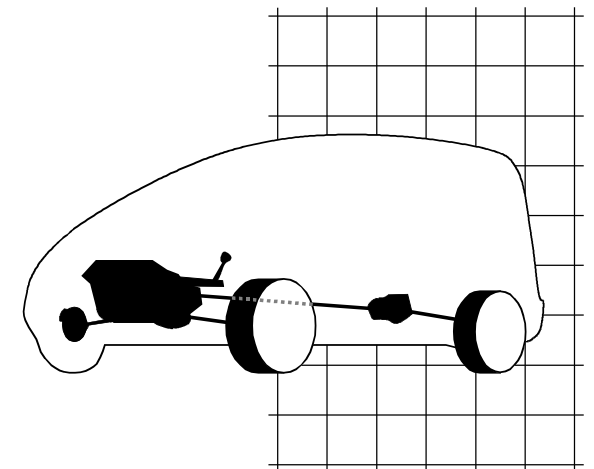


Henning Wallentowitz

Lecture

Longitudinal Dynamics of Vehicles

- Traffic system motor vehicle ●**
- Power and energy demand ●**
- Drivetrain ●**
- Vehicle dynamics ●**



Automotive Engineering I

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1 Introduction

The volume on hand deals with longitudinal dynamics of motor vehicles. This generic term summarizes the processes and components that influence the motor vehicle's movement towards longitudinal vehicle direction. Before dealing with the particular processes and components, the importance of motor vehicles in our society as well as some of the resulting problems should be pointed out briefly.

1.1 Traffic system motor vehicle

Generally speaking, situations and development objectives can be outlined for the road traffic by considering it as a "black-box" according to Fig. 1-1 and by comparing the expense with the result.

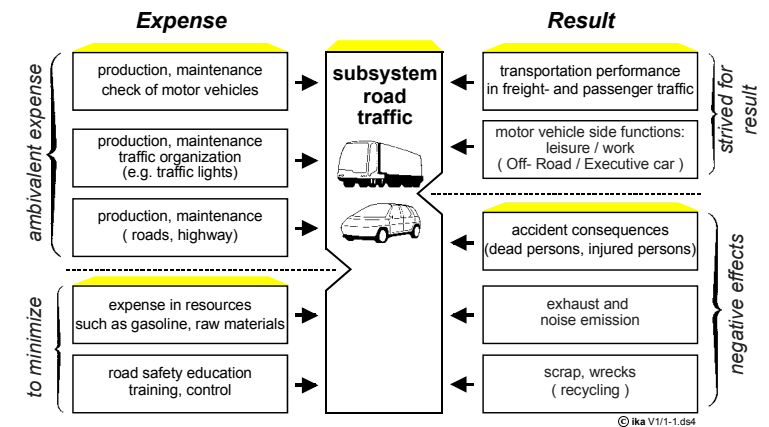


Fig. 1-1: Expense and result of the system "road traffic"

Considering it in this simplified way, the task consists in achieving the transportation performance as a result strived for, on the one hand with minor negative side effects and on the other hand with minor expense - e.g. limited resources such as crude oil. The expense, e.g. for the production of motor vehicles, is described as ambivalent insofar as it is linked with the value added and work places. However, this implies that costs emerging from that have to be minimized.

An economical cost-benefit analysis of these complex interrelations does not exist yet. It fails because all expense- and result factors cannot be quantified monetarily. In the future the reduction of the road traffic's negative effects will gain in considerable

importance. This importance can be seen as a result from the peoples' growing consciousness in ecology and the modern technological possibilities that enable such a reaction to these requests.

1.1.1 Ways of transportation

Generally the motor vehicle can be defined as a non-rail-bound land vehicle which serves to cover distance. According to the way of transportation, we distinguish between passenger- and freight traffic.

a) Passenger traffic

The passenger traffic's development in the Federal Republic of Germany is marked by an increasing degree of motorization. This motorization can be described either by the absolute number of motor vehicles or by the vehicle density. The total amount of all motor vehicles registered in the Federal Republic was about 48.1 millions by the end of 1996; this amount includes approx. 41 million automobiles.

Fig. 1-2 shows the development of the vehicle density in the Federal Republic of Germany. For the past, the typical image of a growth function is given which after a stage of disproportional increase (minor supply, high demand) changes to a stage of constant growth rates. The further course of this function depends on the so-called saturation point. At the moment the upper limit is considered to be between 650 and 700 automobiles per 1,000 inhabitants. In all descriptions shown in this chapter it has to be taken into consideration that all figures up to 1990 refer to the former Federal Republic of Germany and from 1991 onwards to the reunited Germany.

The increase of the automobile density (meaning the degree of motorization) goes hand in hand with the decrease of one single automobile's mileage. It decreased from the average of 14,600 km/a (9,072 miles/a) in 1975 to 12,700 km/a (7,892 miles/a) in 1994.

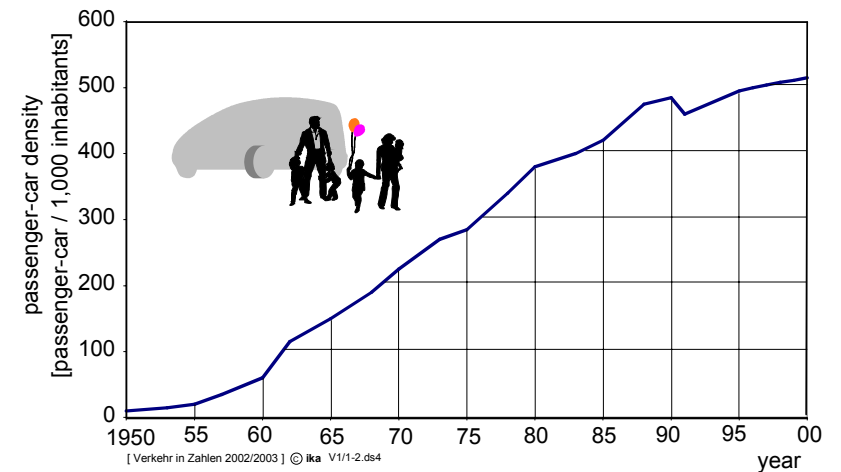


Fig. 1-2: Development of motorization in the Federal Republic of Germany
(Source: Verkehr in Zahlen 2002/2003)

Evaluating different systems of passenger traffic, we distinguish between individual traffic that essentially contains the traffic with cars and motorcycles and public transportation with busses and trains and to an increasing number also with aircrafts. The productivity or accomplished work of each single transportation system is described by the product of transported persons and distance, summed up as passenger kilometres, referring to the year considered.

Fig. 1-3 shows that the automobile (motorized individual traffic) plays a dominant role. Up to the year 1988, the automobile's share in the total transportation performance of passenger traffic increased to 83.5% and since then slightly dropped to 81.4% in 1995. The annual railroad's transportation performance was almost constant over many years, its share has been continually decreasing at the same time (1988: 5.7%). In the 90s the railroad's significance has again gained in importance, its share once again amounted to 7% in 1995. This tendency could last in the future if the railroad's attractiveness increases. All present predictions expect a further rising need for mobility which-in regard to the economic and ecological basic conditions-should be covered by both systems, by roads and rail way.

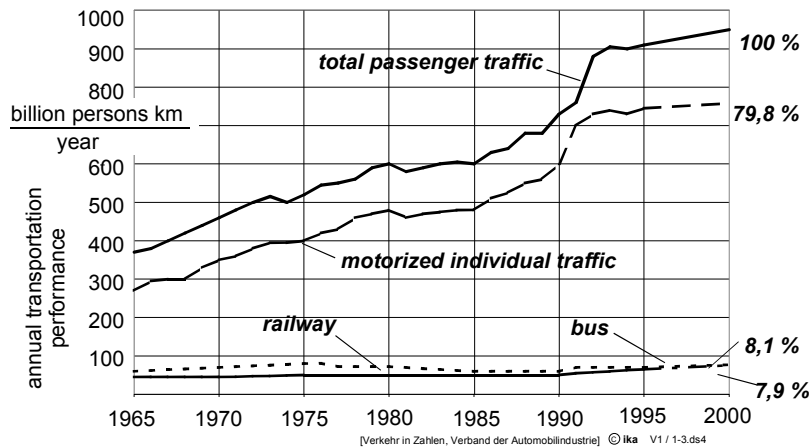


Fig. 1-3: Development of transport performance in passenger traffic
(Source: Verkehr in Zahlen 1996, VDA)

In order to judge the automobile correctly, it has to be pointed out that its importance goes far beyond of a pure traffic system. For the majority of automobile customers social aspects such as image and prestige as well as social acceptance when buying an automobile play an essential role. The vehicle's original purpose, safe transport of passengers and luggage, can take a backseat considering certain vehicles such as sports cars or convertibles. The sharp increase of models filling open market niches explains which various needs a car has to fulfil today and where its specific advantage can be found that other transport systems cannot offer in the same way.

b) Freight traffic

Freight traffic, in particular freight motor traffic, is subdivided by law into short-distance and long-distance traffic, inter alia because of fiscal reasons. According to the German "Güterkraftverkehrsgesetz" (§ 2 Abs. 2 GÜKG) the kind of freight traffic that covers distances within the radius of 50 km (31 miles) around the company site is called short-distance traffic. Road traffic with commercial motor vehicles nearly exclusively covers this so-defined field of transportation.

In order of freight transportation over long-distances, there are several different structured transportation systems available:

Commercial vehicle, railroad, ship, airplane, and pipeline.

The following characteristic features allow a comparison of the transport systems:

- transportation velocity - which is calculated on average under consideration of the time for handling;
- transportation stream/flow - as the amount of freight (masses or volume) that can be transported per hour and direction;
- cross section area/profile - as the area enclosed in the transportation systems' profile;
- payload ratio - which describes the ratio of total weight to payload.

Figures 1-4 and 1-5 compare these criteria for some transportation systems. This comparison shows that commercial vehicle and railway compared to inland vessel and pipeline have a higher transportation velocity and thus manage correspondingly large transport streams. In contrast, the pipeline has the smallest profile area due to its construction and therefore achieves the largest transportation stream related to the cross section area.

means of transportation	sectional profile	speed v	flow $\dot{V} \text{ (m}^3/\text{h)}$	area $A_p \text{ (m}^2\text{)}$	transportation flow $v' = v : A_p$
railway		32 mph (ohne Rangierbetrieb)	20,000	37	
highway		32 mph	14,500	115	
canal		7.5 mph	6,250	470	
pipeline		4.5 mph	2,850	0.4	

Fig. 1-4: Comparison of freight transportation systems

Regarding to the payload ratio, starting with the value of the pipeline and inland vessel over railroad to the commercial vehicle just slightly deteriorates (factor 1.7). In this comparison the airplane receives a very bad result. In order to transport one ton of capacity weight, six tons of "dead weight" are necessary. All systems are still developed in order to improve the payload ratio by minimizing weight. A rise in technical standards, such as noise behaviour or safety of the motor vehicle, makes it even more difficult to carry out this task.

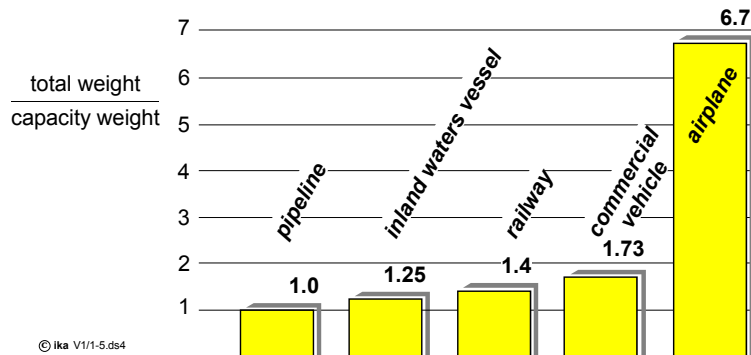


Fig. 1-5: Proportion of total weight to capacity weight for different transport systems

A transportation system's share in the total traffic performance depends on its suitability for certain mayor freight groups as well as on the total traffic volume of the different mayor freight groups. For example, the coal transport's share in the total amount of freight has strongly diminished and to some extends one result is the railroad's stagnant transport performance.

Figure 1-6 shows the single transportation system's share in the total freight traffic performance. Looking at the freight traffic, the still increasing dominance of the motor vehicle becomes obvious. Long-distance freight transport with commercial vehicles rose from 146 billions tkm/a in 1993 to about 200 billions tkm/a in 1995 and in the same period of time railroad transport increased from 65 to about 70 billion tkm/a. At the moment the percentage of both transportation systems in the total amount of traffic (short- and long-distance freight traffic) amounts to approx. 64% for commercial vehicles and 17% for railroad.

