in processes. Two distinct subprocesses are used for handling all tasks in the service and repair fields. The first subprocess covers the predominantly operations and organization-based activity of *job order acceptance*, while the second subprocess covers the predominantly technically based work steps of *service* and *repair implementation.*

Job order acceptance

When a vehicle arrives in the workshop, the job order acceptance system's database furnishes immediate access to all available information on the vehicle. The moment the vehicle enters the shop, the system provides access to its entire history. This includes all service work and repairs carried out on the vehicle up to that point. Furthermore, this sequence involves the completion of all tasks relating to the customer's request, its basic feasibility, scheduling of completion dates, provision of resources, parts and working materials and equipment, and an initial examination of the task and extent of work involved. Depending on the process objective, all subfunctions of the ESI[tronic] product are used within the framework of the *service acceptance* process.

Service and repair implementation

Here, the jobs defined within the framework of the job order acceptance are carried out. If it is not possible to complete the task in a single process cycle, appropriate repeat loops must be provided until the targeted process result is achieved. Depending on the process objective, all subfunctions of the ESI[tronic] product are used within the framework of the service and repair implementation process.

2 Workshop processes

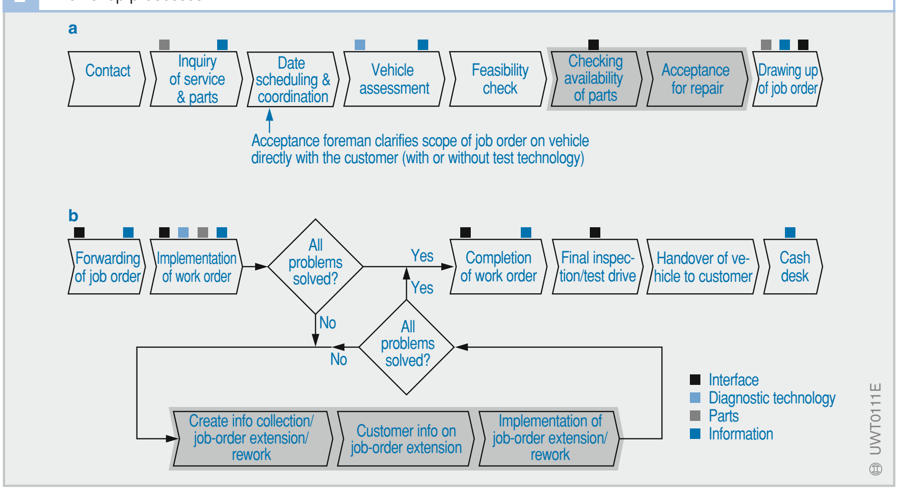


Figure 2
a Job order acceptance
b Service and repair implementation

Electronic service information (ESI[tronic])

System functions for supporting the workshop process

ESI[tronic] is a modular software product for the automotive engineering trade. The individual modules contain the following information:

- Technical information on spare parts and automotive equipment
- Exploded views and parts lists for spare parts and assemblies
- Technical data and setting values
- Flat rate units and times for work on the vehicle
- Vehicle diagnosis and vehicle system diagnosis
- Troubleshooting instructions for different vehicle systems
- Repair instructions for vehicle components, e.g. diesel power units
- Electronic circuit diagrams
- Maintenance schedules and diagrams
- Test and setting values for assemblies
- Data for costing maintenance, repair and service work

Application

The chief users of ESI[tronic] are motor garages/workshops, assembly repairers and the automotive parts wholesale trade. They use the technical information for the following purposes:

- Motor garages/workshops: mainly for diagnosis, service and repair of vehicle systems
- Assembly repairers: mainly for testing, adjusting and repair of assemblies
- Automotive parts wholesale trade: mainly for parts information

Garages/workshops and assembly repairers use this parts information in addition to diagnosis, repair and service information. Product interfaces enable ESI[tronic] to network with other (particularly commercial) software in the workshop environment and the automotive parts wholesale trade in order, for instance, to exchange data with the accounting merchandise information system.

Benefit to the user of ESI[tronic]

The benefit of using ESI[tronic] lies in the fact that the system furnishes a large amount of information which is needed to conduct and safeguard the business of motor garages/workshops. This is made possible by the broadly conceived and modular ESI[tronic] product program. The information is offered on one interface with a standardized system for all vehicle makes.

Comprehensive vehicle coverage is important for workshop business in that the necessary information is always to hand. This is guaranteed by ESI[tronic] because country-specific vehicle databases and information on new vehicles are incorporated in the product planning. Regular updating of the software offers the best opportunity of keeping abreast of technical developments in the automotive industry.

3 ESI[tronic] workshop software for all vehicle makes

Vehicle system analysis (FSA)

Vehicle system analysis (FSA) from Bosch offers a simple solution to complex vehicle diagnosis. The causes of a problem can be swiftly located thanks to diagnosis interfaces and fault memories in the on-board electronics of modern motor vehicles. The *component testing* facility of FSA developed by Bosch is very useful in swiftly locating a fault: The FSA measurement technology and display can be adjusted to the relevant component. This enables this component to be tested while it is still installed.

Measuring equipment

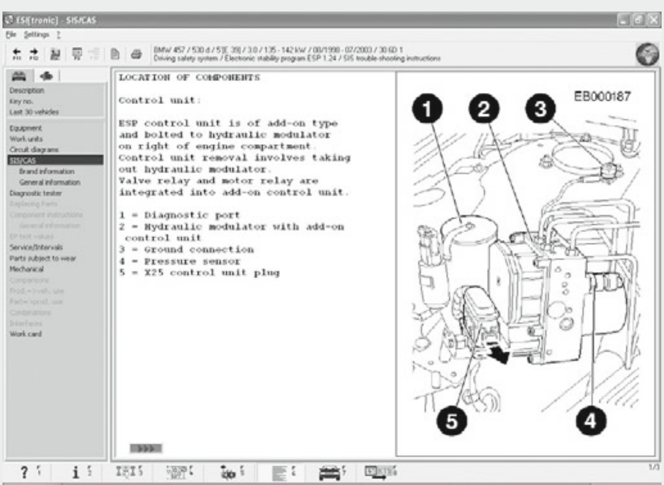
Workshop personnel can choose from various options for diagnosis and trouble-shooting: the high-performance, portable KTS 650 system tester or the workshop-compatible KTS 520 and KTS 550 KTS modules in conjunction with a standard PC or laptop. The modules have an integrated multimeter, and KTS 550 and KTS 650 also have a 2-channel oscilloscope. For work applications on the vehicle, ESI[tronic] is installed in the KTS 650 or on a PC.

Example of the sequence in the workshop

The ESI[tronic] software package supports workshop personnel throughout the entire vehicle repair process A diagnosis interface allows ESI[tronic] to communicate with the electronic systems within the vehicle, such as the ESP electronic control unit. Working at the PC, the technician starts by selecting the SIS (service information system) utility to initiate diagnosis of on-board control units and access the ECU's fault memory.

The diagnostic tester provides the data needed for direct comparisons of specified results and current readings, without the need for supplementary entries. ESI[tronic] uses the results of the diagnosis as the basis for generating specific repair instructions. The system also provides displays with other information, such as component locations, exploded views of assemblies, diagrams showing the layouts of electrical, pneumatic and hydraulic systems etc. Working at the PC, users can then proceed directly from the exploded views to the parts lists with part numbers to order the required replacement components. All service procedures and replacement components are recorded to support the billing process. After the final road test, the bill is produced simply by pressing a few keys. The system also provides a clear and concise printout with the results of the vehicle diagnosis. This offers the customer a full report detailing all of the service operations and materials that went into the vehicle's repair.

4 ESI[tronic]: illustration of the installation position of the ESP hydraulic modulator



Diagnostics in the workshop

The function of these diagnostics is to identify the smallest, defective, replaceable unit quickly and reliably. The guided troubleshooting procedure includes onboard information and offboard test procedures and testers. Support is provided by electronic service information (ESI[tronic]). Instructions for further troubleshooting are provided for a wide variety of possible problems (for example, ESP intervenes prematurely due to variant encoding) and faults (such as no signal from speed sensor).

Guided troubleshooting

The main element is the guided troubleshooting procedure. The workshop employee is guided by a symptom-dependent, event-controlled procedure, which initiates with the symptom (vehicle symptom or fault memory entry). Onboard (fault memory entry) and offboard facilities (actuator diagnostics and offboard testers) are used.

The guided troubleshooting, readout of the fault memory, workshop diagnostic functions and electrical communication with offboard testers take place using PC-based diagnostic testers. This may be a specific workshop tester from the vehicle manufacturer or a universal tester (e.g. KTS 650 by Bosch).

Reading out fault memory entries

Fault information (fault memory entries) stored during vehicle operation are read out via a serial interface during vehicle service or repair in the customer service workshop.

Fault entries are read out using a diagnostic tester. The workshop employee receives information about:

- Malfunctions (e.g. engine temperature sensor)
- Fault codes (e.g. short circuit to ground, implausible signal, static fault)
- Ambient conditions (measured values on fault storage, e.g. engine speed, engine temperature etc.).

Once the fault information has been retrieved in the workshop and the fault corrected, the fault memory can be cleared again using the tester.

A suitable interface must be defined for communication between the control unit and the tester.

Actuator diagnostics

The control unit contains an actuator diagnostic routine in order to activate individual actuators at the customer service workshop and test their functionality. This test mode is started using the diagnostic tester and only functions when the vehicle is at a complete stop below a specific engine speed, or when the engine is switched off. This allows an acoustic (e.g. valve clicking), visual (e.g. flap

1 Flowchart of a guided troubleshooting procedure with CAS[plus]

Identification

Troubleshooting based on customer claim

Read out and display fault memory

Start component testing from fault code display

Display SD actual values and multimeter actual values in component test

Setpoint/actual value comparison allows fault definition

Perform repair, define parts, circuit diagrams etc. in ESI[tronic]

Renew defective part

Clear fault memory

Fig. 1
The CAS[plus] system (**c**omputer **a**ided **s**ervice) combines control unit diagnosis with SIS troubleshooting instructions for even more efficient troubleshooting. The decisive values for diagnostics and repair then appear immediately on screen.

movement), or other type of inspection, e.g. measurement of electric signals, to test actuator function.

Workshop diagnostic functions

Faults that the on-board diagnosis fails to detect can be localized using support functions. These diagnostic functions are implemented in the ECU and are controlled by the diagnostic tester.

Workshop diagnostic functions run automatically, either after they are started by the diagnostic tester, or they report back to the diagnostic tester at the end of the test, or the diagnostic tester assumes runtime control, measured data acquisition, and data evaluation. The control unit then implements individual commands only.

Example

The assignment test checks that the electronic stability program (ESP) activates the wheel brake cylinders of the correct wheels. For this test, the vehicle is driven into the brake tester. After the technician starts the function, the diagnostic tester indicates how to proceed. After the brake pedal is activated, individual channels of the ESP hydraulic modulator are brought, one after another, to the pressure drop position. This allows a determination to be made of whether the corresponding wheel can be rotated. The diagnostic tester indicates the wheel for which the system has reduced the brake pressure. In this way, it can be determined whether the circuitry of the hydraulic modulator and wheel brake cylinders is correct.