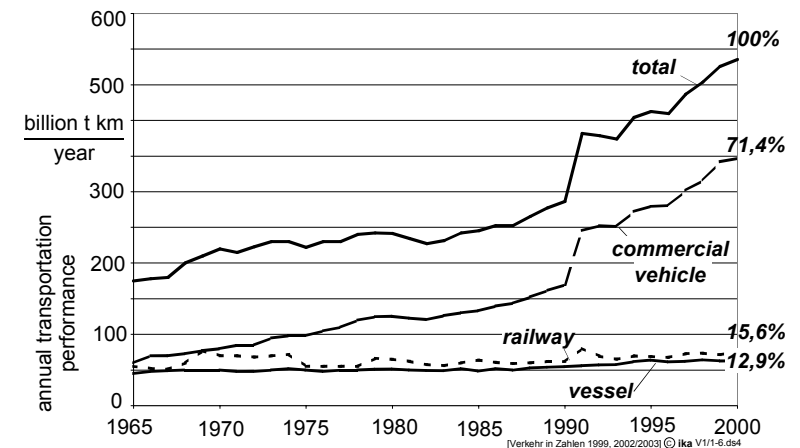


Fig. 1-6: Development of transportation performance in freight traffic
(Source: Verkehr in Zahlen 1996, 2002/2003)

The real advantage of long-distance road traffic is that it completely covers big areas without further expense, while the railroad can only be efficient if a big amount of traffic can be accumulated on a few tracks. For this reason it is strived for a stronger compound of these systems that combines the particular specific advantages. The „rolling country road“ represents an approach to do so. In this case, for certain track sections, entire trucks are loaded onto the railroad which-at the place of destination-take over the further distribution of goods (e.g. transit over the Alpes).

1.1.2 Energy demand

Up to now the traffic development is characterized by a distinct and constant rise in traffic performance, most of all in passenger traffic by means of motor vehicles. A reason for this is that the primary energy for motor vehicles, crude oil, has so far been rather cheap and was available in a sufficient amount. However, the steadily increasing road traffic performance goes hand in hand with rising negative side effects. Changes of essential basic conditions, e.g. concerning the availability of various resources overlap this performance.

The main objectives of the road traffic's and motor vehicles' future development will therefore consist of a continuous reduction of the energy consumption, to substitute the primary source of energy, crude oil, partly and to reduce harmful effects on the environment.

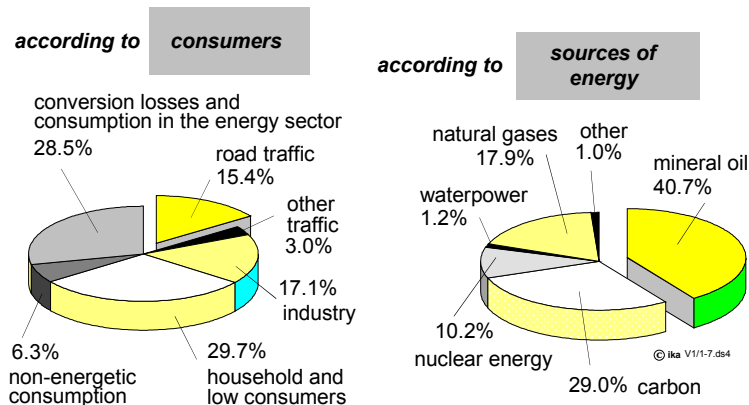


Fig. 1-7: Consumption of primary energy in the Federal Republic of Germany 1993

As Fig. 1-7 shows, the demand for primary energy in the Federal Republic of Germany is mainly covered by crude oil. The road traffic's share in the total demand for primary energy amounted to 15.4% in 1993. By means of the refinery and the energy converters automobile or bus, mineral oil is—with considerable loss of conversion—converted into mechanical energy at the driving axles. As Fig. 1-8 indicates, only 9.5% of mechanical driving energy emerge from a total of a 100% of primary energy used for short-distance passenger traffic. Therefore it is still an important task to improve the energy outcome, particularly that one of automobiles.

Fig. 1-9 shows the drive train losses at the particular components of an automobile's drive train. It becomes obvious by the figure that only a third of the chemical energy feed into the motor is converted into mechanical energy. A further third is lost as thermal energy when cooling down the engine. The remaining third of the chemical energy from the gasoline is lost as thermal loss by the exhaust emission.

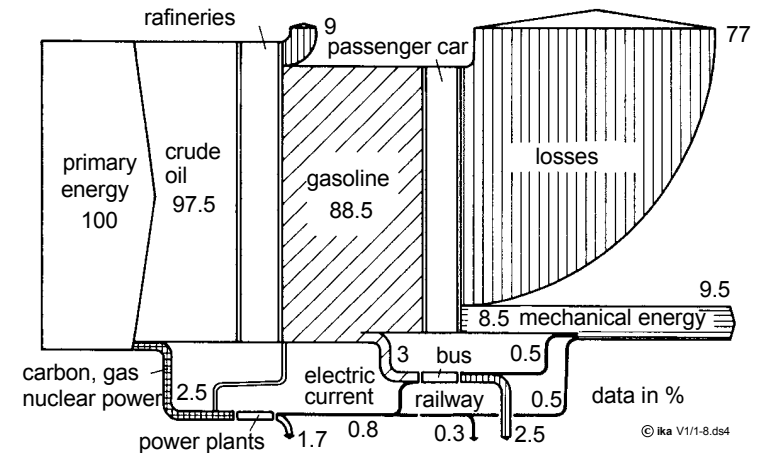


Fig. 1-8: Energy stream of short-distance traffic

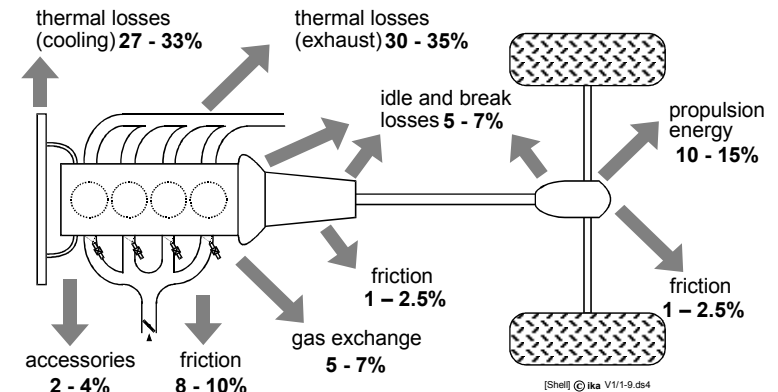


Fig. 1-9: Losses in a passenger car's drive train (Shell)

Since the 70s there has been a broad public and political consciousness for questions concerning the energy supply. During the two oil crises of the years 1973/74 and 1978/79 the price for crude oil increased tenfold within a few years. Meanwhile the price level has noticeably decreased. However, since then it has been realized that the world's crude oil reserves are limited, although over the past years more new oil deposits were found than oil was produced. Considering today's consumption, the availability of crude oil is about 50 years when taking the deposits

as a basis that can be exploited economically. Considering all secured and assumed reserves it amounts to about 200 years.

1.1.3 Influence on environment

Besides the reduction of energy and raw material resources, the use of motor vehicles also leads to environmental pollution, particularly due to exhaust- and noise emission. Burning fossil fuel, water and carbon dioxide are the main substances that are released. Besides these two emissions, a number of various harmful substances (pollutants) is produced.

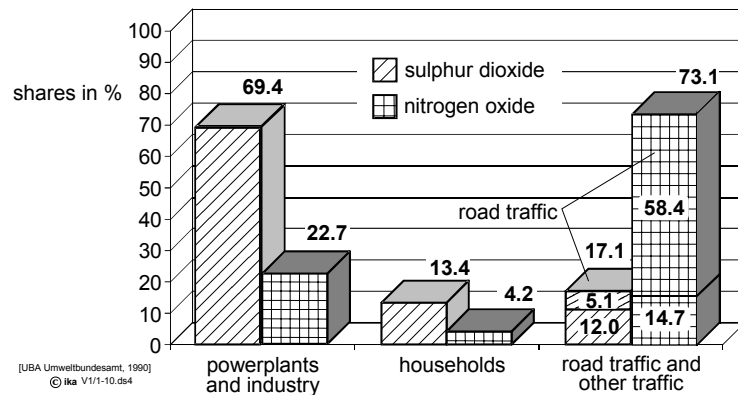


Fig. 1-10: Shares in sulphur and nitrogen oxide emission
(Source: Umweltbundesamt 1990)

The exhaust-gas components that are mentioned above all when speaking of forest dying, sulphur dioxide and nitrogen oxide, are emitted by various responsible parties (Fig. 1-10). Traffic is just slightly involved in the sulphur dioxide emission (sulphurous diesel fuel). By contrast, nitrogen oxides are released by motor vehicles on a grand scale and it turns out to be exceptionally difficult to reduce them.

A reduction of the nitrogen oxide emission by working on the combustion engine itself basically leads to a deterioration of the degree of efficiency. Since it is unknown that significant improvements concerning the quality of combustion engines exhaust emissions can be achieved by research progress, the future emission limits set up by law can only be kept by implementing subsequent emission control systems (ECS).

Besides the emission of harmful substances in regard to the greenhouse effect, also the CO₂-emission has turned out to be a serious problem. The CO₂-emission directly

depends on the fuel consumption and cannot be influenced by the exhaust emission's after treatment. Hence, the development of motor vehicles with a low consumption is a challenge with top priority.

Not only the propulsion defines the environmental compatibility of the vehicle. Also the structure of manufacturing processes and the possibilities of used car-recycling gain increasingly in importance. The automobile's future will decisively depend on the fact if its indisputable high benefit can be combined with the requirements of an intact environment.

1.2 Economic aspects of the Motor Vehicle

1.2.1 Eco-Political Significance

Germany's motor vehicle industry has gained major economic significance. Fig. 1-11 shows the sales figures of the motor vehicle industry. From 1960 to 1991 sales-adjusted by inflation-approximately quadrupled. The significance, also from the point of view of its contribution to the gross national product, has steadily increased from 3.5% to 8.2%. Today nearly every 7th employed person in the Federal Republic of Germany depends directly or indirectly on the automotive industry. Early in 1997 only about 570,000 employees directly participated in the manufacture of vehicles (Fig.1-12). It can also be noticed that the automobile industry is very sensitive to cyclical slumps, such as the ones of the years 1967, 1974, 1981, and 1992.

Compared to the investment in road construction and bridge building, public investment in vehicular infrastructure has significantly diminished since the 70s, though about 25% of the country's total tax revenue meanwhile comes from the automotive sector.

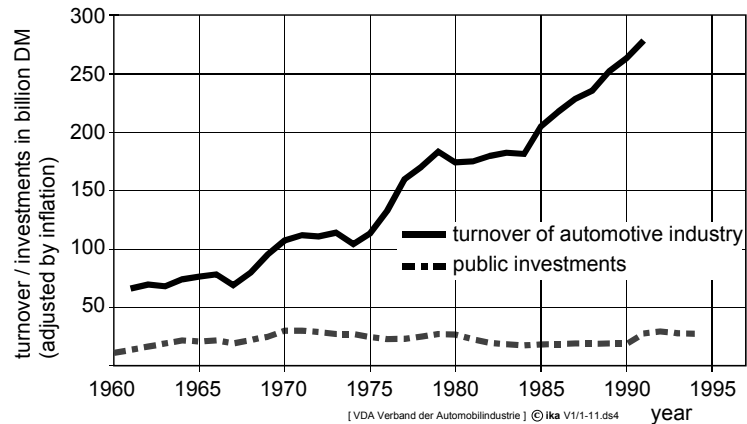


Fig. 1-11: Turnover in automobile manufacturing and public investment in road construction and bridge building (Source: Verband der Automobilindustrie 1994)

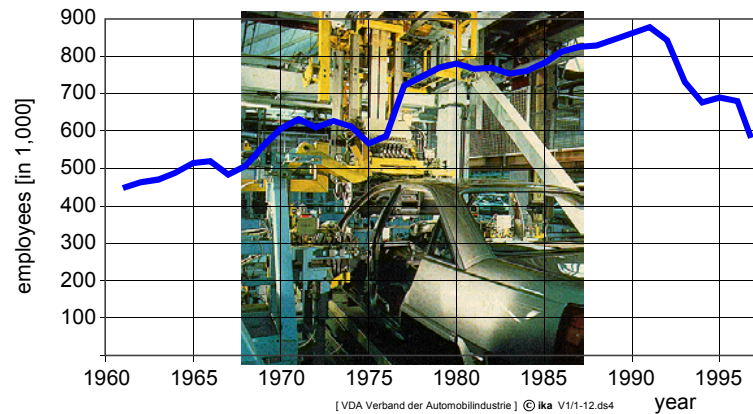


Abb. 1-12: Number of people employed in automobile manufacturing
(Source: Verband der Automobilindustrie 1997)

Fig. 1-13 shows the break-up of gasoline prices at present. An essential reason for today's traffic infrastructure being overloaded is the imbalance that exists between the prosperity of private enterprises and public restrictions. This reflects the attitude of many drivers who personally appreciate the advantages of the automobile, but have a skeptical view of its enormous distribution and accordingly accept restrictive political decisions.

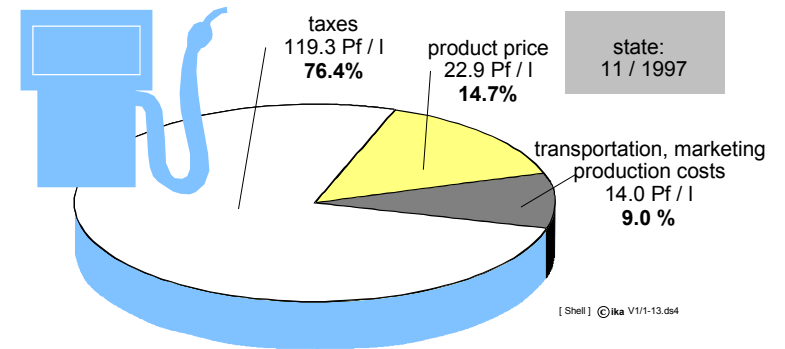


Fig. 1-13: Break-up of the price of Gasolene (Source: Shell)

1.2.2 Costs of a Motor Vehicle

In the preceding section the eco-political significance of the motor vehicle was already pointed out. The basic eco-political conditions on the vehicular traffic system are on a large scale, the result of political decisions. The manufacturers and users of motor vehicles however, mainly consider the vehicle from the economic point of view, provided that emotional aspects do not dominate as is the case with some automobiles that are privately used.

a) Production Costs - Sales Prices

The private automobile is a so-called high-income commodity, which means that with an increasing income, the budget, which an average household allocates to personalised transportation, increases disproportionately. In times of a growing national economy and increasing income, the automobile industry enjoys prosperity resulting from favorable possibilities for growth. This growth was reflected by the sales figures in Fig. 1-14 as well as by the sales prices in Fig. 1-15.

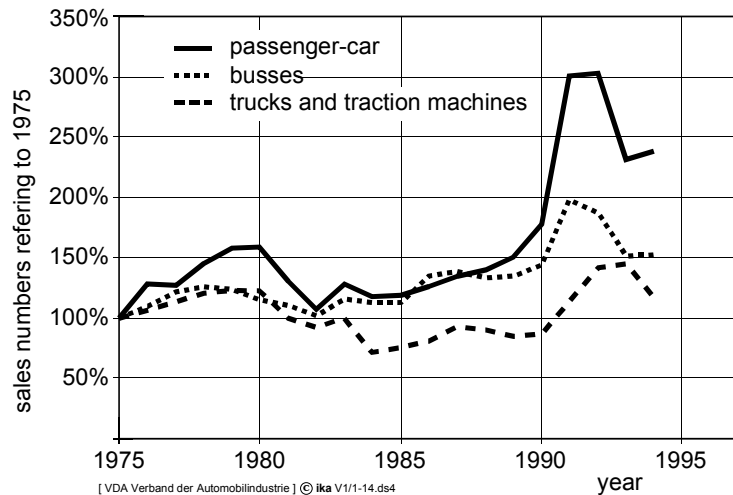


Fig. 1-14: Development of the automobiles' sales figures
(Source: Verband der Automobilindustrie 1994)

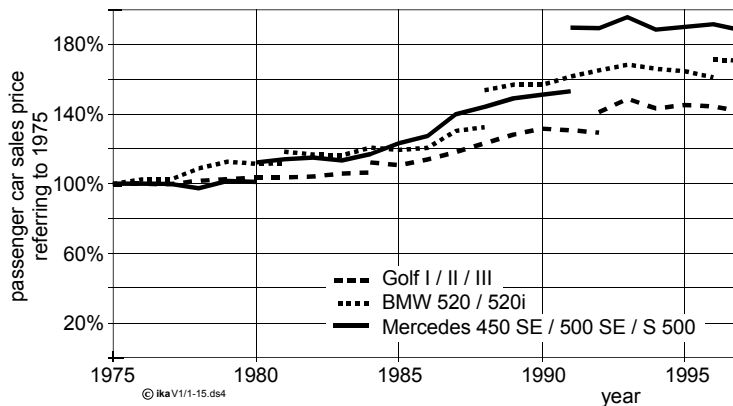


Fig. 1-15: Development of automobiles' sales prices referring to 1975

The sales prices of automobiles considered in the recent 20 years shows a sharp increase of prices in the 80s and a noticeable fall in the 90s. As an example, the picture shows a VW-Golf, a BMW 520 or 520i, and a Mercedes 450 SE or 500 SE or S 500. At the sight of stagnant income in broad classes of the society in recent years, rising price stability can be noticed. Meanwhile new models that offer less equipment are often offered cheaper than their predecessors.

The priority in the new development of vehicles today, is a reduction in production costs. Among others, some of the measures in order to achieve these objectives include the integration of various functions within component or modular systems, equal platforms for many different passenger car designs (VW/Audi) or V6 and V8 engines with fundamentally similar dimensions in order to produce them on the same assembly line (DB passenger car/truck).

Apart from the lowering of costs a simultaneous improvement of operation and quality is the target. However, most motor vehicle concepts and solutions are compromises. Moreover optimized production engineering, logistics, and work organization can lower costs. High production volumes have a particularly favourable influence on costs in vehicle design, primarily because development costs and overheads can be converted more positively and production plants utilised to capacity in a better way.

The large proportion of the eventual production costs are already accrued in the developmental phase, which amounts to about 70%. This implies that the people responsible for development, are even more than the technical engineers for production, not only responsible for the technical but also economic success of a new design.

The right sales pricing, of course, also plays an important role in the success or failure of a product. This pricing is normally not entirely based on production costs. Within a product palette, sales prices vary much more than production costs. Often the most expensive design is 2 or 3 times more expensive than the cheapest one. When imagining that a manufacturer would offer only a single design at an average price, the sense of such pricing becomes obvious. On the one hand the manufacturer would lose possible customers for whom the vehicle would be too expensive and on the other hand its price margin would remain unused in the case of those customers who would be willing to pay more. Therefore motorization and equipment variations cannot only be seen as a possibility for product distinction but also primarily for price distinction.

b) Running Costs

In the discussion of vehicle costs it has to be considered that motor vehicles can either be privately or commercially used. According to their use, they are classified from an economical point of view into, consumer goods - primarily passenger cars that are privately used or capital goods - commercial vehicles and passenger cars that are commercially used. This distinction is of no significance from the point of view of costs but essential, however, when it comes to taxation.

Costs can be distinguished into variable expenses depending on how much the vehicle is used and fixed expenses that already arise when owning a vehicle.

Fixed expenses:

- Capital costs
- Depreciation
- Labor costs*
- Administration*
- Tax
- Insurance

Variable expenses:

- Fuel
- Other working materials
- Maintenance and repairs
- Tires

Several aspects valid under the present basic conditions in the Federal Republic of Germany are to be explained using two examples. The cost structure for a VW-Golf and a Mercedes S 500 are shown. Table 1-1 -shows the underlying data. They are meant as approximate values.

Fig. 1-16 and Fig. 1-17 show the resulting annual costs. On the one hand it becomes obvious that even when considering a high annual mileage of 30.000 km (18.642 miles) as a basis, the proportion of gasoline costs for an upper-class vehicle is very low (15.9%). Even for a smaller middle-class vehicle, the proportion of gasoline costs amounts to only 22.8%. If it was possible to cut the fuel consumption of both vehicles in half, the total costs would only decrease by 8% or 11%. In this context the variation of fuel prices would be of interest (Fig. 1-18)

* only for commercial used cars

		Golf GL	MB S 500
Service life	[years]	4	4
Purchase price	[DM]	25,000	150,000
Residual price after use	[DM]	13,000	60,000
Cost of a tire set	[DM]	600	1,500
Calculation rate of interest	[%]	8.0	8.0
Annual mileage	[km]	15,000	30,000
Annual taxes	[DM]	184.80	660
Annual insurance premium	[DM]	1,000	5,000
Fuel price	[DM/l]	1.60	1.60
Fuel consumption	[l/100km]	7	15
Surcharge on engine oil	[%]	5	5
Surcharge on service and repairs	[%]	25	25
Tire life	[km]	50,000	50,000

Fig. 1-1: Cost structure of a passenger car used for 4 years

In contrast to general opinion, fuel prices are presently on a low level. This indicates once again that the necessity for decreased consumption is at the moment not due to economics but from an environmental point of view.

As a further variation, Fig. 1-16 and Fig. 1-17 show the running costs specific to 4-year-old used-cars. Due to the high depreciation of a new car, an upper-class automobile gets a significant reduction in costs. Here it is not considered that maintenance and repair expense cannot be precisely forecasted if the automobile gets older and that the possibilities of tax depreciation can strongly reduce the effective costs of new automobiles when they are commercially used.

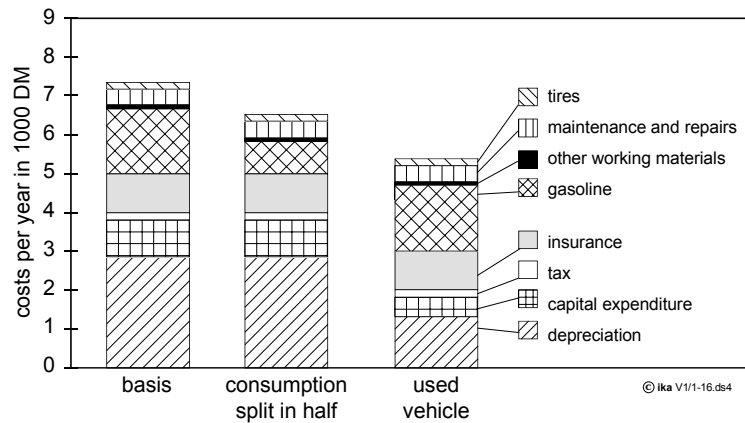


Fig. 1-16: Cost structure of a passenger car, example VW-Golf

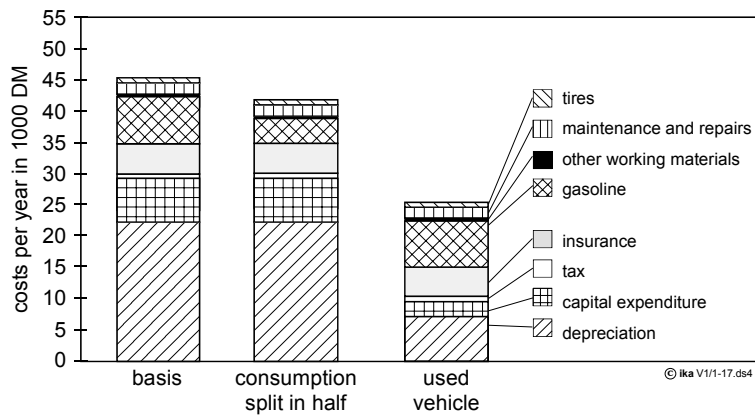
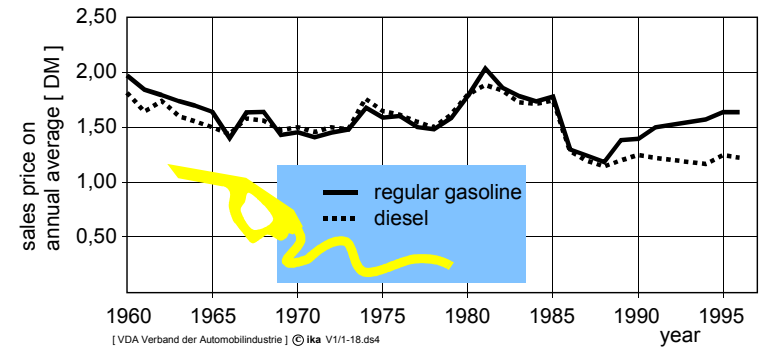


Fig. 1-17: Cost structure of a passenger car, example MB S 500

Fig. 1-18: Development of gas prices
(Source: Verband der Automobilindustrie 1996)

Economic considerations are of more significance for commercial vehicles as compared to passenger cars. As capital goods they are exclusively purchased in order to transport goods or passengers. The purchase decision is normally not influenced by non-economic aspects. Accordingly, the development of commercial vehicles is nearly exclusively undertaken keeping in mind optimum profitability. It is true that technically expensive solutions can increase the vehicle's utility value but at the same time, it is associated with higher production costs and more or less higher sales prices. Thus in every individual case, careful attention has to be paid in deciding which measures make sense from the customer's point of view and which do not. For the manufacturer, this poses a rather difficult challenge since he cannot optimize a vehicle model or a vehicle production series to meet individual applications, but has to find a compromise for many different applications, e.g. for long-distance traffic, delivery traffic, and construction traffic.

Today the further development of vehicles becomes less than ever an end in itself, but is absolutely necessary in order to maintain its high profitability and social acceptance.

2 Power and Energy Demand

The power and energy demand necessary in order to set a motor vehicle in motion is determined by the resistive forces on the road that have to be overcome. These resistive forces are classified into road resistances limited to a steady-state driving condition, meaning at a constant speed, and road resistances that are generated by vehicle acceleration.

The resistance forces that act on a vehicle moving at a constant speed, can be classified into three groups:

- Wheel resistance,
- Aerodynamic drag,
- Climbing resistance.

In a transient driving condition, the additional component of driving resistance includes:

- Acceleration resistance.

2.1 Wheel resistance

The wheel resistance comprises all those forces of resistance originating at the rolling wheel. The overall wheel resistance is composed of the following component resistances:

- Component from the Tire,
- Component from the Road,
- Component resulting from Slip,
- Component of Bearing Friction.

The component of wheel resistance resulting from bearing friction is comparatively small and shall not be dealt with here explicitly.

2.1.1 Component from the Tire

When the pneumatic tire rolls along a straight path on an ideal road (flat and dry), meaning that the circumferential centerline and rolling direction coincide, a force of resistance contrary to the rolling direction is induced.