Offboard tester

The diagnostic capabilities are expanded by using additional sensors, test equipment, and external evaluators. In the event of a fault detected in the workshop, offboard testers are adapted to the vehicle.

2 Functions of the KTS 650

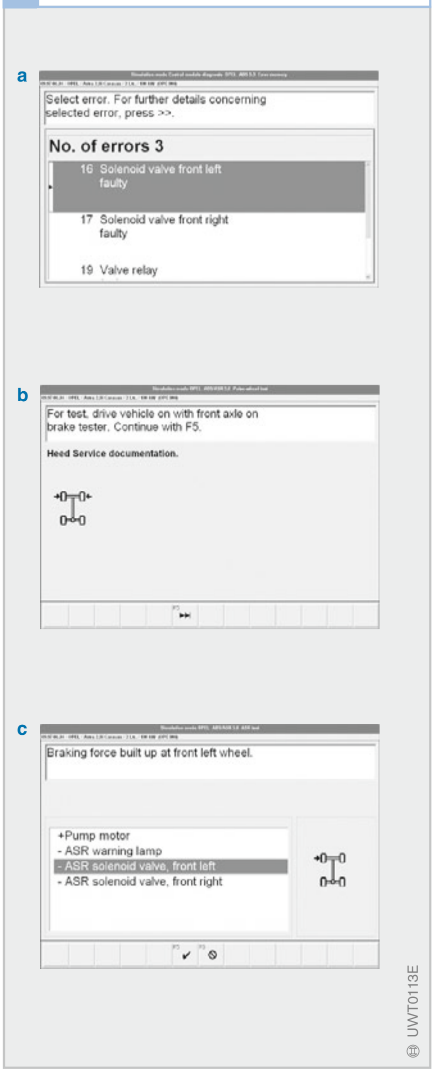


Fig. 2
- a Display of the fault memory contents
- b Procedure instructions for workshop diagnostic functions
- c Check of pressure maintenance function

Testing equipment

Effective testing of the system requires the use of special testing equipment. While earlier electronic systems could be tested with basic equipment such as a multimeter, ongoing advances have resulted in electronic systems that can only be diagnosed with complex testers.

The system testers of the KTS series are widely used in workshops. The KTS 650 (Fig. 1) offers a wide range of capabilities for use in the vehicle repairs, enhanced in particular by its graphical display of data such as test results. These system testers are also known as diagnostic testers.

Functions of the KTS 650

The KTS 650 offers a wide variety of functions, which are selected by means of buttons and menus on the large display screen. The list below details the most important functions offered by the KTS 650.

Identification

The system automatically detects the connected ECU and reads actual values, fault memories and ECU-specific data.

Reading/erasing the fault memory

The fault information detected during vehicle operation by on-board diagnosis and stored in the fault memory can be read with the KTS 650 and displayed on screen in plain text.

Reading actual values

Current values calculated by the ECU can be read out as physical values (e.g. wheel speeds in km/h).

Actuator diagnostics

The electrical actuators (e.g. valves, relays) can be specifically triggered for function testing purposes.

1 KTS Series testing equipment

Fig. 1
- a Multimedia-capable, mobile KTS 650 diagnostic tester
- b Universal, convenient solution for vehicle workshops; KTS 550 in conjunction with PC or laptop
- c Universal solution for vehicle workshops; KTS 520 in conjunction with PC or laptop

Test functions
The diagnostic tester triggers programmed test procedures in the ECU. These allow testing of whether the channels of the ABS hydraulic modulator are correctly assigned to the wheel brake cylinders.

Multimeter function
Electrical current, voltage and resistance can be tested in the same way as with a conventional multimeter.

Time graph display
The continuously recorded measured values are displayed graphically as a signal curve, as with an oscilloscope (e.g. signal voltage of the wheel speed sensors).

Additional information
Specific additional information relevant to the faults/components displayed can also be shown in conjunction with the electronic service information (ESI[tronic]) (e.g. troubleshooting instructions, location of components in the engine compartment, test specifications, electrical circuit diagrams).

Printout
All data (e.g. list of actual values or document for the customer) can be printed out on standard PC printers.

Programming
The software of the ECU can be encoded using the KTS 650 (e.g. variant coding of the ESP ECU).

The extent to which the capabilities of the KTS 650 can be utilized in the workshop depends on the system to be tested. Not all ECUs support its full range of functions.

2 Functions of the KTS 650

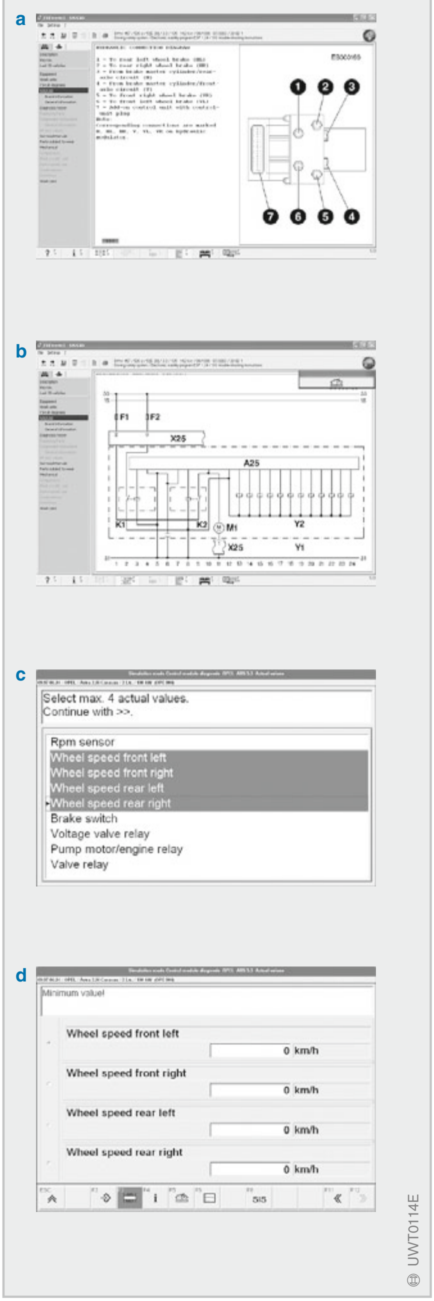


Fig. 2
- a Hydraulic connection diagram of the hydraulic modulator
- b Electrical connection diagram of the hydraulic modulator
- c Selection for measuring actual values
- d Measuring the wheel speeds

Brake testing

Inspection and maintenance

The condition of a vehicle's braking system directly affects its safety as well as that of its occupants and/or the goods it is transporting. That is why the servicing of the braking system is such an important part of the care and maintenance of a vehicle.

Transport legislation requires that vehicle braking systems are inspected at regular intervals. Manufacturers' authorized dealerships, or approved independent workshops and brake repair services (such as Bosch Service) carry out inspection, maintenance and, where necessary, repairs of the brake system.

In Germany, vehicle owners or custodians must present their vehicles for inspection at an officially approved testing center at regular intervals and at their own expense. In Germany, for instance, the last month by which the vehicle must file for a major roadworthiness inspection is indicated by a special check tag on the vehicle's rear license plate.

The natural wear of brake system components such as the brake pads demands that the system is regularly serviced outside of the statutory inspections.

In addition to checking the effectiveness of the brakes on a brake tester, servicing should involve regular assessment and maintenance of the following components:
- brake pads and/or brake shoes,
- brake disks, and
- brake drums.

On hydraulic braking systems, the following must also be regularly checked and serviced:
- the master cylinder,
- wheel brake cylinders,
- the brake hoses,
- the brake lines,
- the brake fluid level and
- the brake fluid condition.

Other units, such as the brake booster, brake force distributor, brake force limiter etc. are frequently maintenance-free.

For compressed-air braking systems, the following also need to be checked:
- air compressors,
- compressed-air cylinders,
- antifreeze unit,
- valves, cylinders,
- pressure regulators,
- braking force regulators,
- coupling heads and
- the air-tightness of the entire system.

Brake pads and shoes

The brake shoes and pads are the parts of the braking system that are subject to the greatest wear as the retardation of the vehicle is achieved by pressing the shoes/pads against the rotating drums/disks. Proper maintenance of these components is absolutely essential for the safety of a braking system.

Checking wear

Assuming they have been correctly fitted, the rate at which brake pads/shoes wear is dependent on the properties of the friction material (e.g. its frictional coefficient), the manner in which the vehicle is driven and the loads it carries.

On most vehicles, reliable checking of the brake pad wear on disk brakes requires the removal of the wheels. Attempting to assess the level of wear with the wheels in place risks inaccurate conclusions.

Checking the wear of brake shoes on drum brakes generally involves removing not only the wheels but also the brake drums.

On some more modern vehicles, inspection holes allow the brake shoe wear to be checked without the brake drums having to be removed, although they are inadequate for a thorough inspection of overall brake shoe condition.

Adjustment
There is normally a small gap (clearance) between the brake pad/shoe and the disk/drum that prevents continuous abrasion of the friction material against the disk or drum. As the friction material wears, that gap becomes larger and, in the case of drum brakes, necessitates regular readjustment of the shoes (assuming the brakes do not incorporate a self-adjusting mechanism).

Disk brakes with an integral parking brake mechanism automatically readjust themselves.

Straightforward disk brakes are likewise self-adjusting. This means that the brake pads automatically shift to take up the extra gap as they wear so that in effect the clearance between the pad and the disk never changes.

The need for readjustment of the brake shoes on drum brakes without a self-adjusting mechanism can be detected by the amount of free play when pressing the brake pedal.

If, for different brake systems (such as simplex or duplex brakes), the brake shoes are adjusted, the information from the brake manufacturer must be observed.

Nevertheless, the following basic principles will always apply:
Regardless of the type of drum brake, the brakes on both sides must always be adjusted at the same time. On vehicles with drum brakes all around, all four brakes must be adjusted at once.

The brakes must be cold before they are adjusted. The service brakes should be adjusted before the handbrake.

Replacing brake pads and shoes
Disk brake pads have to be replaced when the thickness of the friction material is worn down to 2 mm.

On systems with wear sensors on the brake pads, a warning lamp on the instrument panels indicates to the driver that the pads are in need of imminent replacement as soon as the remaining thickness is down to 3.5 mm.

On drum brakes the brake shoe friction lining thickness must not be less than 1.5 mm on cars and 4 mm on commercial vehicles. If the shoes are unevenly worn, or if the linings are cracked or chipped, they too must be replaced.

When replacing brake pads or shoes, it is important that the new pads/shoes conform to the specifications of the original equipment manufacturer.

Important!
The use of brake pads/shoes that do not match the specifications of the brake manufacturer may render the vehicle's insurance policy void.