This force of resistance mostly depends on the structure of the tire. This can be further decomposed into different components which are:

- Flexing resistance,
- Frictional resistance,
- Fan resistance.

a) Flexing resistance

The deformation (flexion) of a pneumatic tire as a result of static compression and expansion, results in a loss of work because of internal friction. If the tire rolls under a load, the process of compression and expansion continually repeats for each circumferential point on the tire. This is illustrated by a wheel replacement system (Fig. 2-1) whose circumference is held in position against the rim by linear spring-and-damper elements. In addition, the tire tread can also be imagined to include a composition of elemental springs and dampers. These springs and dampers come into play when the tire contact patch is run through (tire contact patch on the road). By this, additional frictional influences are created that have to be ascribed to the flexing resistance. The spring and damper characteristics of the tire tread also have an effect on the tire's tractive power on the road. The choice of a low-damping running band mixture simultaneously leads to a deterioration of the tractive friction.

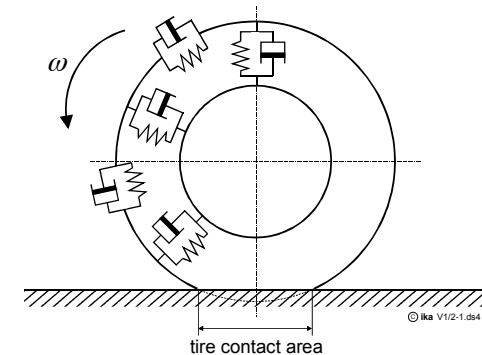


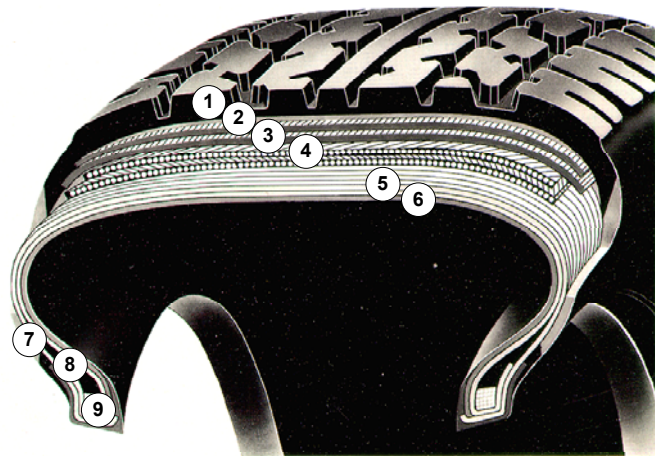
Fig. 2-1: Wheel replacement model

When the replacement model rotates, in each of the “elementary vibration dampers” work is transformed into heat. The resulting flexing resistance corresponds to the quotient of accomplished damping work and the covered distance:

$$F_{R \text{ flex}} = \frac{\text{damping work}}{\text{distance}} \quad \text{Eq. 2-1}$$

Experience has shown that when comparing tires with the same ply material (see Fig. 2-2) the tire with the larger number of plies has the higher damping, since the relative motion of the plies in relation to each other generates damping work.

There is a significant difference between bias ply and radial ply tires. The so-called cord angle, defined as the angle formed between the single cord plies of a carcass fabric with the center line of the circumference of the wheel, plays a decisive role. The cord angle in radial tires amounts to 85° - 90° while with bias ply tires it varies between 20° - 40° . Fig. 2-2 shows the design of a passenger car radial tire and the function of the individual components.



- | | |
|---|--|
| ① Running strip: profile and mixture influence the characteristics | ⑥ Interior conductor: replaces the tube |
| ② Base: lowers the rolling resistance | ⑦ Side section: protects the carcass against damages |
| ③ Nylon bindings: increase the ability for maximum speeds | ⑧ Bead filler: serves for steering accuracy and driving stability |
| ④ Steel cord bracing ply: increase the form- and driving stability | ⑨ Bead core: provides a firm rim position |
| ⑤ Carcass: ties up the tire filling pressure | |
- [mot - Spezial, Räder und Reifen]
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Fig. 2-2: Design of a radial ply tire (Source: mot-Spezial, Räder und Reifen)

While radial tires are reinforced in the tread area by a suitable belt, in the case of bias plies, it is necessary to stiffen or increase the number of plies in the entire carcass including the load carrying sidewall. This explains the larger spring stiffness of a bias tire. This leads to a simultaneous increase in internal friction, a stronger damping,

and thus a larger flexing resistance. Tire damping tests at different rolling speeds have shown that damping decreases at increasing speeds (Fig. 2-3).

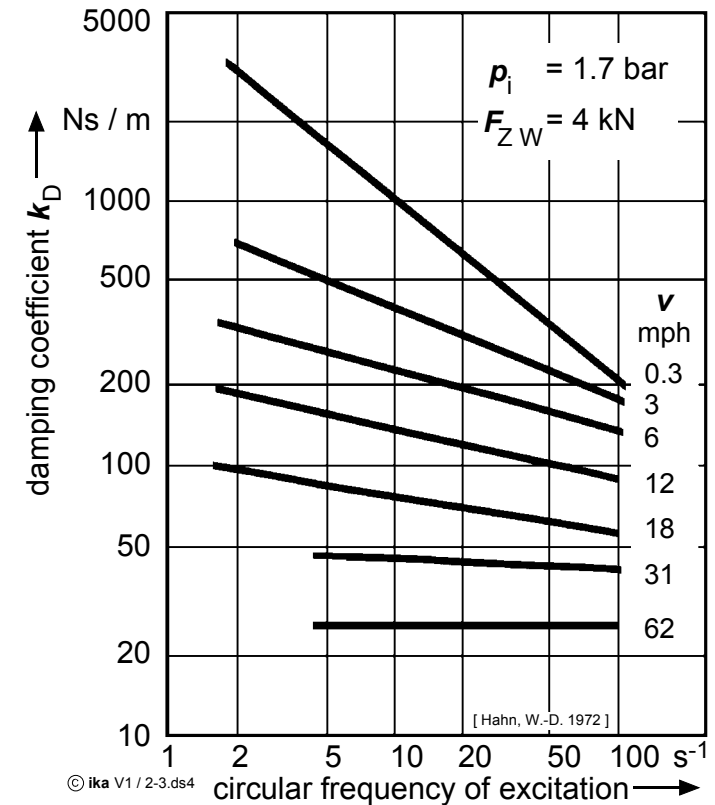


Fig. 2-3: Damping coefficients of a passenger car's tire (Source: Hahn, W.-D. 1972)

At increasing speeds, the effect that lowers the flexing resistance is superimposed by an opposite effect: the deflection of the tire rolling under load in the contact patch causes the tire circumference to vibrate beyond the contact patch as a result of forces of inertia that increase with speed. The damping of this vibration produces heat which results in an increased flexing resistance.

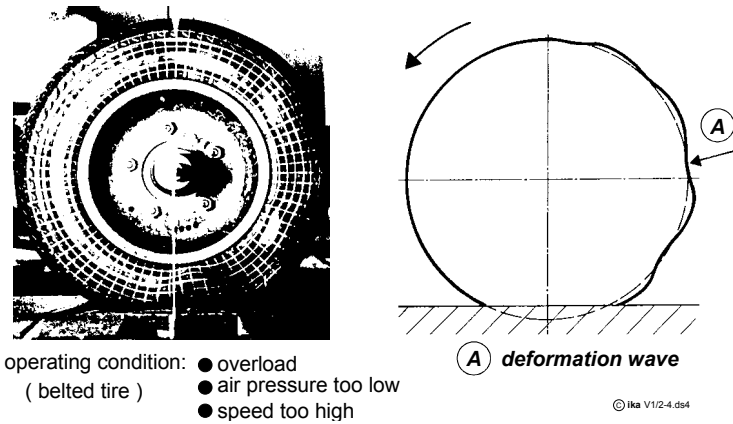


Fig. 2-4: Deformation wave formation

This process of formation of a so-called deformation wave behind the tire contact patch (Fig. 2-4) predominates the effect of the decreasing damping in its influence on the characteristic of flexing resistance depending on speed. Based on tire design, the flexing resistance indicates a linear increase with a low gradient, up to a speed of approx. 35 m/s beyond which the characteristic becomes progressive.

In order to emphasize its significance, we can already say that the flexing resistance is normally the most significant component of the wheel resistance.

b) Frictional resistance

In the wheel replacement model shown in Fig. 2-1 the spring-damper elements of the running strip pass through the tire contact patch. At this point the circumference of the tire is compressed to the length of its chord width, the tire contact length. This leads to relative motion between the road and the running band at the tire contact patch, which is the so-called Slip in longitudinal as well as transversal direction.

Slip causes abrasion. In this case energy is transformed that has to be overcome as additional resistance by the power train.

c) Fan resistance

The rotational motion of the tire leads to flow losses that can however only be considered significant within the context of air-flow around the entire vehicle. As a result, this component they are mostly added up to the overall aerodynamic drag (see Chapter 2.2). This form of resistance will not be dealt with further.

d) Rolling resistance coefficient

In summary, the component of wheel resistance resulting from the tire is the sum of flexion resistance $F_{R \text{ flex}}$, frictional resistance $F_{R \text{ frict}}$, and fan resistance $F_{R \text{ fan}}$. This sum is also called Rolling Resistance.

$$F_{R \text{ roll}} = F_{R \text{ flex}} + F_{R \text{ frict}} + F_{R \text{ fan}} \quad \text{Eq. 2-2}$$

This forms a useful summary because in practice the single components of flexion- and frictional resistance cannot be separated.

Fig. 2-5 shows the characteristic of the rolling resistance force over the wheel load for a commercial vehicle tire.

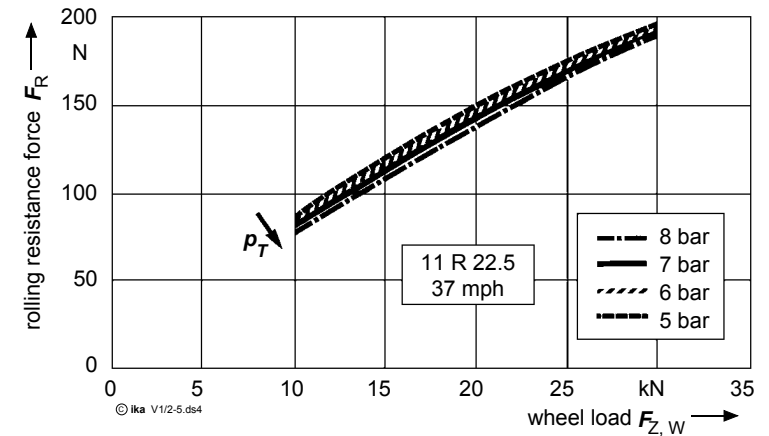


Fig. 2-5: Load dependence of rolling resistance force for a commercial vehicle's radial tire

As a result of a nearly linear characteristic of the rolling resistance force over the wheel load, an index relative to the load can be defined, the dimensionless coefficient of rolling resistance f_{Rroll} :

$$f_{Rroll} = \frac{F_{Rroll}}{F_{Z,W}} \quad \text{Eq. 2-3}$$

As mentioned above, the emergence of the rolling resistance was derived with the help of the spring motions in the tire sidewall and the tire tread. In fact, measurements of pressure distributions in the tire contact patch in longitudinal and transverse directions show asymmetrical pressure distributions (Fig. 2-6).

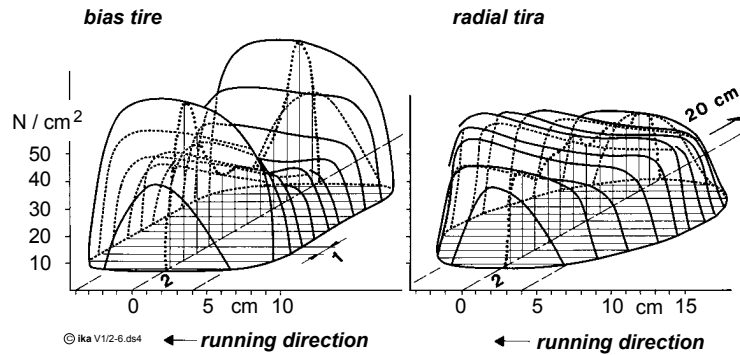


Fig. 2-6: Pressure distribution in the tire contact patch of a bias ply (left) and radial ply tire

When only the longitudinal section in the wheel center is considered, the pressure distribution shown in Fig. 2-7 results.

Fig. 2-8 shows the resulting forces and moments that act on the wheel, vehicle body, and road. A non-driven wheel is considered.