Brake pads, disks, shoes and drums must always be replaced on both sides (i.e. both front or both rear wheels) at the same time, as otherwise the vehicle may "pull" to one side under braking.

Brake disks and drums
Brake disks and drums are made of steel or cast iron and therefore do not wear as quickly as the pads and shoes. Nevertheless, they still have to be maintained at regular intervals.

The contact surfaces of the brake disks and brake drums must be checked for:
- striations,
- cracks,
- corrosion,
- abrasion and
- differences of thickness.

For disk and drum brakes, these defects can be identified with the naked eye during a visual check.

Brake disks can also develop excessive runout or warping. The degree of runout at the outer edge of the disk must not exceed 0.2 mm and has to be checked using a dial gauge. Brake disks with more than the allowable runout must be replaced.

If scored or unevenly worn brake disks are reground, they must not be reduced to more than a minimum permissible thickness.

Brake drums may become misshapen (so that they are no longer perfectly circular) or develop hairline cracks. Loss of circularity is caused by overheating. It can be detected by pulsating feedback from the brake pedal or, of course, on a brake tester. Brake drums can be reground provided the degree of wear or damage is not excessive. When doing so, the maximum allowable internal drum diameter for the particular vehicle must not be exceeded. If the degree or nature of the damage is such that regrinding the drums is not possible, the only option is to replace them. Drums must always be reground or replaced on both sides (both front or both rear wheels) at the same time in order to ensure even braking.

Master cylinder

The wearing parts of the master cylinder are primarily the cup seals, which are made of a special rubber compound. They are responsible for creating the seal between the piston and the cylinder wall. Corrosion, which can develop as a result of water absorption by the brake fluid, causes pitting of the cylinder wall. That roughness then damages the piston seals by abrading them so that they start to leak.

Depending on the severity of the problem, this can result in partial or even total loss of brake pressure. The response of the brake pedal when depressed will indicate whether the primary or the isolating seal is leaking.

1 Testing brakes on a brake tester

Wheel brake cylinders

As with the master cylinder, the cup seals in the wheel brake cylinders are subject to wear. They can similarly develop leaks and cause corrosion on the cylinder walls. In addition, wheel brake cylinders can also develop leaks around the sealing caps. This can lead to brake fluid contaminating the brake pads/linings and reducing brake efficiency.

The following checks can be carried out to test the condition of the seals:

Low-pressure test

A pressure gauge is connected to the wheel brake cylinder and a pressure of 2 to 5 bar is applied and maintained using a special pedal positioner. There must be no drop in pressure for a period of 5 minutes.

High-pressure test

A pressure of 80-100 bar is applied. Over a period of 10 minutes, the pressure may not drop by more than 10% of its original level.

Pilot pressure test

The pedal positioner is removed and the pressure drops back to the pilot level (if applicable; only applies to cylinders with cup seals) of 0.4-1.7 bar. The pressure should not fall below 0.4 bar over a period of five minutes.

Brake hoses and lines

In theory, brake pipes and hoses are maintenance-free. Nevertheless, they are subject to environmental effects such as corrosion due to water and salt and impact damage from stones, grit and gravel.

Because of those factors, brake pipes and hoses should be regularly inspected. Brake lines should primarily be checked for corrosion, while the hoses should be inspected for abrasion and splits. The unions should be checked for leaks.

Brake fluid level and condition

The brake fluid level is checked on the brake fluid reservoir. The fluid level should be between the “MAX” and “MIN” marks. This check provides one means of detecting whether there are leaks in the braking system. If the fluid level is at or below the “MIN” mark, the system should be checked for leaks. On some vehicles, a warning lamp on the instrument panel indicates to the driver that the fluid level is approaching the minimum mark.

As brake fluid can absorb water by diffusion through the brake hoses, it should be completely replaced every one to two years.

This is absolutely essential for the safety of the braking system.

[1]) **Caution:**
If the level of fluid in the reservoir is very low, simply adding more fluid must on no account be viewed as the solution. The cause of the fluid loss must be established and rectified. Dark or cloudy brake fluid must be replaced immediately.

Maintenance checklist

The components of hydraulic brake systems are subjected to considerable stresses. Heat, cold and vibration can all lead to material fatigue in the course of time. Splash water, especially salt water, and dirt cause corrosion and diminish the ability of components and mechanisms to operate smoothly. Consequently, impairment of function can result.

For safety reasons, therefore, specific regular checks and maintenance work are absolutely essential.

The best time for carrying out such work is at the end of the winter season because the exposed components of the brake system are subjected to the most extreme weather conditions in the winter.

The checks and maintenance operations include

- visual inspections
- function checks
- leakage tests
- internal examination of brakes
- efficiency tests.

This maintenance checklist details the various components in alphabetical order and indicates the checks and tests required for each one. The abbreviations used are explained below.

Key to abbreviations:

A Remove
E Fit
F Lubricate
G Restore function
I Repair
N Replace/renew
NA Rework
P Check, assess
R Clean
S Adjust/align/correct

Maintenance tasks	
Brake fluid reservoir [1])	
Cap	P/N
Reservoir	P/R/N
Attachment	P/I
Warning lamp switch (if present)	P/I/N
Brake fluid	
Level	P/S
Appearance, color	P/N
Moisture content	P/S
Handbrake lever (parking brake)	
Travel, no. of ratchet notches	P/S
Ratchet function	P/I
Freedom of action	P/G/F
Lever stop (if present)	P/S/I
Return spring (if present)	P/S/F
Braking force limiter	
External damage	P/N
Attachment	P/I/N
Pipe connections	P/I/N
Function	P/N
Limited pressure (observe testing conditions)	P/S
Braking force regulator	
External damage	P/N
Attachment	P/I/N
Pipe connections	P/I/N
Linkage, lever	P/I/F
Travel spring	P/N/F
Function	P/N
Limited pressure (observe testing conditions)	P/S
Brake servo unit	
External damage	P/N
Attachment	P/I
Hoses (splits etc.)	P/N
Function	P/N
Leakage	P/N
Brake pedal (service brakes)	
Pedal	P
Pedal rubber (wear, condition)	P/N
Pedal travel	P/S
Connecting rod play	P/S
Freedom of action of shaft	P/G/F
Pedal stop	P/S/I
Pedal return spring	P/S/F

Brake lines [2]	
External damage	R/P/N
Attachment	P/I/N
Corrosion	P/N
Brake hoses	
External damage	P/N
Attachment	P/N
Kinking, length	P/N
Routing (e. g. twisting)	P/I/N
Suitability for pressure medium	P/N
Age	P/N
Master cylinder	
External damage	P/N
Attachment	P/I/N
Pipe connections	P/I/N
Seal against brake servo unit	P/I/N
Low-pressure seal	P/I/N
High-pressure seal	P/I/N
Brake light switch	P/N
Brake lights	P/I
Brakes (general)	
Basic adjustment of drum brakes	P/S
Clearance adjustment on disk brakes	P/S
Non-return valve	
External damage	P/N
Attachment	P/I
Hoses (splits etc.)	P/N
Function	P/N
Leakage	P/N
Disk brakes (brake pads)	
Damage (cracks etc.)	P/N
Shining, hardening etc.	P/N
Friction pad thickness [3]	P/N
Pad guides	P/R/F
Suitability for vehicle	P/N
Disk brakes (brake caliper)	
External damage	R/P/N
Attachment	R/I/N
Brake pad channels	P/R
Guides	P/G/F
Piston freedom	P/I/N
Piston position	P/S
Dust seals	P/N
Small parts (expander springs, bolts etc.)	P/N
Bleed valve, dust cap	P/G/N

Disk brakes (brake disks)	
Damage (cracks etc.)	P/N
Thermal overload	P/N
Wear [4]	P/N
Wear pattern [4]	P/NA/N
Minimum thickness [4]	P/NA/N
Runout [4]	P/NA/N
Drum brakes (general)	
Backplate (damage)	R/P
Wheel brake cylinders	P/I/N
Dust seals	P/I/N
Parking brake mechanism and linkage	P/I/F
Adjusting mechanism	P/G/F
Handbrake cable	P/F
Brake shoes and linings	P/I/N
Shoe anchor bearings	R/F
Return springs	P/N
Drum brakes (handbrake cable, linkage)	
External damage (cable sheath)	P/N
Attachment	P/I/N
Correct routing and fitting	P/I/N
Guides, rollers etc.	P/I/F
Cable (fraying etc.)	P/N
Freedom of action	P/G/F
Adjusting mechanism	P/G/F
Basic adjustment	P/S
Drum brakes (brake drum)	
Damage (cracks etc.)	P/N
Thermal overload	P/N
Wear [5]	P/NA/N
Concentricity [5]	P/NA/N
Warping [5]	P/NA/N
Service brake efficiency test	
Braking force, front wheels	P/I
Braking force difference (front)	P/I
Braking force, rear wheels	P/I
Braking force difference (rear)	P/I
Actuating force	P/I
Parking brake efficiency test	
Braking force	P/I
Braking force difference	P/I

[2] **Caution:**
Do not use abrasive materials or tools on coated brake lines. Corroded or damaged lines must be replaced.

[4] **Caution:**
Refer to maximum wear limits.

[5] **Caution:**
Refer to maximum wear limits.

[3] **Caution:**
Minimum thickness for disk brake pads is 2 mm, excluding backplate.

Fuel-injection pump test benches

Accurately tested and precisely adjusted fuel-injection pumps and governor mechanisms are key components for obtaining optimized performance and fuel economy from diesel engines. They are also crucial in ensuring compliance with increasingly strict exhaust-gas emission regulations. The fuel-injection pump test bench (Fig. 1) is a vital tool for meeting these requirements.

The main specifications governing both test bench and test procedures are defined by ISO standards; particularly demanding are the specifications for rigidity and geometrical consistency in the drive unit (5).

As time progresses, so do the levels of peak pressure that fuel-injection pumps are expected to generate. This development is reflected in higher performance demands and power requirements for pump test benches. Powerful electric drive units, a large flyweight and precise control of rotational speed guarantee stability at all engine speeds. This stability is an essential requirement for repeatable, mutually comparable measurements and test results.

Flow measurement methods

An important test procedure is to measure the fuel pumped each time the plunger moves through its stroke. For this test, the fuel-injection pump is clamped on the test bench support (1), with its drive side connected to the test bench drive coupling. Testing proceeds with a standardized calibrating oil at a precisely monitored and controlled temperature. A special, precision-calibrated nozzle-and-holder assembly (3) is connected to each pump barrel. This strategy ensures mutually comparable measurements for each test. Two test methods are available.

Glass gauge method (MGT)

The test bench features an assembly with two glass gauges (Fig. 2, 5). A range of gages with various capacities are available for each cylinder. This layout can be used to test fuel-injection pumps for engines of up to 12 cylinders.