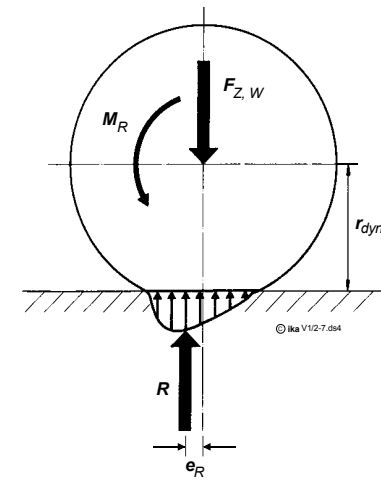


Fig. 2-7: Forces and moments on the wheel

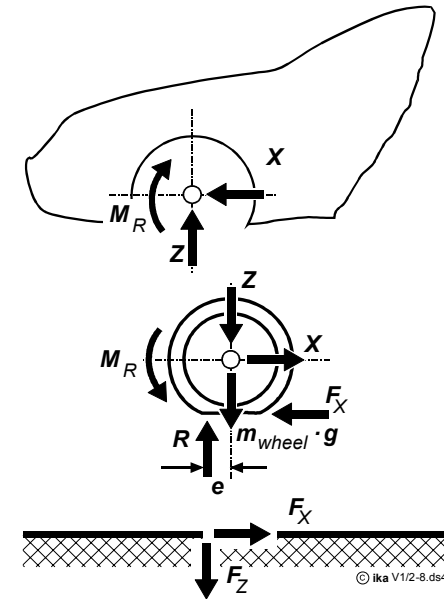


Fig. 2-8: Forces and moments on the vehicle body, wheel and road

If the individual vertical forces acting at the tire contact patch are added up, the reaction force R which is equal in magnitude to the wheel load results. Due to the asymmetrical pressure distribution in the plane of the contact patch, the force acts a point around the eccentricity e_R in front of the wheel axis. A reaction moment

$$M_{R\text{roll}} = R \cdot e_R = F_{Z,W} \cdot e_R \quad \text{Eq. 2-4}$$

counteracts the rotational motion. Hence, to set the wheel in motion, a horizontal force, which when multiplied by the dynamic tire radius r_{dyn} , corresponds to the moment of the vertical forces, is required. This force, in magnitude, corresponds to the rolling resistance force:

$$M_{R\text{roll}} = F_{R\text{roll}} \cdot r_{\text{dyn}} \quad \text{Eq. 2-5}$$

As a simplification, the rolling resistance coefficient can be estimated as:

$$F_{R\text{roll}} \cdot r_{\text{dyn}} = F_{Z,W} \cdot e_R \quad \text{Eq. 2-6}$$

where: $F_{R\text{roll}} = f_R \cdot F_{Z,W}$ Eq. 2-7

resulting in: $f_{R\text{roll}} = \frac{e_R}{r_{\text{dyn}}}$ Eq. 2-8

Within the framework of typical motor vehicle calculations, it is assumed that the rolling resistance is constant over the wheel load and driving speed. When considering the load dependence more precisely (Fig. 2-5), a degressively increasing characteristic of the rolling resistance force results. This leads to a decreasing rolling resistance coefficient over the wheel load, as shown in Fig. 2-9 for a truck radial ply tire.

Simultaneously Fig. 2-9 shows the influence of tire pressure on the rolling resistance coefficient. A decreasing resistance co-efficient results with rising air pressure. Since an increased internal pressure leads to a further stiffening of the tire, the deflection decreases at a steady wheel load. This leads to a lowering of the expended flexing energy and due to the smaller contact patch to a decline of the frictional component of resistance.

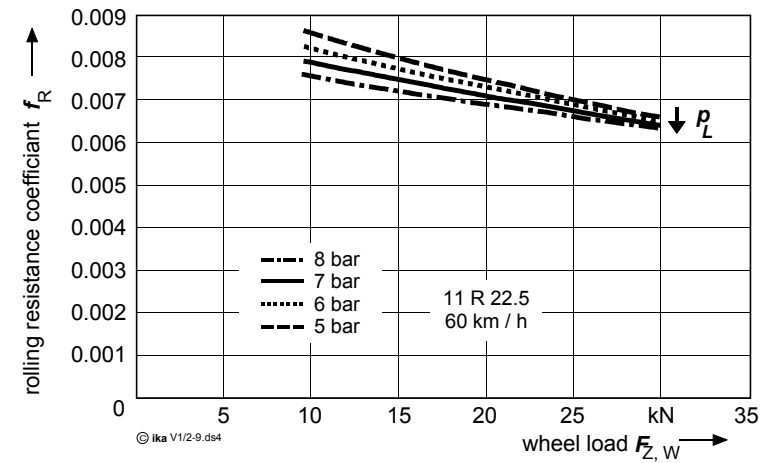


Fig. 2-9: Rolling resistance coefficient of a truck's radial ply tire as a function of wheel load and internal pressure

Fig. 2-10 shows the influence of velocity on the rolling resistance. The initially gradually rising characteristic of the rolling resistance coefficient, which rises progressively at a fluctuating speed range according to interpretation, is due to the superimposed effect of the deformation wave on the flexion resistance. This effect increases with velocity.

In the case of H-type (up to 210 km/h (131 mph)) and V-type (up to 240 km/h (149 mph)) tires, the increased tire rigidity attenuates the influence of the deformation wave formation. Thus the progressively rising phase of the rolling resistance begins at a higher speed and in a less noticeable way.

For computer simulations of rolling resistance, the dependence of rolling resistance on velocity can be approached by an investigation, which is obtained empirically

$$f_{R\text{roll}} = C_0 + C_1 \cdot v + C_2 \cdot v^4 \quad \text{Eq. 2-9}$$

while tire-specific values are used for the constants C_0 , C_1 , and C_2 . For a reference speed of $v^* = 100 \text{ km/h}$ (62 mph), for a HR-type tire the constants are approximately:

$$C_0 \approx 9,91 \cdot 10^{-3} \quad C_1 \approx 1,95 \cdot 10^{-5} \quad C_2 \approx 1,76 \cdot 10^{-9}$$

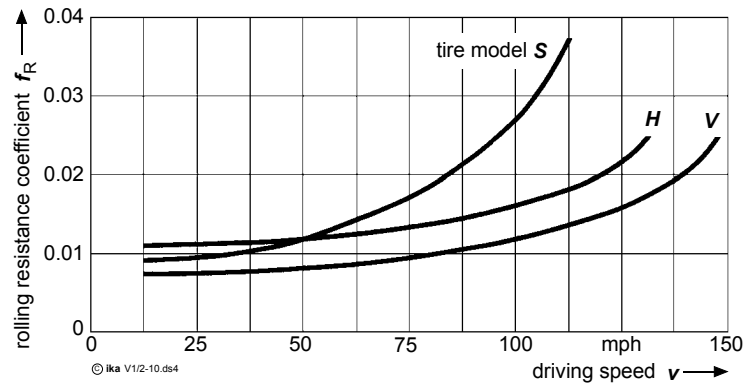


Fig. 2-10: Rolling resistance coefficients for different radial ply tires

In summary, the following can be stated:

Apart from the external influences - velocity, wheel load, and tire pressure, the rolling resistance depends on tire design, materials and the rubber mixture used. Thus the radial ply tire generally has a smaller rolling resistance compared the bias ply tire. Moreover the tire tread configuration has an influence on the rolling resistance. Small tread depths and a fine tread configuration reduce the resistance force while the compensating effect of the fine-structured tire treads decreases with increasing speed.

The magnitudes of the individual components of resistance, i.e. flexing, frictional, and aerodynamic resistance cannot be precisely recorded by measurements. It can only be said with certainty that the flexing resistance forms the largest component of the rolling resistance. In order to measure the total rolling resistance, there are basically two different methods available. The road test has advantages from the point of view of a realistic road and basic conditions. It is however difficult to maintain the given test parameters as a result of external sources of disturbance such as weather conditions, non uniform or changing road surface, and traffic, all of which are necessary for reproduceable tire tests.

This problem does not occur when measurements are carried out on stationary tire test benches. In this case the tire which is kept stationary, is brought in contact with a movable rolling surface area. The data is acquired by power sensors mounted on links holding the wheel. Three types of rolling surface areas are used for these applications:

- A roller with outer bearing surface,
- A roller with inside bearing surface,
- A smooth band moved by two rolls.

A frequently used and cost-effective method of testing tires involves the roller test bench with outer bearing surface. Besides the high loading capacity and compactness, the large space offered on the wheel allows various wheel guide concepts and wheel positions as well as convenient tire mounting.

Due the effect of the centrifugal force, it is difficult to fix the different road surfaces onto the roller. Tests with a wet tire tread are only partially possible.

The effect of centrifugal force on the other hand supports the fixing of the tire tread pavement in test benches with inner bearing surfaces. Hence, this test bench is especially suitable when measurements involve differing road surfaces and to determine the wet rolling behaviour of the tire. The limited space on the wheel complicates tire mounting and wheel control.

Measuring errors result in all roller test benches due to the cylindrical profile of the bearing surface. In contrast to flat roads, the tire contact area and the deflection extends at the same load. The result is increasing frictional and flexural resistances. The error can be minimised by roller radii which are relatively large compared to the wheel radius. Correction factors have to be introduced in order to apply the measurement results for flat roads.

Flat track test benches realize a smooth rolling surface to a large extent. Besides they offer unlimited space for wheel control and wheel motions as well as for tire mounting. By the change of band, changed road surfaces can be realized within certain limits. In the same way wet running tests are possible. Belt guidance by rolls, however, requires a large technical expense since bearing surface vibrations can produce measurement errors. Furthermore band abrasion increases the running costs.

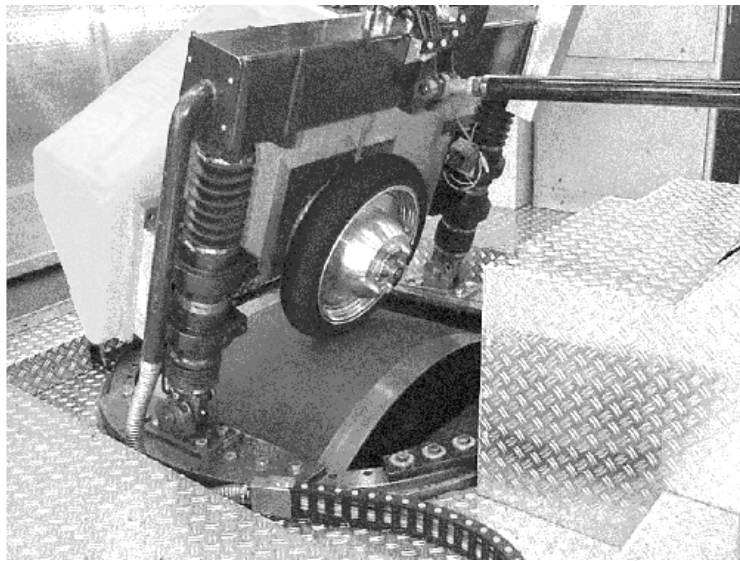
Fig. 2-11 shows the three most important types of test benches with their advantages and disadvantages.

test stand design	advantages	disadvantages
 drum with outside path	sufficient space, simple mounting of tires	difficult wet running tests
 drum with inside path	exchangeable tire tread, wet running possible	limited space, difficult mounting of tires
 flatpath teststand	even base	difficult bend guiding, band abrasion by vibrations

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Fig. 2-11: Construction forms of tire test stands

Fig. 2-12 shows the roller test bench for motorcycle and car tires with external running surface design used at the Institute for Automotive Engineering in Aachen (ika).



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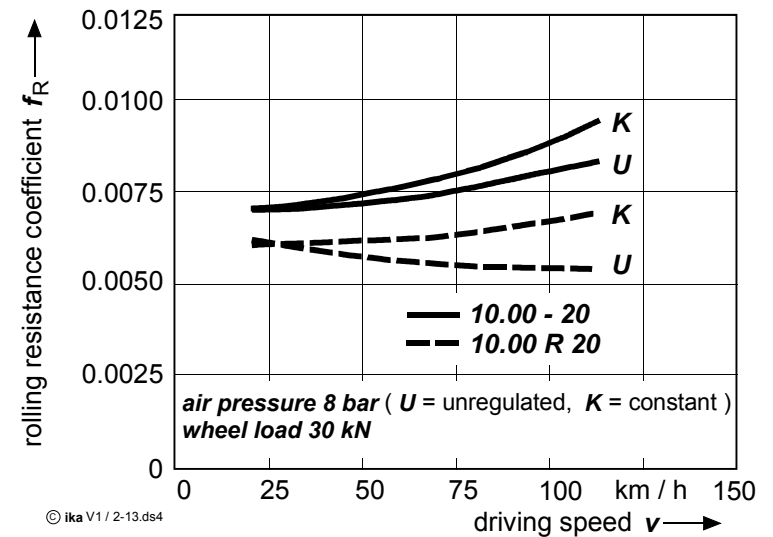
Fig. 2-12: Roller tire test bench with external running surface at the ika

The characteristic curve of the co-efficient of rolling resistance dependent on the velocity basically depends on the measurement conditions. There are two ways of measurement:

1. The nominal pressure is set on a stationary tire or a tire rolling at a constant velocity. Subsequently each measuring point is started without air pressure control. By the tire's flexing energy the air temperature inside the tire increases. The air pressure increases.
2. The inside pressure is continuously controlled during the entire running time and adjusted to the nominal pressure.

The second method eliminates the effect of a further varying parameter, the tire air pressure, while the former method gets closer to the tire's real application at the vehicle. However, other influences such as cooling air stream are not considered.

Comparing both measuring results (Fig. 2-13), the rolling resistance values are smaller at an unregulated internal pressure against the ones with subsequently adjusted tire air pressure. This can be explained by a drop in the rolling resistance due to the increased internal pressure.