1 Bosch fuel-injection pump test bench with electronic test system (KMA)

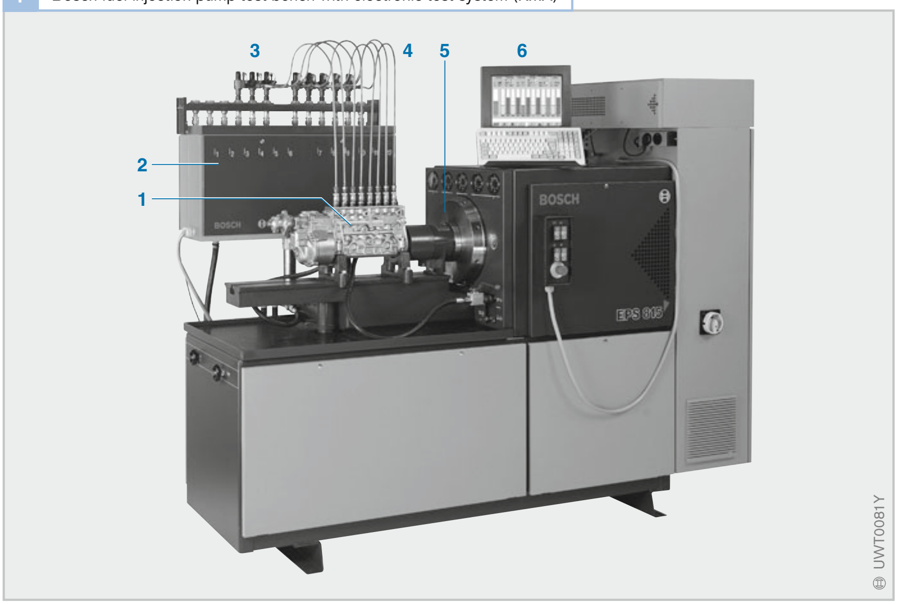
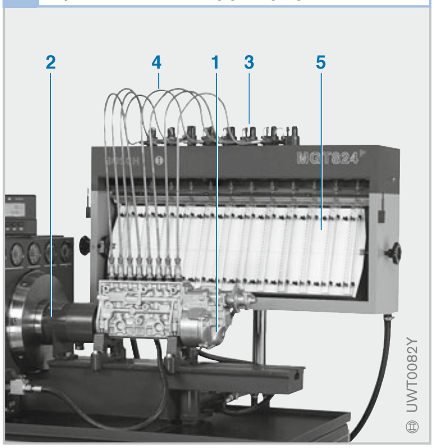


Fig. 1
1 Fuel-injection pump on test bench
2 Quantity test system (KMW)
3 Test nozzle-and-holder assembly
4 High-pressure test line
5 Electric drive unit
6 Control, display and processing unit

2 Layout of test stand using glass-gauge methods (MGT)



3 Measurement cell concept (KMA)

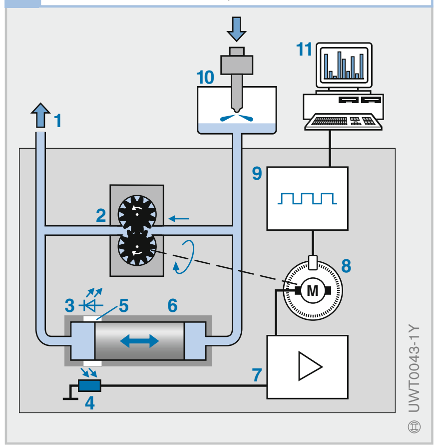


Fig. 2
1 Fuel-injection pump
2 Electric drive unit
3 Test nozzle-and-holder assembly
4 High-pressure test line
5 Glass gages

Fig. 3
1 Return line to calibrating oil tank
2 Gear pump
3 LED
4 Photocell
5 Window
6 Plunger
7 Amplifier with electronic control circuitry
8 Electric motor
9 Pulse counter
10 Test nozzle-and-holder assembly
11 Monitor (PC)

In the first stage, the discharged calibrating flows past the glass gages to return directly to the oil tank. As soon as the fuel-injection pump reaches the rotational speed indicated in the test specifications, a slide valve opens, allowing the calibrating oil from the fuel-injection pump to flow to the glass gages. Supply to the glass containers is then interrupted when the pump has executed the preset number of strokes.

The fuel quantity delivered to each cylinder in cm^3 can now be read from each of the glass gages. The standard test period is 1,000 strokes, making it easy to interpret the numerical result in mm^3 per stroke of delivered fuel. The test results are compared with the setpoint values and entered in the test record.

Electronic flow measurement system (KMA)

This system replaces the glass gauges with a control, display and processor unit (Fig. 1, 6). While this unit is usually mounted on the test bench, it can also be installed on a cart next to the test bench.

This test relies on continuous measuring the delivery capacity (Fig. 3). A control plunger (6) is installed in parallel with the input and output sides of a gear pump (2). When the pump's delivery quantity equals the quantity of calibrating oil emerging from the test nozzle (10), the plunger remains in its center position. If the flow of calibrating oil is greater, the plunger moves to the left – if the flow of calibrating oil is lower, the plunger moves to the right. This plunger motion controls the amount of light traveling from an LED (3) to a photocell (4). The electronic control circuitry (7) records this deviation and responds by varying the pump's rotational speed until its delivery rate again corresponds to the quantity of fluid emerging from the test nozzle. The control plunger then returns to its center position. The pump speed can be varied to measure delivery quantity with extreme precision.

Two of these measurement cells are present on the test bench. The computer connects all of the test cylinders to the two measurement cells in groups of two, proceeding sequentially from one group to the next (multiplex operation). The main features of this test method are:

- Highly precise and reproducible test results
- Clear test results with digital display and graphic presentation in the form of bar graphs
- Test record for documentation
- Supports adjustments to compensate for variations in cooling and/or temperature

Testing in-line fuel-injection pumps

The test program for fuel-injection pumps involves operations that are carried out with the pump fitted to the engine in the vehicle (system fault diagnosis) as well as those performed on the pump in isolation on a test bench or in the workshop. This latter category involves

- Testing the fuel-injection pump on the pump test bench and making any necessary adjustments
- Repairing the fuel-injection pump/governor and subsequently resetting them on the pump test bench

In the case of in-line fuel-injection pumps, a distinction has to be made between those with mechanical governors and those which are electronically controlled. In either case, the pump and its governor/control system are tested in combination, as both components must be matched to each other.

The large number and variety of in-line fuel-injection pump designs necessitates variations in the procedures for testing and adjustment. The examples given below can, therefore, only provide an idea of the full extent of workshop technology.

Adjustments made on the test bench

The adjustments made on the test bench comprise

- Start of delivery and cam offset for each individual pump unit
- Delivery quantity setting and equalization between pump units
- Adjustment of the governor mounted on the pump
- Harmonization of pump and governor/control system (overall system adjustment)

For every different pump type and size, separate testing and repair instructions and specifications are provided which are specifically prepared for use with Bosch pump test benches.

The pump and governor are connected to the engine lube-oil circuit. The oil inlet connection is on the fuel-injection pump's camshaft housing or the pump housing.
For each testing sequence on the test bench, the fuel-injection pump and governor must be topped up with lube oil.

Testing delivery quantity

The fuel-injection pump test bench can measure the delivery quantity for each individual cylinder (using a calibrated tube apparatus or computer operating and display terminal, see "Fuel-injection pump test benches"). The individual delivery quantity figures obtained over a range of different settings must be within defined tolerance limits. Excessive divergence of individual delivery quantity figures would result in uneven running of the engine. If any of the delivery quantity figures are outside the specified tolerances, the pump barrel(s) concerned must be readjusted. There are different procedures for this depending on the pump model.

Governor/control system adjustment

Governor

Testing of mechanical governors involves an extensive range of adjustments. A dial gauge is used to check the control-rack travel at defined speeds and control-lever positions on the fuel-injection pump test bench. The test results must match the specified figures. If there are excessive discrepancies, the governor characteristics must be reset. There are a number of ways of doing this, such as changing the spring characteristics by altering spring tension, or by fitting new springs.

Electronic control system

If the fuel-injection pump is electronically controlled, it has an electromechanical actuator that is operated by an electronic control unit instead of a directly mounted governor. That actuator moves the control rack and thus controls the injected fuel quantity. Otherwise, there is no difference in the mechanical operation of the fuel-injection pump.

During the tests, the control rack is held at a specific position. The control-rack travel must be calibrated to match the voltage signal of the rack-travel sensor. This done by adjusting the rack-travel sensor until its signal voltage matches the specified signal level for the set control-rack travel.

In the case of control-sleeve in-line fuel-injection pumps, the start-of-delivery solenoid is not connected for this test in order to be able to obtain a defined start of delivery.

Adjustments with the pump in situ

The pump's start of delivery setting has a major influence on the engine's performance and exhaust-gas emission characteristics. The start of delivery is set, firstly, by correct adjustment of the pump itself, and secondly, by correct synchronization of the pump's camshaft with the engine's timing system. For this reason, correct mounting of the injection pump on the engine is extremely important. The start of delivery must therefore be tested with the pump mounted on the engine in order to ensure that it is correctly fitted.

There are a number of different ways in which this can be done depending on the pump model. The description that follows is for a Type RSF governor.

On the governor's flyweight mount, there is a tooth-shaped timing mark (Fig. 1).
In the governor housing, there is a threaded socket which is normally closed off by a screw cap. When the piston that is used for calibration (usually no. 1 cylinder) is in the start-of-delivery position, the timing mark is exactly in line with the center of the threaded socket. This "spy hole" in the governor housing is part of a sliding flange.

Fitting the fuel-injection pump

Locking the camshaft

The fuel-injection pump leaves the factory with its camshaft locked (Fig. 1a) and is mounted on the engine when the engine's crankshaft is set at a defined position. The pump lock is then removed. This tried and tested method is economical and is adopted increasingly widely.

Start-of-delivery timing mark

Synchronizing the fuel-injection pump with the engine is performed with the aid of the start-of-delivery timing marks, which have to be brought into alignment. Those marks are to be found on the engine as well as on the fuel-injection pump (Fig. 2 overleaf). There are several methods of determining the start of delivery depending on the pump type.

Normally, the adjustments are based on the engine's compression stroke for cylinder no. 1 but other methods may be adopted for reasons related to specific engine designs. The engine manufacturer's instructions must therefore always be observed. On most diesel engines, the start-of-delivery timing mark is on the flywheel, the crankshaft pulley or the vibration damper. The vibration damper is generally mounted on the crankshaft in the position normally occupied by the V-belt pulley, and the pulley then bolted to the vibration damper. The complete assembly then looks rather like a thick V-belt pulley with a small flywheel.