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Fig. 2-13: Influence of the regulated and unregulated tire pressure on the rolling resistance of commercial vehicle tires

From this chapter it is evident that the wheel resistance is influenced by the characteristics of the tire. In addition the tire performs a significant role in the scope of transverse and vertical dynamics of vehicles (springs and transmission of lateral forces). The concrete composition of the rubber mixture (Fig. 2-14), the tire tread pattern, and the inner structure (Fig. 2-2) are decisive factors representing the quality of a tire

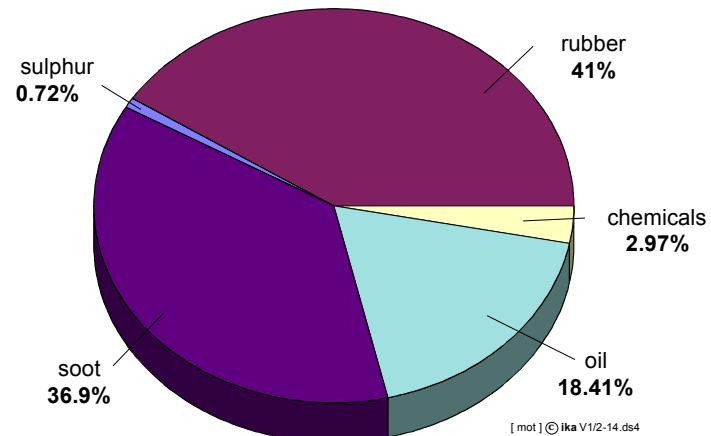


Fig. 2-14: Rubber mixture for a tire tread (Uniroyal Rallye 440)

Finally, the differing demands made on a modern tire and progress made in tire development are presented briefly.

Fig. 2-15 shows the various objectives that a manufacturer had to fulfill in the years 1960, 1970, and 1992. It can be seen that bias tires had very good convenience characteristics earlier, but were not sufficiently fulfill the characteristics relevant for vehicle dynamics. It is the opposite in the case of earlier radial tires while the modern tire represents a good compromise concerning the fulfilment of all requirements. Meanwhile some manufacturers offer different tire sets that allow the driver to set his main focus on certain criteria (e.g. comfort or sporty).

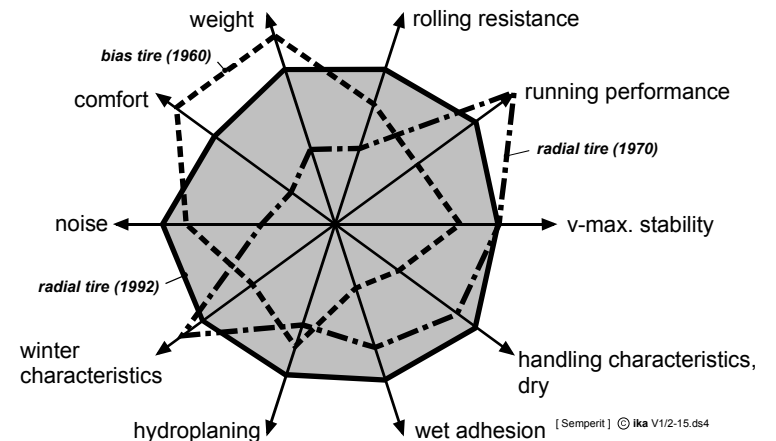


Fig. 2-15: Development aims for tire development (Semperit)

Economy is the top priority in the development of truck tires. Reducing the rolling resistance plays a significant role as a result of the decreased fuel consumption. Recent developments have achieved rolling resistance coefficients up to $f_R=0.005$ (Fig. 2-16).



Fig. 2-16: Rolling resistance-minimized truck tire (Continental)

2.1.2 Component from the Road

Uneven, plastic, and wet roads contribute to a rise in rolling resistance.

a) Uneven roads

Minor unevennesses in the road are cushioned by the tire and absorbed. Additionally the entire wheel moves relative to the vehicle structure over spring-and-damper elements. In this case kinetic energy is transformed into heat in the tire as well as in the body damper. This means that the spring work regained for the movement when rebounding is smaller by a magnitude equal to the damping work when compared to the work done initially during compression. Fig. 2-17 provides an explanation by the simple example of a wheel without tire.

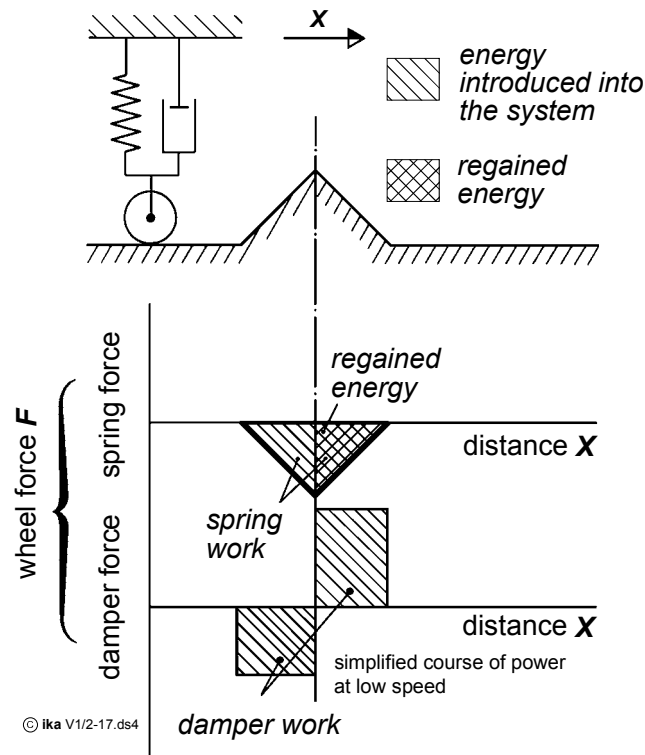


Fig. 2-17: Energy recovery of a wheel suspension when running over an obstacle

Due to regained energy, the spring work does not have an effect on the wheel resistance, whereas the damper work ΔW relative to the covered distance constitutes the additional component of wheel resistance due to an uneven road $F_{R \text{ uneven}}$:

$$F_{R \text{ uneven}} = \frac{\sum_0^x \Delta W}{X} \quad \text{Eq. 2-10}$$

b) Plastic road surfaces

In addition to rigid or slightly elastic pavements such as asphalt or concrete that do not permanently deform as a result of a wheel rolling under a regular load, there are roads on which the rolling process of the loaded wheel produces a plastic deformation in the form of a visible trace, e.g. soil, sand, grass, or snow. These plastically deforming roads in contrast to firm bases, cause an additional wheel resistance force $F_{R \text{ plast}}$ which essentially consists of three partial resistances:

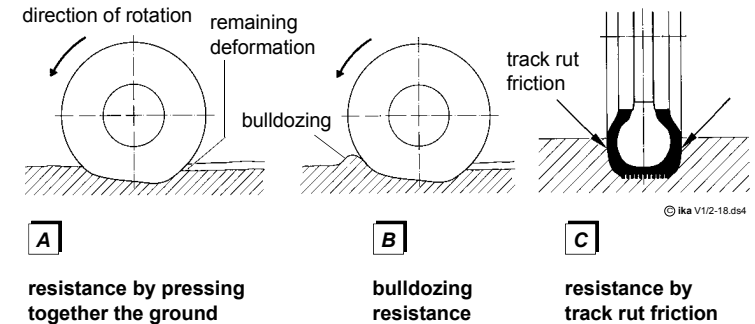


Fig. 2-18: Resistances on plastic road

- Resistance as a result of compaction

The deformation of the road surface caused by the wheel rolling under load on a plastic road, based on the type of surface, elastically forms back just to a small extent. A trace remains (Fig. 2-18).

- Bulldozing resistance

When the wheel rolls through a plastic road surface, masses of soil are pushed in front of the tire until they are either thickened or displaced to side. This displacement of the of the road material, into the drive direction as well as transverse to it, requires forces that are summed up by the term bulldozing resistance (Fig. 2-18).

- Resistance by track rut friction

A further but smaller partial resistance is produced due to friction between the tire sidewall and road material, the track rut friction (Fig. 2-18).

The additional resistance for plastic roads, in contrast to the rolling resistance on a rigid road, increases with an increasing tire pressure, since the tire contact area is lowered by an increased inside pressure and the tire sinks deeper into the ground (Fig. 2-19).

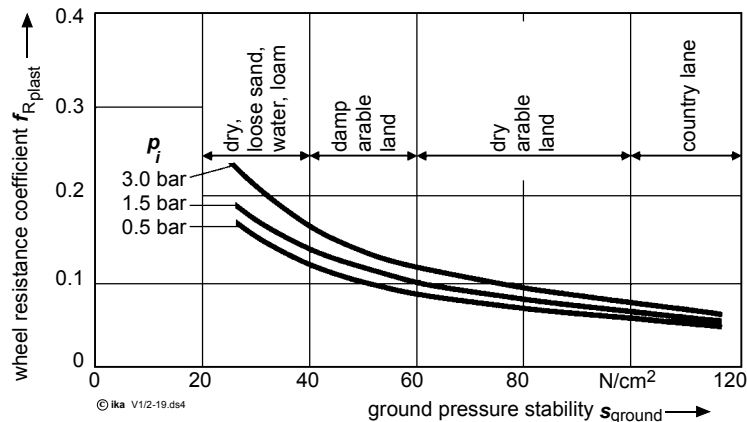


Fig. 2-19: Rolling resistance coefficient's dependence on the ground pressure solidity on plastic road

In total, the wheel resistance results from the sum of the resistive force generated as a result of compacting together and displacing the road surface, by track rut friction, and by the rolling resistance occurring on a rigid road:

$$F_R = F_{R\text{roll}} + F_{R\text{plast}} = F_{Z,W} \cdot (f_{R\text{roll}} + f_{R\text{plast}}) \quad \text{Eq. 2-11}$$

The co-efficients of wheel resistance derived from the wheel resistance are indicated for different roads. This co-efficient of resistance is simply called rolling resistance coefficient. As previously mentioned, we can infer that it represents a double usage of the term rolling resistance coefficient.

In the following the wheel resistance coefficients for different roads are listed:

Road Surface	f_R - value
new, firm asphalt; concrete; small pavement; cobblestone pavement	0.005 – 0.015
rolled, firm gravel; wear down, washboard asphalt	0.02 – 0.03
tarred, wear down, washboard gravel	0.03 – 0.04
very well dirt roads	0.04 – 0.05
dirt roads	0.05 – 0.15
Sand	0.15 – 0.35

Table 2-1: Rolling resistance coefficients depending on road surface

c) Wet roads

In order to obtain contact between the running strip of the tire and the road surface on a wet road, the tire has to break through the water layer. In order to displace the water, so-called baffle forces have to be superseded. The water layer and the section of the running strip in contact with the road for simplification may be subdivided into three zones (the approaching zone, the transition zone, and the contact zone) (Fig. 2-20). In the approaching zone, the principal part of the water is displaced in the running-in-area of the tire without coming into contact with the road. In the transition zone the tire is already deformed and partially comes into contact with the road. The contact zone describes the actual tire contact area, the direct contact surface between running surface and road that makes the energy transition possible. Only a negligible amount of water is displaced outwards by the tire profile.

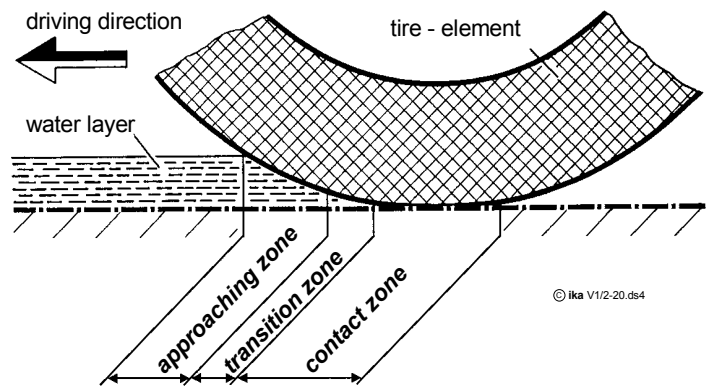


Fig. 2-20: Contact zones between a wet road and tires

As indicated by tests, the baffle resistance almost exclusively depends on the volume of water displaced in unit time, hence for a water level h , the tire width b , and the rolling velocity v_R . Tire design, air pressure, or wheel load have almost no or a very small influence.

In order to calculate the baffle resistance $F_{R \text{ baffr}}$, the following empirical relation can be used:

$$F_{R \text{ baffr}} = \left(\frac{w}{10} \cdot \frac{v_R}{N} \right)^E \quad \text{Eq. 2-12}$$

where:

$F_{R \text{ baffr}}$	=	baffle resistance	in N
w	=	tire width	in cm
v_R	=	velocity	in km/h

The magnitudes N and E depending on the water level are shown in Fig. 2-21.

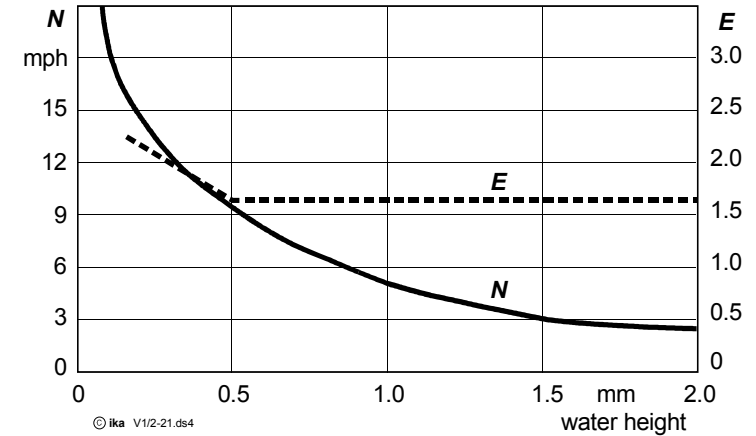
Fig. 2-21: Dependence of variables N and E on the water level

Fig. 2-22 compares the baffle resistance dependent on velocity and water level calculated by the above equation and the determined measuring results. At the same time it can be seen that the values received for bias and radial tires differ only slightly.

Fig. 2-23 shows different driving situations on a wet road. The first case involves a good tire tread pattern at a velocity of 80 km/h (50 mph). The approaching zone has a length of approx. 12 mm (0.47 inches), the dry contact zone approx. 68 mm (2.7 inches). At a speed of 150 km/h (93 mph), the approaching zone has already lengthened to 18 mm (0.71 inches). In the absence of a sufficient tire profile (case 3) at a speed of 80 km/h (50 mph), there is already a closed water layer between the tire and the road.

The resulting wheel resistance on wet road is added by the baffle resistance force and the rolling resistance on dry road:

$$F_R = F_{R \text{ roll}} + F_{R \text{ baffr}} \quad \text{Eq. 2-13}$$

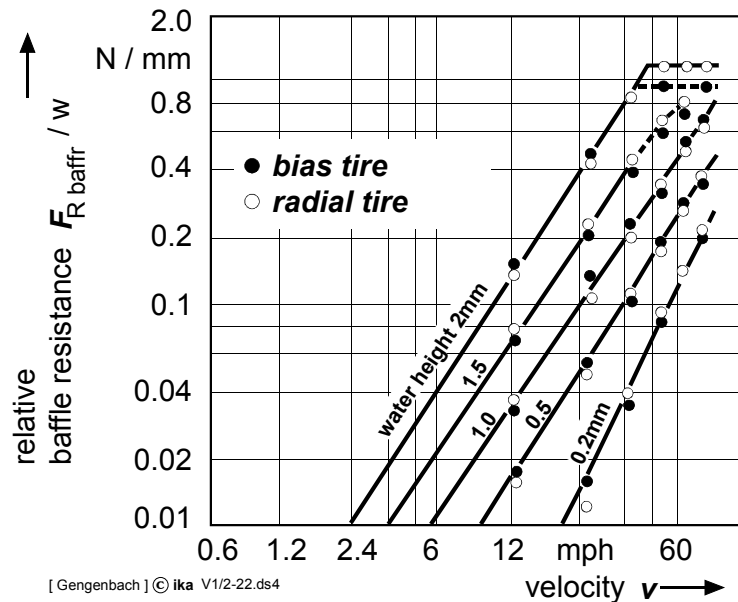


Fig. 2-22: Baffle resistance dependent on velocity for cross-ply and radial tires (Source: Gengenbach)

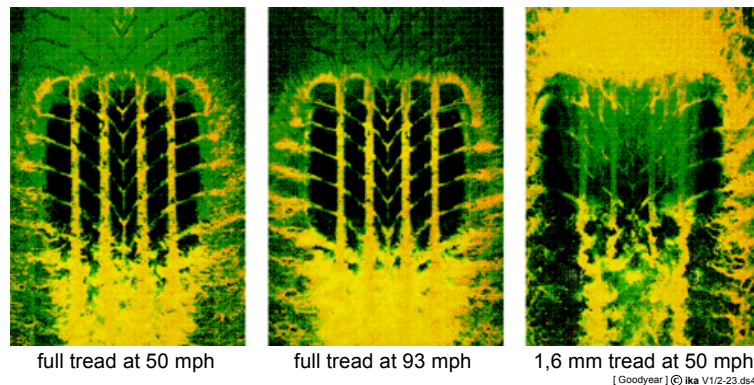


Fig. 2-23: Hydroplaning (Goodyear)

2.1.3 Component resulting from Slip

The previous considerations of wheel resistance assumed that the direction of rotation of the wheel was perpendicular to the axis of rotation and the axis of rotation parallel to the road. The

- Lateral force and
- Changed axle geometry

leads to a deviation from these conditions

a) Lateral Force

If a wheel rolls under the influence of a side force, mostly during cornering but also caused by side wind or inclined road surface, an angle results between the direction of motion of the wheel and the plane of the wheel. This angle is called the Slip angle (see lecture "AE II"). The vectoral representation of the forces acting on the wheel is shown in Fig. 2-24 as an example for the influence of the centrifugal force during cornering. The centrifugal force acts perpendicular to the direction of motion, while the rolling resistance towards the plane of the wheel. The lateral force on the other hand, acts normal to the wheel plane because of the slip angle α .

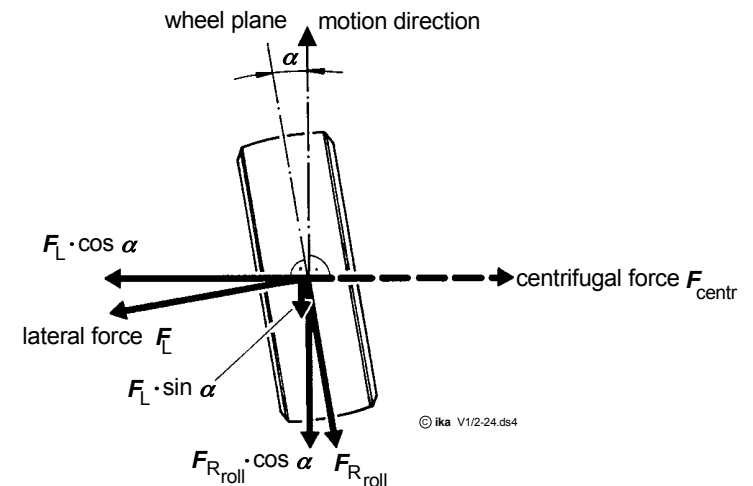


Fig. 2-24: Power relations on the wheel under slip resistance

The force of resistance counteracting the direction of motion results from the corresponding components of the lateral force F_S and the rolling resistance $F_{R\text{ roll}}$:

$$F_R = F_{R\text{ roll}} \cdot \cos \alpha + F_S \cdot \sin \alpha \quad \text{Eq. 2-14}$$

The right-hand side of the equation can be paraphrased in a way that the resistance share can be immediately seen from the slip angle:

$$F_R = F_{R\text{ roll}} - F_{R\text{ roll}} \cdot (1 - \cos \alpha) + F_S \cdot \sin \alpha \quad \text{Eq. 2-15}$$

Referring to the wheel load and as a result of the slip angle we obtain the wheel resistance coefficient:

$$f_{R\text{ slip}} = \frac{F_S \cdot \sin \alpha - F_{R\text{ roll}} \cdot (1 - \cos \alpha)}{F_{Z,W}} \quad \text{Eq. 2-16}$$

With the rolling resistance for uninterrupted straight-ahead drive it follows:

$$f_{R\text{ slip}} = \frac{F_S \cdot \sin \alpha}{F_{Z,W}} - f_{R\text{ roll}} \cdot (1 - \cos \alpha) \quad \text{Eq. 2-17}$$

Since the lateral force F_S in the case of negligible slip angles increases in proportion to the slip angle,

$$F_S = C \cdot \alpha \quad \text{Eq. 2-18}$$

in the case of small angles ($\cos \alpha \approx 1$), the wheel resistance coefficient can approximately be understood as a constant multiplied with the square of the slip angle.

$$f_{R\text{ slip}} = C^* \cdot \alpha^2 \quad \text{Eq. 2-19}$$

Fig. 2-25 shows this context for analyzed measurements. Compared with the numerical values in Table 2-1 it can be noticed that wheel resistances from slip at an angle of approx. 2° can already reach the same order of magnitude as rolling resistances in a straight-ahead drive.

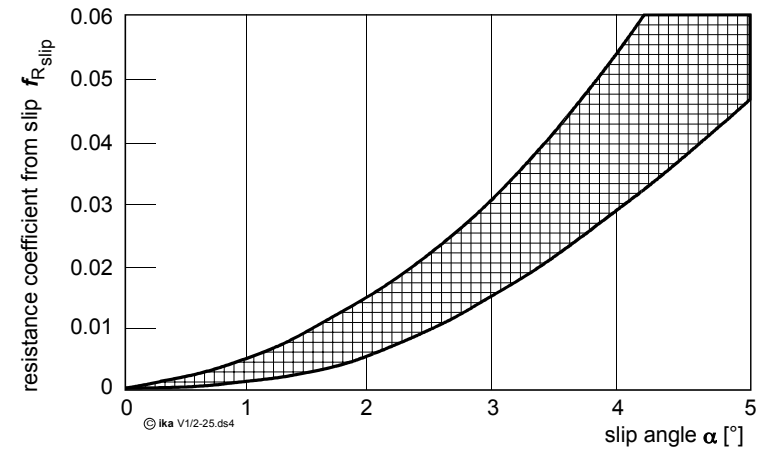


Fig. 2-25: Slip resistance coefficient depending on slip angle

b) Changed axle geometry

For the optimization of driving behavior (see lecture “AE-II”), wheel toe and/or camber angles are often set between the wheels and axles. This increases the wheel resistance since they produce the same effect as a wheel running under a slip angle.

- Wheel toe

The wheel toe angle δ is the angle between the vertical wheel planes of an axle and the vertical plane of the road, measured on the horizontal plane of the road. This induces a slip angle on every wheel even in the absence of an external lateral force on the vehicle. A component of the lateral force contrary to the driving direction results (Fig. 2-26).

Analogous to the slip angle, the wheel resistance of a wheel under toe is given by:

$$F_{R\text{ wheel toe}} = F_S \cdot \sin \frac{\delta}{2} = f_{R\text{ wheel toe}} \cdot F_{Z,W} \quad \text{Eq. 2-20}$$

For the wheel toe resistance of an axle this value has to be doubled.

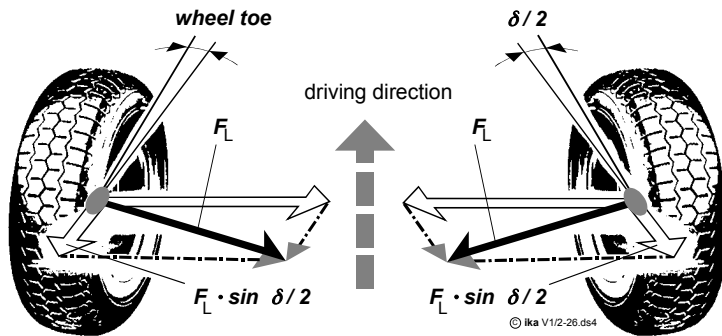


Fig. 2-26: Wheel toe resistance by wheel toe angle

- Wheel camber

The angle between the central vertical plane of the wheel and the vertical plane of the road measured in the transverse direction is called camber angle (Fig. 2-27).

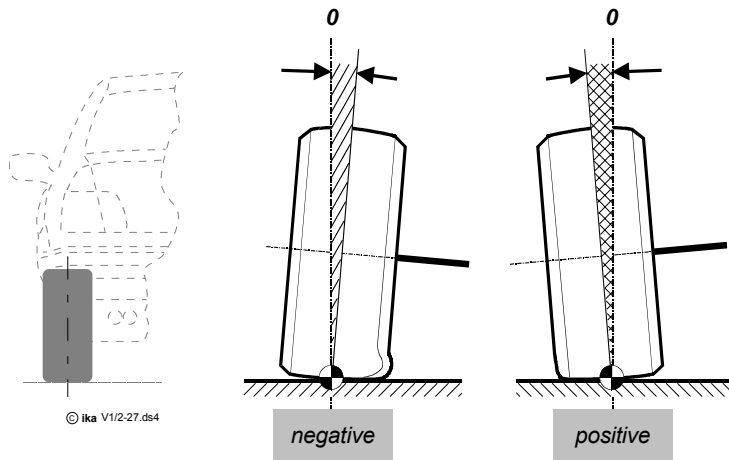


Fig. 2-27: Wheel camber angle

The wheel camber causes a changed rolling process in the direction of the tire vertical to the road. As a result, the running band is unsteadily loaded and in addition a sidewall deforms. Due to this the wheel resistance force slightly increases.

2.1.4 Summary of the components of Wheel Resistance

According to the preceding considerations, the total wheel resistance is calculated as the sum of the partial resistances:

$$F_R = F_{R \text{ roll}} + F_{R \text{ uneven}} + F_{R \text{ plast}} + F_{R \text{ baffr}} + F_{R \text{ slip}} + F_{R \text{ wheel toe}} + F_{R \text{ camber}} \quad \text{Eq. 2-21}$$

where $F_{R \text{ roll}} = F_{R \text{ flex}} + F_{R \text{ fric}} + F_{R \text{ fan}} \quad \text{Eq. 2-22}$

In the case of a straight-ahead drive on a dry road - basis for most road resistance calculations - the wheel resistance can be equated to the rolling resistance:

$$F_R \approx F_{R \text{ roll}} = f_{R \text{ roll}} \cdot F_{Z,W} \quad \text{Eq. 2-23}$$

2.2 Aerodynamic Drag

When a body is set in motion in a fluid atmosphere, a force of resistance, the aerodynamic drag, acts on it. Today, aerodynamics influence the body design in the development of nearly all new automobiles. Before presenting the components of aerodynamic drag, however, an overview of the basics of aerodynamics is dealt with.