1 Devices for setting and checking start of delivery (port-closing sensors)

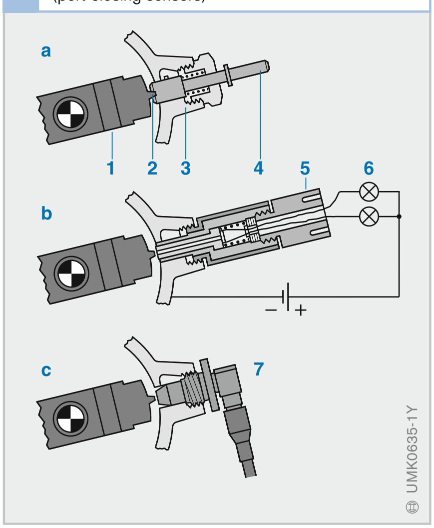


Fig. 1
Illustration shows Type RSF governor; other types have a sliding flange

- a Locked in position by locking pin
- b Testing with an optical sensor (indicator-lamp sensor)
- c Testing with an inductive sensor (governor signal method)

1. Governor flyweight mount
2. Timing mark
3. Governor housing
4. Locking pin
5. Optical sensor
6. Indicator lamp
7. Inductive speed sensor

Checking static start of delivery

Checking with indicator-lamp sensor
The tooth-shaped timing mark can be located with the aid of an optical sensor, the indicator-lamp sensor (Fig. 1b), which is screwed into the socket in governor housing. When it is opposite the sensor, the two indicator lamps on the sensor light up. The start of delivery in degrees of crankshaft rotation can then be read off from the flywheel timing marks, for example.

High-pressure overflow method
The start-of-delivery tester is connected to the pressure outlet of the relevant pump barrel (Fig. 3). The other pressure outlets are closed off. The pressurized fuel flows through the open inlet passage of the pump barrel and exits, initially as a jet, into the observation vessel (3). As the engine crankshaft rotates, the pump plunger moves towards its top dead center position. When it reaches the start-of-delivery position, the pump plunger closes off the barrel's inlet passage. The injection jet entering the observation vessel thus dwindles and the fuel flow is reduced to a drip. The start of delivery in degrees of crank shaft rotation is read off from the timing marks.

Checking dynamic start of delivery

Checking with inductive sensor
An inductive sensor that is screwed into the socket in the governor housing (Fig. 1c) supplies an electrical signal every time the governor timing mark passes when the engine is running. A second inductive sensor supplies a signal when the engine is at top dead center (Fig. 4). The engine analyzer, to which the two inductive sensors are connected, uses those signals to calculate the start of delivery and the engine speed.

Checking with a piezoelectric sensor and a stroboscopic timing light
A piezoelectric sensor is fixed to the high-pressure delivery line for the cylinder on which adjustment is to be based. As soon as the fuel-injection pump delivers fuel to that cylinder, the high-pressure delivery line expands slightly and the piezoelectric sensor transmits an electrical signal. This signal is received by an engine analyzer which uses it to control the flashing of a stroboscopic timing light. The timing light is pointed at the timing marks on the engine. When illuminated by the flashing timing light, the flywheel timing marks appear to be stationary. The angular value in degrees of crankshaft rotation can then be read off for start of delivery.

2 Timing marks on the engine used for setting the fuel-injection pump

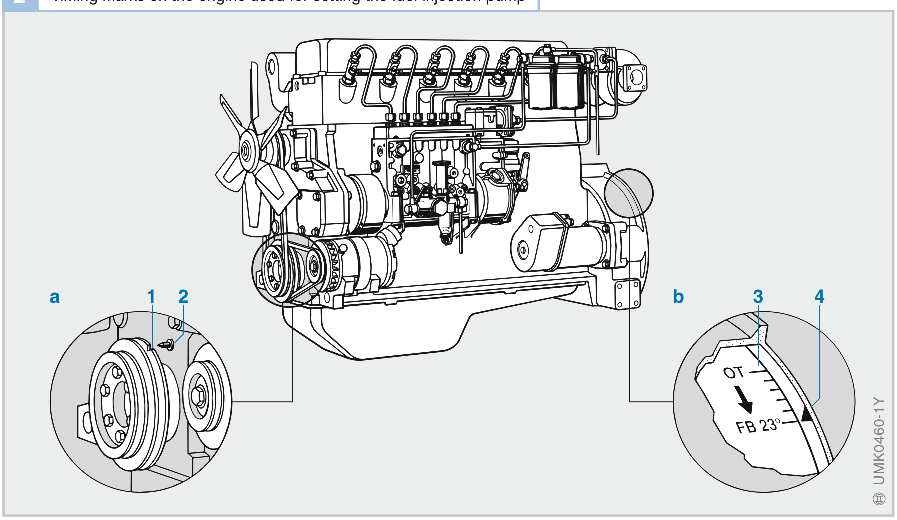


Fig. 2
- a V-belt pulley timing marks
- b Flywheel timing marks

1. Notch in V-belt pulley
2. Marker point on cylinder block
3. Graduated scale on flywheel
4. Timing mark on crankcase

Venting

Air bubbles in the fuel impair the proper operation of the fuel-injection pump or disable it entirely. Therefore, if the system has been temporarily out of use it should be carefully vented before being operated again. There is generally a vent screw on the fuel-injection pump overflow or the fuel filter for this purpose.

Lubrication

Fuel-injection pumps and governors are normally connected to the engine lube-oil circuit as the fuel-injection pump then requires no maintenance.

Before being used for the first time, the fuel-injection pump and the governor must be filled with the same type of oil that is used in the engine. In the case of fuel-injection pumps that are not directly connected to the engine lube-oil circuit, the pump is filled through the filler cap after removing the vent flap or filter. The oil level check takes place at the same time as the regular engine oil changes and is performed by removing the oil check plug on the governor. Excess oil (from leak fuel) is then drained off or the level topped up if required. Whenever the fuel-injection pump is removed or the engine overhauled, the oil must be changed. Fuel-injection pumps and governors with separate oil systems have their own dipsticks for checking the oil level.

4 Checking dynamic start of delivery

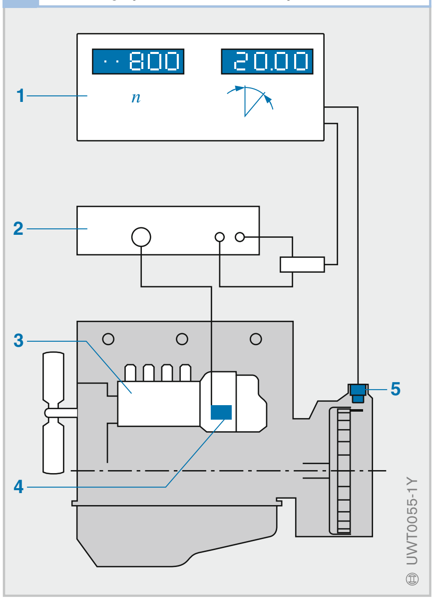


Fig. 4
Schematic diagram of in-line fuel-injection pump and governor using port-closing sensor system

1 Engine analyzer
2 Adaptor
3 In-line fuel-injection pump and governor
4 Inductive speed sensor (port-closing sensor)
5 Inductive speed sensor (TDC sensor)

3 Schematic diagram of start-of-delivery calibrating unit (high-pressure overflow method)

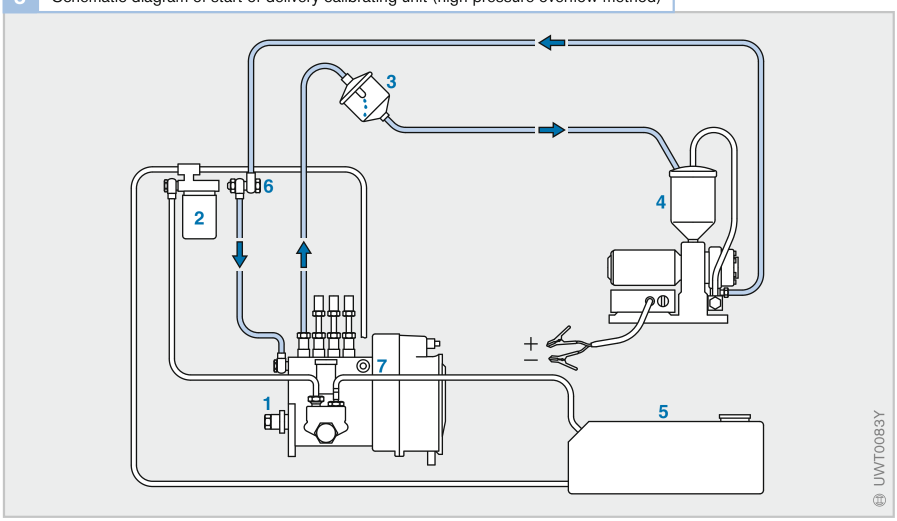


Fig. 3

1 Fuel-injection pump
2 Fuel filter
3 Observation vessel
4 Start-of-delivery calibrating unit
5 Fuel tank
6 Oversize banjo bolt and nut
7 Screw cap

Testing helix and port-controlled distributor injection pumps

Good engine performance, high fuel economy and low emissions depend on correct adjustment of the helix and port-controlled distributor injection pump. This is why compliance with official specifications is absolutely essential during testing and adjustment operations on fuel-injection pumps.

One important parameter is the start of delivery (in service bay), which is checked with the pump installed. Other tests are conducted on the test bench (in test area). In this case, the pump must be removed from the vehicle and mounted on the test bench. Before the pump is removed, the engine crankshaft should be rotated until the reference cylinder is at TDC. The reference cylinder is usually cylinder No. 1. This step eases subsequent assembly procedures.

Test bench measurements

The test procedures described here are suitable for use on helix and port-controlled axial-piston distributor pumps with electronic and mechanical control, but not with solenoid-controlled distributor injection pumps.

Test bench operations fall into two categories:

- Basic adjustment and
- Testing

The results obtained from the pump test are entered in the test record, which also lists all the individual test procedures. This document also lists all specified minimum and maximum results. The test readings must lie within the range defined by these two extremes.

A number of supplementary, special-purpose test steps are needed to assess all the different helix and port-controlled axial-piston distributor pumps; detailed descriptions of every contingency, however, extend beyond the bounds of this chapter.

1 Equipment for testing the distributor injection pump (on test bench)

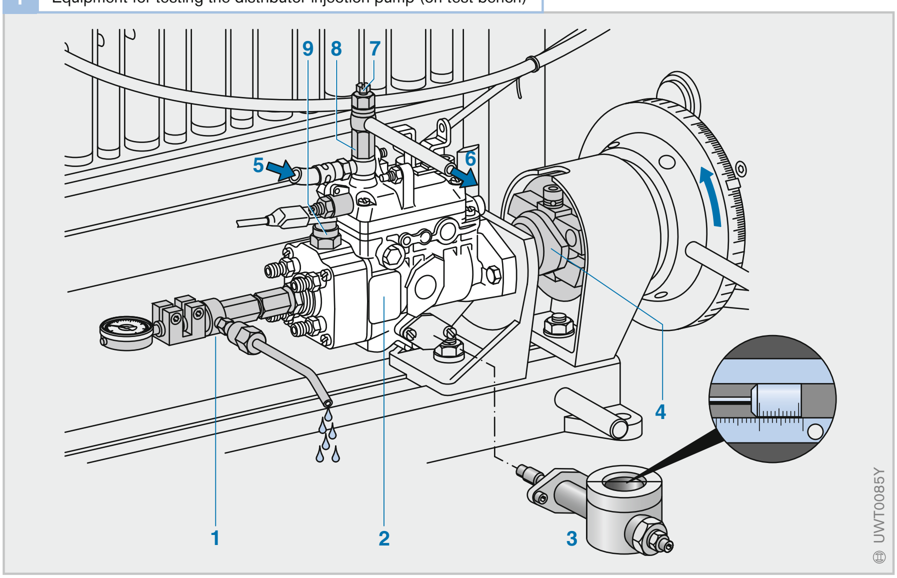


Fig. 1
1 Test layout with drain hose and dial gauge
2 Distributor injection pump
3 Timing device travel tester with vernier scale
4 Pump drive
5 Calibrating oil inlet
6 Return line
7 Overflow restrictor
8 Adapter with connection for pressure gauge
9 Electric shutoff valve (ELAB) (energized)

Basic adjustments

The first step is to adjust the distributor injection pump to the correct basic settings. This entails measuring the following parameters under defined operating conditions.

LPC adjustment

This procedure assesses the distributor plunger lift between **B**ottom **D**ead **C**enter (BDC) and the start of delivery. The pump must be connected to the test-bench fuel supply line for this test. The technician unscrews the 6-point bolt from the central plug fitting and then installs a test assembly with drain tube and gauge in its place (Fig. 1, 1).

The gauge probe rests against the distributor plunger, allowing it to measure lift. Now the technician turns the pump's input shaft (4) by hand until the needle on the gauge stops moving. The control plunger is now at **T**op **D**ead **C**enter (TDC).

A supply pressure of roughly 0.5 bar propels the calibrating oil into the plunger chamber behind the distributor plunger (5). For this test, the solenoid-operated shutoff valve (ELAB) (9) is kept energized to maintain it in its open position. The calibrating oil thus flows from the plunger chamber to the test assembly before emerging from the drain hose.

Now the technician manually rotates the input shaft in its normal direction of rotation. The calibrating oil ceases to flow into the plunger chamber once its inlet passage closes. The oil remaining in the chamber continues to emerge from the drain hose. This point in the distributor plunger's travel marks the start of delivery.

The lift travel between **B**ottom **D**ead **C**enter (BDC) and the start of delivery indicated by the gauge can now be compared with the setpoint value. If the reading is outside the tolerance range, it will be necessary to dismantle the pump and replace the cam mechanism between cam disk and plunger.

Supply-pump pressure

As it affects the timing device, the pressure of the supply pump (internal pressure) must also be tested. For this procedure, the overflow restrictor (7) is unscrewed and an adapter with a connection to the pressure gauge (8) is installed. Now the overflow restrictor is installed in an adapter provided in the test assembly. This makes it possible to test the pump's internal chamber pressure upstream of the restrictor.

A plug pressed into the pressure-control valve controls the tension on its spring to determine the pump's internal pressure. Now the technician continues pressing the plug into the valve until the pressure reading corresponds to the setpoint value.

Timing device travel

The technician removes the cover from the timing device to gain access for installing a travel tester with a vernier scale (3). This scale makes it possible to record travel in the timing device as a function of rotational speed; the results can then be compared with the setpoint values. If the measured timing device travel does not correspond with the setpoint values, shims must be installed under the timing spring to correct its initial spring tension.

Adjusting the basic delivery quantity

During this procedure the fuel-injection pump's delivery quantity is adjusted at a constant rotational speed for each of the following four conditions:

- Idle (no-load)
- Full-load
- Full-load governor regulation and
- Starting

Delivery quantities are monitored using the MGT or KMA attachment on the fuel-injection pump test bench (refer to section on "Fuel-injection pump test benches").

First, with the control lever's full-load stop adjusted to the correct position, the full-load governor screw in the pump cover is adjusted to obtain the correct full-load delivery quantity at a defined engine speed. Here, the governor adjusting screw must be turned back to prevent the full-load stop from reducing delivery quantity.

The next step is to measure the delivery quantity with the control lever against the idle-speed stop screw. The idle-speed stop screw must be adjusted to ensure that the monitored delivery quantity is as specified.

The governor screw is adjusted at high rotational speed. The measured delivery quantity must correspond to the specified full-load delivery quantity.

The governor test also allows verification of the governor's intervention speed. The governor should respond to the specified rpm threshold by first reducing and then finally interrupting the fuel flow. The breakaway speed is set using the governor speed screw.

There are no simple ways to adjust the delivery quantity for starting. The test conditions are a rotational speed of 100 rpm and the control lever against its full-load shutoff stop. If the measured delivery quantity is below a specified level, reliable starting cannot be guaranteed.

Testing

Once the basic adjustment settings have been completed, the technician can proceed to assess the pump's operation under various conditions. As during the basic adjustment procedure, testing focuses on

- Supply-pump pressure
- Timing device travel
- Delivery quantity curve

The pump operates under various specific conditions for this test series, which also includes a supplementary procedure.

Overflow quantity

The vane-type supply pump delivers more fuel than the nozzles can inject. The excess calibrating oil must flow through the overflow restriction valve and back to the oil tank. It is the volume of this return flow that is measured in this procedure. A hose is connected to the overflow restriction valve; depending on the selected test procedure. The other end is then placed in a glass gauge in the MGT assembly, or installed on a special connection on the KMA unit. The overflow quantity from a 10-second test period is then converted to a delivery quantity in liters per hour.

If the test results fail to reach the setpoint values, this indicates wear in the vane-type supply pump, an incorrect overflow valve or internal leakage.

Dynamic testing of start of delivery

A diesel engine tester (such as the Bosch ETD 019.00) allows precise adjustment of the distributor injection pump's delivery timing on the engine. This unit registers the start of delivery along with the timing adjustments that occur at various engine speeds with no need to disconnect any high-pressure delivery lines.

Testing with piezo-electric sensor and stroboscopic timing light

The piezo-electric sensor (Fig. 2, 4) is clamped onto the high-pressure delivery line leading to the reference cylinder. Here, it is important to ensure that the sensor is mounted on a straight and clean section of tubing with no bends; the sensor should also be positioned as close as possible to the fuel-injection pump.

The start of delivery triggers pulses in the fuel-injection line. These generate an electric signal in the piezo-electric sensor. The signal controls the light pulses generated by the timing light (5). The timing light is now aimed at the engine's flywheel. Each time the pump starts delivery to the reference cylinder, the timing light flashes, lighting up the TDC mark on the flywheel. This allows correlation of timing to flywheel position. The flashes occur only when delivery to the reference cylinder starts, producing a static image. The degree markings (6) on the crankshaft or flywheel show the crankshaft position relative to the start of delivery.

Engine speed is also indicated on the diesel engine tester.

Setting start of delivery

If the results of this start-of-delivery test deviate from the test specifications, it will be necessary to change the fuel-injection pump's angle relative to the engine.

The first step is to switch off the engine. Then the technician rotates the crankshaft until the reference cylinder's piston is at the point at which delivery should start. The crankshaft features a reference mark for this operation; the mark should be aligned with the corresponding mark on the bellhousing. The technician now unscrews the 6-point screw from the central plug screw. As for basic adjustment process on the test bench, the technician now installs a dial-gauge assembly in the opening. This is used to observe distributor plunger travel while the crankshaft is being turned. As the crankshaft is turned counter to its normal direction of rotation (or in the normal direction on some engines), the plunger retracts in the pump. The technician should stop turning the crankshaft once the needle on the gauge stops moving. The plunger is now at bottom dead center. Now the dial gauge is reset to zero. The crankshaft is then rotated in its normal direction of rotation as far as the TDC mark. The dial gauge now indicates the travel executed by the distributor plunger on its way from its bottom dead center position to the TDC mark on the reference cylinder. It is vital to comply with the precise specification figure for this travel contained in the fuel-injection pump's datasheet. If the dial gauge reading is not within the specification, it will be necessary to loosen the attachment bolt on the pump flange, turn the pump housing and repeat the test. It is important to ensure that the cold-start accelerator is not active during this procedure.

Measuring the idle speed

The idle speed is monitored with the engine heated to its normal operating temperature, and in a no-load state, using the engine tester. The idle speed can be adjusted using the idle-speed stop screw.

2 Checking start of delivery with piezo-electric sensor and timing light

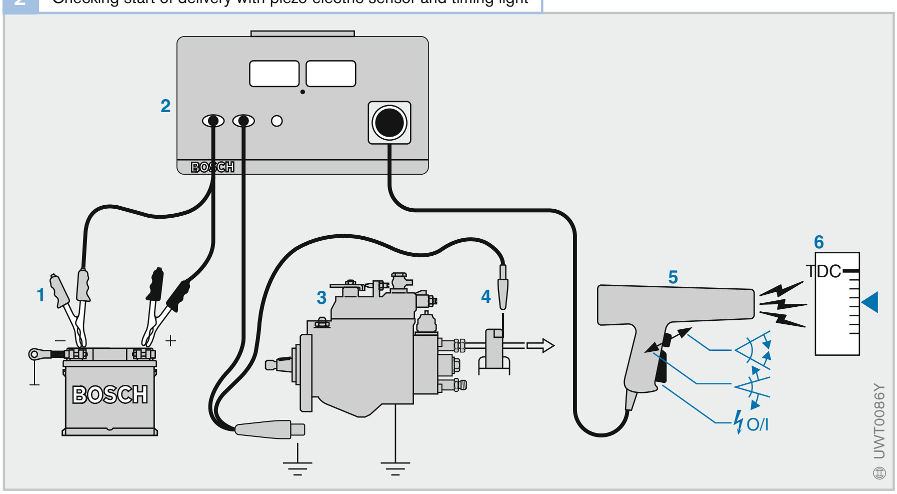


Fig. 2
1 Battery
2 Diesel tester
3 Distributor injection pump
4 Piezo-electric sensor
5 Stroboscopic timing light
6 Angle and TDC marks

Keep your hands away from the nozzle jet.
Spray from the nozzle stings and penetrates the skin.
There is a risk of blood poisoning.

Wear safety goggles.

Nozzle tests

The nozzle-and-holder assembly consists of the nozzle and the holder. The holder includes all of the required filters, springs and connections.

The nozzle affects the diesel engine's output, fuel economy, exhaust-gas composition and operating refinement. This is why the nozzle test is so important.

An important tool for assessing nozzle performance is the nozzle tester.

Nozzle tester

The nozzle tester is basically a manually operated fuel-injection pump (Fig. 1). For testing, a high-pressure delivery line (4) is used to connect the nozzle-and-holder assembly (3) to the tester. The calibrating oil is contained in a tank (5). The required pressure is generated using the hand lever (8). The pressure gage (6) indicates the pressure of the calibrating oil; a valve (7) can be used to disconnect it from the high-pressure circuit for specific test procedures.

1 Nozzle tester with nozzle-and-holder assembly

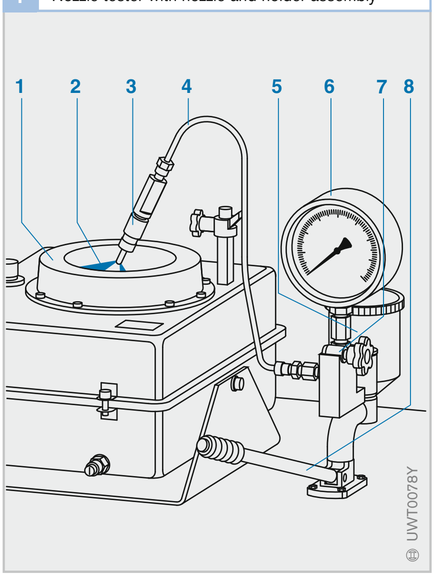


Fig. 1
1 Suction equipment
2 Injection jet
3 Nozzle-and-holder assembly
4 High-pressure test line
5 Calibrating oil tank with filter
6 Pressure gage
7 Valve
8 Hand lever

The EPS100 (0684200704) nozzle tester is specified for testing nozzles of Sizes P, R, S and T. It conforms to the standards defined in ISO 8984. The prescribed calibrating oil is defined in ISO standard 4113. A calibration case containing all the components is required to calibrate inspect the nozzle tester.

This equipment provides the basic conditions for reproducible, mutually compatible test results.

Test methods

Ultrasonic cleaning is recommended for the complete nozzle-and-holder assemblies once they have been removed from the engine. Cleaning is mandatory on nozzles when they are submitted for warranty claims.

Important: Nozzles are high-precision components. Careful attention to cleanliness is vital for ensuring correct operation.

The next step is to inspect the assembly to determine whether any parts of the nozzle or holder show signs of mechanical or thermal wear. If signs or wear are present, it will be necessary to replace the nozzle or nozzle-and-holder assembly.

The assessment of the nozzle's condition proceeds in four test steps, with some variation depending on whether the nozzles are pintle or hole-type units.

Chatter test

The chatter test provides information on the smoothness of action of the needle. During injection, the needle oscillates back and forth to generate a typical chatter. This motion ensures efficient dispersion of the fuel particles.

The pressure gage should be disconnected for this test (close valve).

Pintle nozzle

The lever on the nozzle tester is operated at a rate of one to two strokes per second. The pressure of the calibrating oil rises, ultimately climbing beyond the nozzle's opening pressure. During the subsequent discharge, the nozzle should produce an audible chatter; if it fails to do so, it should be replaced.

When installing a new nozzle in its holder, always observe the official torque specifications, even on hole-type nozzles.

Hole-type nozzle
The hand lever is pumped at high speed. This produces a hum or whistling sound, depending on the nozzle type. No chatter will be present in some ranges. Evaluation of chatter is difficult with hole-type nozzles. This is why the chatter test is no longer assigned any particular significance as an assessment tool for hole-type nozzles.

Spray pattern test

High pressures are generated during this test. Always wear safety goggles.

The hand lever is subjected to slow and even pressure to produce a consistent discharge plume. The spray pattern can now be evaluated. It provides information on the condition of the injection orifices. The prescribed response to an unsatisfactory spray pattern is to replace the nozzle or nozzle-and-holder assembly.

The pressure gage should also be switched off for this test.

Pintle nozzle
The spray should emerge from the entire periphery of the injection orifice as even tapered plume. There should be no concentration on one side (except with flatted pintle nozzles).

Hole-type nozzle
An even tapered plume should emerge from each injection orifice. The number of individual plumes should correspond to the number of orifices in the nozzle.

Checking the opening pressure

Once the line pressure rises above the opening pressure, the valve needle lifts from its seat to expose the injection orifice(s). The specified opening pressure is vital for correct operation of the overall fuel-injection system.

The pressure gage must be switched back on for this test (valve open).

Pintle nozzle and hole-type nozzle with single-spring nozzle holder
The operator slowly presses the lever downward, continuing until the gage needle indicates the highest available pressure. At this point, the valve opens and the nozzle starts to discharge fuel. Pressure specifications can be found in the “nozzles and nozzle-holder components” catalog.

Opening pressures can be corrected by replacing the adjustment shim installed against the compression spring in the nozzle holder. This entails extracting the nozzle from the nozzle holder. If the opening pressure is too low, a thicker shim should be installed; the response to excessive opening pressures is to install a thinner shim.

Hole-type nozzle with two-spring nozzle holder
This test method can only be used to determine the initial opening pressure on two-spring nozzle-and-holder assemblies.

The is no provision for shim replacement on some nozzle-and-holder assemblies. The only available response with these units is to replace the entire assembly.

Leak test

The pressure is set to 20 bar above the opening pressure. After 10 seconds, formation of a droplet at the injection orifice is acceptable, provided that the droplet does not fall.

The prescribed response to an unsuccessful leak test is to replace the nozzle or nozzle-and-holder assembly.

Index

ABS 195
ABS control loop 196
AC firing 216
ACC 202
ACC control system 203
acceleration 125
acceleration sensor 212, 215
accident prevention 107
accident causes 107
active safety systems 105
actuator diagnostics 248, 250
actuator triggering 179
actuators 173
adaptive cruise control 202
adaptive light pattern 191
adaptive lighting systems 190
adaptive rear lights 190
adaptive rearlighting system 190
adjusting the basic delivery quantity 265
adjustment 204, 253
adjustments made on the test bench 260
advanced secondary injection 48
advanced start of injection 43
aerodynamic drag 125
agricultural machinery 20
aiming strategy 191
air system 163
air/fuel ratio 54
air/fuel ratio curve 41
alternator 142
altitude and turbocharger/supercharger
pressure limits 34
amplification 207
angle measurement 206
α/n system 59
antenna diagram 207
Antilock Braking System (ABS) 131, 195
antenna system 207
application force 128
aquaplaning 121
ARS 190
AST 82
AST components 84
AT 88
ATF 91
Atkinson cycle 238
automated shift transmission 82
automatic headlamp leveling control 189
automatic state machine 234
automatic transmission 88
automatic transmission fluid 91
automotive light sources 180
average deceleration 129
axle-split parallel hybrid (AS-HEV) 226

band brakes 134
basic adjustments 265
battery changeover 149
battery charging relay 151
battery master switch 150
battery relay 151
battery status recognition 147
battery-cutoff relay 151
belt-force limiters 211
Benz, Bertha 2
Benz, Carl Friedrich 5
boosting 220
Bosch, Robert 6, 14
Brake Assistant (BA) 131
brake disks and drums 253
brake fluid level 255
brake hoses and lines 255
brake hysteresis 128
brake intervention 203
brake pads 137, 252
brake shoes 137, 252
brake tester 254
brake testing 252
brake torque 199
brake-circuit configuration 141
braked wheel 116
braking deceleration 129
braking distance 129
braking during cornering 111
braking factor 129
braking force 120
braking sequence 128
braking torque 119, 120, 128
braking-force distribution 128, 242
braking-system hysteresis 128
buckle tighteners 211

cable-operated rear wheel drum brakes 135
car 2, 19
car braking systems 130, 140
carriage 2
central combined airbag 9 ECU 214
centrifugal force 127
chain of interaction 49
charge recording 59

K. Reif (Ed.), *Fundamentals of Automotive and Engine Technology*,
DOI 10.1007/978-3-658-03972-1, © Springer Fachmedien Wiesbaden 2014

charge-balance calculation 144
charging the battery 143
chatter test 268
checking dynamic start of delivery 262
checking static start of delivery 262
checking wear 252
checking with a piezoelectric sensor 262
checking with indicator-lamp sensor 262
checking with inductive sensor 262
circumferential force 118
classification of car braking systems 138
closed-loop control 160
coefficient of friction 120
Coil On Plug 170
combined ECUs 213
combustion 53
combustion chambers 36
combustion knock 66
combustion pressure limits 33
combustion torque 60
commercial vehicles 11, 112
communication 160
compact-coil design 170
comparison of diesel and gasoline engines 21
components of the cylinder charge 56
compressed-air injection 10
compression 24, 53, 54
compression pressure 29
compression ratio 24, 54
compression stroke 23
compression temperature 29
compression volume 24
connecting devices 77
constant high pressure 48
construction machinery 20
consumers 143
contact systems 156
continuously variable transmission 96, 229
control algorithms 204
control systems in the drivertrain 231
controllable pump pressure 92
controlled system 197
controlled variables 197, 198
controller hierarchy 194
conventional braking systems 130
cooperative braking maneuvers 241
cooperatively regenerative braking system 241
COP 170
cornering 127
cornering lights (Europe) 191
course setting 202
crankshaft 24
crankshaft drive 61
crescent oil pump 92
critical traffic situations 107
crosswinds 126
cumulative fuel consumption 236
current output 142
current regulator 174
CVT 96
CVT components 100
CVT oil pump 101
cylinder charge 56

Daimler, Gottlieb 4
data processing 174
DCT 86
deactivation switch 212
dead volume 51
deceleration 125
degrees of hybridization 222
design of the vehicle electrical system 144
development history 2
development of electronic systems 158
diagnostics in the workshop 248
dielectrical resonance oscillator 206
diesel engine 22
Diesel fuel heater 30
Diesel, Rudolf Christian 5, 9
Diesel's patent 9
digital signal processor 208
dipped beam 184, 185
direct injection 11, 36
direct-injection (DI) engines 36, 50, 51
disk brake 137
distance sensor 202
distribution of drive torque 234
distribution of forces 117
distributor injection pump 264
distributorless (stationary) voltage distribution 75
divided combustion chamber 37
doppler effect 206
downsizing 238
drag coefficient 124, 125
drive configurations 224
drive torque 199
driveability 80
driver behavior 106
drivetrain 78

driving behavior 108
driving maneuvers 111
driving profile 145
driving stability systems 137
driving states 234
DRO 206
drum brakes 135
DSP 208
dual-clutch transmission 86, 228
dual-spark ignition coils 169
dwell period 72
dynamic supercharging 58
dynamic system 190
dynamic testing of start of delivery 266
dynamics of lateral motion 126
dynamics of linear motion 124

EC motors 85
ECU 173
EDC 172
effective braking time 129
effective efficiency 28
efficiency 28
efficiency chain 63
efficiency index 28
efficiency of cycle factor 28
electric 4 WD functionality 227
electrical activation 165
electrical energy generation 237
electrical energy management 146
electrical energy supply 142
electrical systems 158
electrically actuated throttle valve 161
electrohydraulic brakes (SBC) 131
electromagnetic fuel injector 164
electromechanical brakes (EMB) 131
electronic battery sensor 147
electronic braking systems 131
electronic braking-force distribution 131
electronic control system 260
electronic control unit 195
electronic diesel control 172, 175
electronic flow measurement system 259
electronic impact protection system 210
electronic service information 246
electronic stability program 194
electronic systems 158
electronics 158
engine characteristic data 21
engine characteristics 178
engine efficiency 26, 62
engine knock 67
engine speed limits 34
engine-control intervention 203
engines with indirect injection (IDI) 50, 51
equivalence value 235
ESP (Electronic Stability Program) 131, 194
ESP control loop 194
evaluating driver behavior 109
evaporative emission control system 161
excess-air factor 54, 64
excess-air factor λ 40
exhaust 53
exhaust stroke 23
exhaust-gas temperature limits 34
exhaust-gas turbocharging 59, 161
external brake coefficient 128
external exhaust-gas recirculation 161
external shoe brakes 134
external torque demands 179

fast fourier transform 208
fault information 248
fault memory entries 248
FFT 208
first diesel engine 10
fitting the fuel-injection pump 261
5-speed transmission 81
fixed-installation engine 18
flame propagation 55
flame-front propagation 69, 169
flow measurement methods 258
fluid flow 93
FMCW method 205
FMCW radar 209
force transmission system 140
forces 128
forces acting on a vehicle 117
Ford, Henry 5
for-stroke cycle 23
4-speed automatic transmission 88
four-stroke diesel engine 23
four-stroke principle 52
4-stroke spark-ignition (SI) gasoline engine 52
frequency modulation 205
frequency shift 206
fresh gas 56
friction 123
friction force 120
friction optimization 239

frictional losses 63
front airbag 211
fuel consumption 145
fuel consumption in everyday practice 39
fuel conversion factor 28
fuel heating 29
fuel injector operation 164
fuel injectors 161, 164
fuel outlet 165
fuel system 161, 163
fuel-consumption map 65
fuel-injection control 176
fuel-injection parameters 42, 172
fuel-injection process 177
fuel-injection pump 15, 262
fuel-injection pump test benches 258
fuel-injection system 35
full hybrid 222
full load 31
function modules 160

gas exchange 57
gas inflators 216
gas-discharge lamps 181, 186
gasoline (SI) engine 52
gasoline direct injection 166
general-purpose battery 148
generator mode 220
glass gauge method 258
Governor 17, 260
governor/control system adjustment 260
guided troubleshooting 248

H4 halogen lamp 180
halogen lamp 180
headlamp leveling control 188
headlamp systems 185
heavy goods vehicle 20
HF generation 206
HFM 59
HH configuration 141
HI configuration 141
high beam 184, 185
high-frequency 206
high-pin-count plug-in connections 155
high-pressure fuel injector 166, 167
high-pressure overflow method 262
high-pressure test 255
Hill Descent Control (HDC) 131
historical development of ABS 200
history 2, 8
history of radar 201
history of the alternator 157
history of the brake 132
hole-type nozzle 269
hole-type nozzle with two-spring
nozzle holder 269
hot-film air-mass meter 59
hybrid control 231
hybrid drives 218
hybrid driving 219
hydraulic brake activation 136
hydraulic modulator 195

ideal fuel consumption 65
idle 31
ignition 55
ignition cables 77
ignition circuit 68, 168
ignition coils 68, 161, 169, 170
ignition driver stage 68, 76
ignition energy 72
ignition energy balance 73
ignition parameters 71
ignition spark 69, 74, 168
ignition stroke 23
ignition system 161
ignition timing 64
ignition voltage 72
ignition-angle corrections 71
II configuration 141
impact detection 212
incandescent (vacuum) bulb 180
incandescent bulb 180
indirect injection 37
induction 53
induction stroke 23
induction-mixture distribution 54, 55
inductive ignition system 68, 168
initial braking phase 197
injected-fuel quantity 32, 44
injection adaptation 30
injection direction 51
injection duration 44
injection functions 48
injection pattern 46, 47
injection pressure 50
injection-duration correction 165
injection-pressure curve 46
inspection 252

intake-manifold injection 164
intelligent airbag systems 212
interference suppressors 77
interference-suppression resistors 77
internal brake coefficient 128
internal shoe drum brake 135
internal torque demands 179
internal-combustion engine 237

job order acceptance 245

kinetic friction 121
kingpin offset 119
KMA 259
knock 66
knock control 67, 70
knock recognition 67

lambda, λ 40, 54
lambda levels 41
lateral force 118, 122
lateral slip 122
lateral slip angle 122
leak test 269
Lepelletier planetary-gear set 89
lever arm 61
light commercial vehicles 19
light emitting diodes 183
lighting technology 180
line cross-sections 153
line protection 153
link-chain 101
litronic 186
LL configuration 141
load-change reaction 111
locking the camshaft 261
low-pressure test 255
low beam 184, 185
lower part-load range 31
low-frequency block diagram 208
low-pin-count plug in connections 156
LPC adjustment 265
lubrication 263

magnetic circuit 171
main headlamps (Europe) 184
main headlamps (North America) 185
maintenance 252
maintenance checklist 256
maintenance tasks 256
manual transmission 81
manually shifted transmission 80
master cylinder 254
Maybach, Wilhelm 4
measuring equipment 247
measuring the idle speed 267
mechanical efficiency 28
mechanical supercharging 59
mechanical variator adjustment 98
method of operation 52
micromechanics 217
mid-range cars 34
mild hybrid 222
mixture distribution 40
mixture ignition 73
moment of ignition 69, 169
momentary deceleration 129
motor-vehicle bulbs 192
motor-vehicle safety 104
motronic engine-management system 161
M system 38
multi-fuel engines 21
multiplate clutches 94
muscular-energy braking systems 139

network architecture 202
no load 31
non-steady-state operation 32
normal force 118
nozzle holder designs 51
nozzle secondary injection 51
nozzle tester 268
nozzle tests 268

occupant-protection system 210
offboard tester 249
oil pump 92
open-loop control 160
operating concept 185
operating conditions 33
operating modes 219
operating principle 173, 186
operating range 232, 233
operating statuses 29
operating strategies 232, 235
operating variable acquisition 161
operating variable processing 161
operating-point optimization 234
operation of hybrid vehicles 231
Otto, Nikolaus August 4

output torque 25
overflow quantity 266
overrun 32
oversteer 127

p/n system 59
parallel hybrid drive 225
parallel hybrid with different transmissions 228
parallel hybrid with one clutch 225
parallel hybrid with two clutches 226
parking lock 95
parking-brake system 138
part load 31
passenger-compartment sensing 216
passive safety systems 105
pencil coil 171
phase locked loop 206
pilot pressure test 255
pintle nozzle 268, 269
pintle nozzle and hole-type nozzle with single-
spring nozzle holder 269
planetary-gear sets 94, 229
PLL 206
plug connection 171
plug-in connections 154
plug-in hybrid 223
port-distributor 264
positive-pressure gradient 48
power 53, 60, 61, 178
power assisted car braking system 130
power curves 61
power output 25, 143
power take-up elements 90
power-assisted braking systems 139
power-brake systems 139
power-branching hybrid drive 228
power-flow 81
precombustion chamber system 37
precombustion-chamber engine 10
pre-injection 48
pressure build-up time 129
pressure curve 70
pressure-volume diagram 62
progress of combustion 40
projection headlamp 188
propagation time 205
propagation time measurement 205
propulsion torque 179
pumping losses 58, 63
purely electric driving 220
push-belt 100
p-V chart 26
p-V diagram 62, 63

radar signal processing 208
radial tires 114
railway locomotives 20
ranging radar 205
rate-of-discharge curve 46
Ravigneaux set 95
reaction time 112
reading actual values 250
reading/erasing the fault memory 250
real process 26, 27
real p-V diagram 62
recuperative braking 240
reflection headlamp 187
regenerative braking 221, 240
regenerative braking system 240
relative air charge 56
relative fuel quantity 56
remote version 170
replacing brake pads and shoes 253
required fuel mass 44
residual gas 57
response time 129
retarded secondary injection 48
retarded start of injection 43
rfq 56
road torque 199
rolling resistance 124
rolling wheel 116
rollover protection systems 213
rotating high-voltage distribution 75
rush-hour driving 144

safety systems 104
screening 77
seat belts 210
seat-belt pretensioners 210
secondary-brake system 138
Seiliger process 26
sensors 173
sequence in the workshop 247
series hybrid drive 224
series-parallel hybrid drive 224
service and repair implementation 245
service brakes 138, 243
setpoint generator 173
setting options 203

setting start of delivery 267
ships 20
shoe brakes 133
side airbag 212
side-slip angle 122
sideways forces 122
Simpson set 95
single-spark ignition coils 169
6-speed automatic transmission 90
slip 116
smoke limit 33
spark head 72
spark plugs 68, 161
spark tail 72
specific emission of unburned hydrocarbone 45
specific fuel consumption 45, 64
specific nitrogen oxide (NO2) emission 45
specific soot emission 45
speed of light 205
spray pattern test 269
start of delivery 43, 263
start of injection 42, 50
start of-delivery calibrating unit 263
start/stop function 221
start/stop functionality 221
startability 147
start-assist systems 30
starter battery 148
starting 29
starting system 149
start-of delivery timing mark 261
static system 189
steady-state operation 32
steady-state skid-pad circulation 111
steam buggy 2
stratified-charge concept 55, 64
stroboscopic timing light 262
suitability criteria 18
supercharging 58
supply-pump pressure 265
swirl-chamber system 38
system blocks 173

TCS 198
TCS interventions 199
test bench measurements 264
test functions 251
testing brakes 254
testing delivery quantity 260
testing equipment 250
testing helix 264
testing in-line fuel-injection pumps 260
the disturbance values 197
the ideal cycle 62
the reference variable 197
theory of inertia 117
thermal efficiency 28, 62
thermal losses 62
thermal radiators 180
throttle-valve model 59
timing characteristics 49
timing device travel 265
timing marks 262
tire forces 118
tire slip 123
tires 114
tires slip 116
toroid transmission 102
torque 25, 60, 61, 128
torque balance 199
torque control 178
torque converter 93
torque curves 61
torque-controlled EDC systems 178
torques at the drivetrain 60
total braking distance 129
total braking force 128
total braking time 129
total resistance to motion 124
traction-control system 198
transition response 111
transmission and reception circuit 206
transmission history 79
transmission history 2 103
transmission in the drivetrain 78
transmission requirements 80
transmissions for motor vehicles 78
transmit frequency 207
turning forces 117
two-battery vehicle electrical system 147, 148
two-mode hybrid 230
two-stage pressure gradient 48

understeer 127
undivided combustion chamber 36

value creation structure 159
valve timing 24, 53
valve timing diagram 53
variable braking 128
variable camshaft control 161
variable pump flow 92
variator 98, 100
variator control 99
VCO 206
vehicle dynamics 114
vehicle electrical system control unit 148
vehicle electrical systems 142, 149
vehicle operation 106
vehicle system analysis 246
venting 263
vertical tire force 118, 123
voltage distribution 75, 169
voltage-controlled oscillator 206

watchdog 215
wheel 2
wheel brake cylinders 255
wheelbrakes 140
wheel-speed sensors 195
winter operation 144
wiring harnesses 152
working cycle 52
workshop business 244
workshop diagnostic functions 249
workshop processes 245
workshop technology 244

x configuration 141
xenon 186

yaw moment 119
yaw-rate sensor 213