2.2.1 Basics of Aerodynamics

Initially, a friction-free flow is considered. In the transverse plane, the potential flow around a cylinder is symmetrical in the stream area in the flow direction and perpendicular to it. This results in two ram points: one in the front and the other behind the cylinder where according to Bernoulli the entire kinetic energy is transformed into potential energy (ram pressure). According to the Potential Theory a loss-free and symmetrical return to the initial state before the body was subjected to a flow is achieved. In this case, as an integral value of the pressure distribution over the area, the aerodynamic forces having acted on a body sum up to zero. According to this, there is no flow separation or turbulence in the flow area (Fig. 2-28).

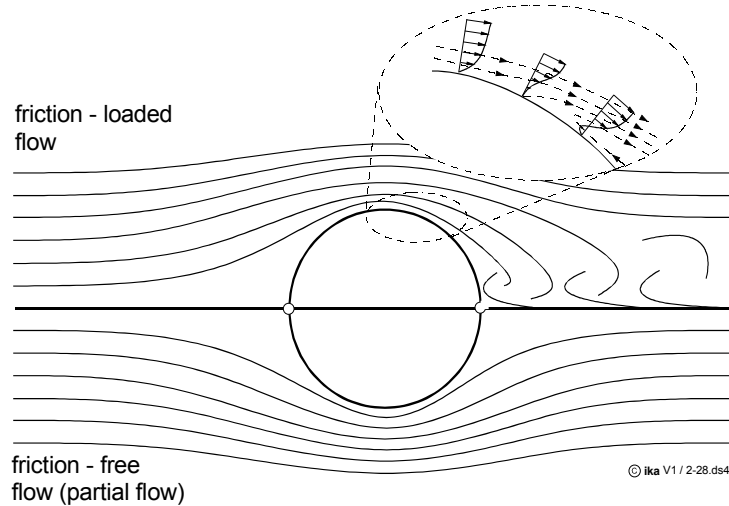


Fig. 2-28: Friction-free/ frictional cylinder flow (Source: Sorgatz, Fahrzeugaerodynamik)

The theoretically correct model mentioned in the idealization only does not correspond to the physical processes observed in reality. According to the Potential Theory, the characteristic curve of pressure for the flow model illustrates that the equilibrium of forces takes place at high magnitudes of forces in a direction transverse to the flow direction (Fig. 2-29).

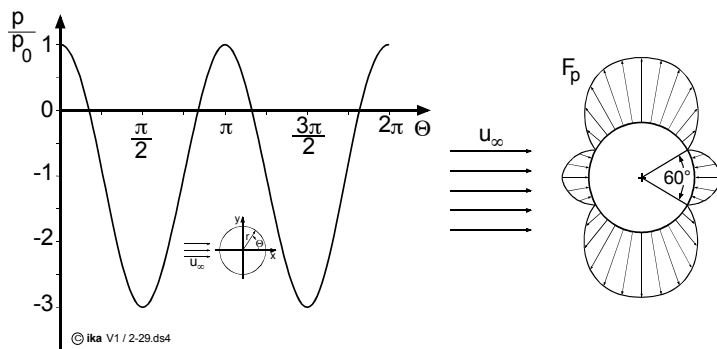


Fig. 2-29: Transversal flow of a cylinder (Source: Sorgatz, Fahrzeugaerodynamik)

In reality this streamline characteristic curve up to the largest cross-section corresponds very well to this model. In this case, the flowing medium above and below the ram point is initially accelerated. According to the Law of Conservation of Energy (Bernoulli's Equation) the local pressure decreases with an increasing flow velocity. For any point i along the streamline it follows:

$$\frac{\rho}{2} \cdot v_i^2 + p_i = \frac{\rho}{2} \cdot v_\infty^2 \quad \text{Eq. 2-24}$$

The amount of $\frac{\rho}{2} \cdot v_\infty^2$ is also called Ram Pressure.

In order to describe real flows, friction has to be considered. Due to its viscosity, the flowing medium adheres to the wall of the body in the region of direct contact. In the contact zone, the velocity of the medium also amounts to zero in the case of a stationary body. For a distance y between two surfaces that have a relative velocity v against each other, a linear velocity gradient results (Fig. 2-30).

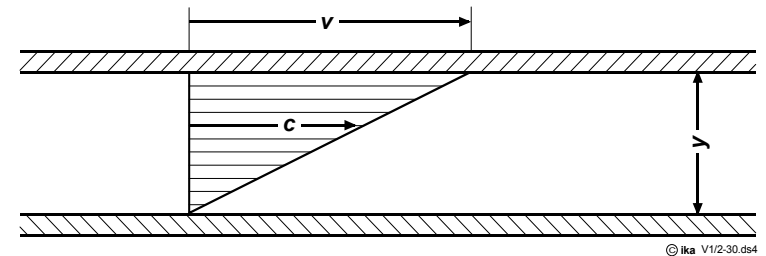


Fig. 2-30: Viscous draft between two moving plates

According to Newton's Law of Friction, the shear stress is given by:

$$\tau = -\eta_{\text{dyn}} \cdot \frac{dv}{dy} \quad \text{Eq. 2-25}$$

where τ = shear stress

η_{dyn} = dynamic viscosity

In case of free flow in the half space over a surface, the flow velocity at a distance asymptotically approaches the value of Bernoulli's Equation. This zone is called boundary layer. Its thickness is small in the field of accelerated flow but increases constantly in the case of constant or moreover in the case of decelerated flow. Due to

the asymptotic approach towards the Bernoulli-state, as limit value a 99% approach can be defined e.g. for settling the boundary layer thickness. Outside the boundary layer, the velocity of flow once again amounts to v_∞ . In Fig. 2-31 the boundary layer flow over a plate in the longitudinal direction is presented.

Initially, the flow is stationary towards the parallel layers of the plate in the form of laminar flow. The boundary layer thickness δ increases according to the following relation:

$$\delta \approx \sqrt{\nu / v_\infty} \cdot \sqrt{x} \quad \text{Eq. 2-26}$$

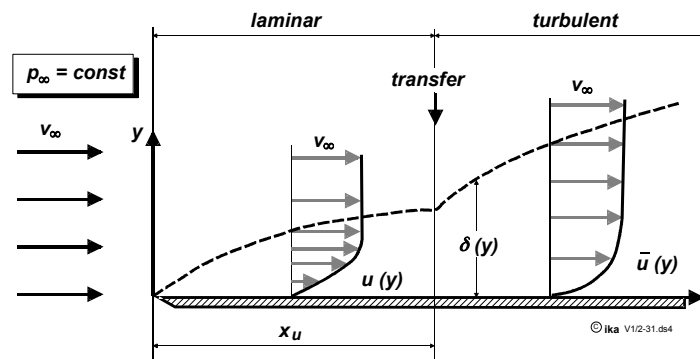


Fig. 2-31: Boundary layer formation at a lengthwise flown plate

The laminar boundary layer flow remains stable up to a certain length x_u . The boundary layer then becomes turbulent and its mean velocity profile significantly increases by stronger impetus exchange in the sidewall area. The position of the point of transition is defined by the Reynold's number formed with the characteristic length. The Reynold's number is defined as:

$$\text{Re} = \frac{v_\infty \cdot l}{\nu} \quad \text{Eq. 2-27}$$

where: v_∞ = flow velocity

ν = kinematic viscosity¹

The kinematic viscosity e.g. amounts to 0.15 cm²/s for air, 0.01 cm²/s for water, and 3.8 cm²/s for machine oil each at a temperature of 20°C (68°F). The Reynold's number is very important for aerodynamic pilot experiments. Two flows are similar if their Reynold's numbers correspond. This means that the relations of pressure forces, frictional forces, and inertia forces correspond at identical points (Fig. 2-32).

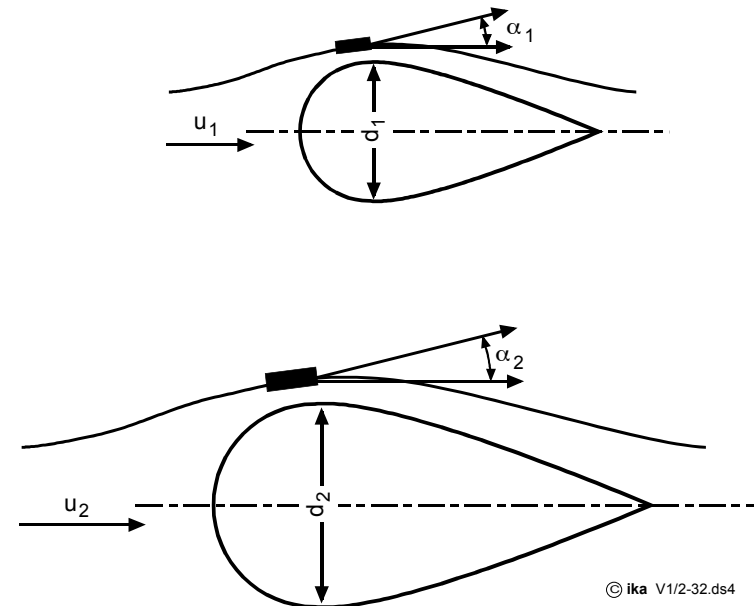


Fig. 2-32: Flow on two geometrically similar bodies

With the dimensionless Reynold's number, flow relations can be generally characterized. The transfer at a plate with longitudinal flow with pressure remaining constant over the characteristic length takes place, for example, at $\text{Re}=5 \cdot 10^5$ (Fig. 2-31). Increasing pressure and higher wall roughness shorten the area of laminar flow while decreasing pressure shifts the transfer point towards larger characteristic

¹ The kinematic viscosity is related to the medium's density ρ with the dynamic viscosity

$$\eta: \quad \nu = \eta / \rho$$

lengths. At a vehicle's surface, the length of the laminar boundary layer amounts to approx. 20-30 cm (8-12 inches).

The turbulent boundary layer the flow is always instationary. The flow velocities differs in magnitude as well as direction. Only a velocity U_{mean} in time exclusively runs parallel to the wall. Besides an entire velocity gradient, the additional turbulent interactions produce higher shear stresses and an quicker increasing boundary layer thickness over the characteristic length.

Thus:

$$\delta \approx \sqrt[5]{\nu / v_{\infty}} \cdot \sqrt[5]{x^4} \quad \text{Eq. 2-28}$$

The increasing pressure in flow direction decelerates the flow, most of all right next to the wall. This can lead to the formation of reverse currents of fluid particles close to the wall. According to Fig. 2-33 the preceding particles are displaced away from the wall by the particles that flow in the reverse direction.

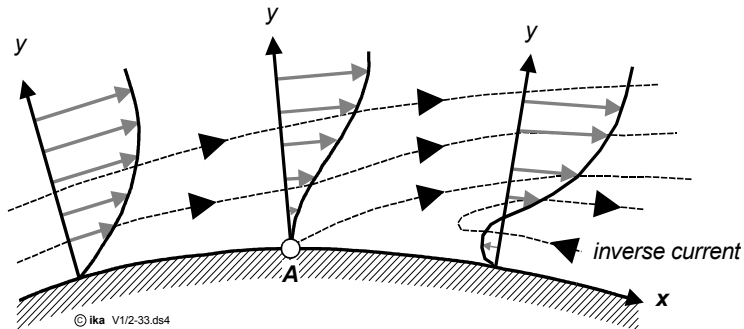


Fig. 2-33: Separation of boundary layer flow on a wall

The swirls in the boundary layer lead to turbulences whose origin and formation can be caused by incidental disturbances on the surface of the body or by impetus within the flow itself. This can lead to random separation, local inverse flows up to the formation of regular swirls, or cyclical vibrations of the swirl. The four phase pictures of a transversally streamed rectangular cylinder at low Reynold's numbers illustrate the relationships between the boundary layer and the dead water area formed in the train with the caster swirl. This caster swirl cyclically changes its running direction (Fig. 3-34). E.g. for $Re > 1,000$, real swirls are formed for cycle cylinders.

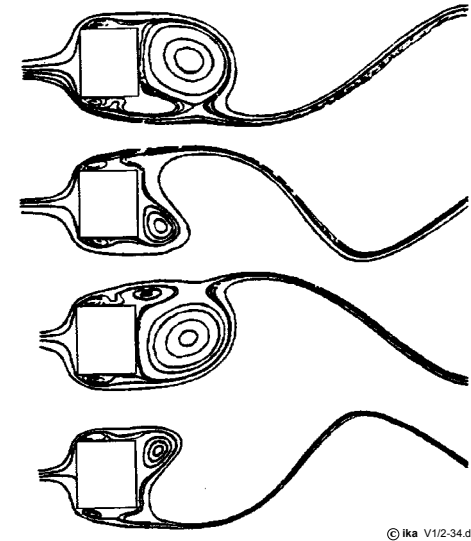


Fig. 2-34: Transversal flow of a rectangular cylinder

For $Re > 100,000$ up to $500,000$, the boundary layer at the cylinder becomes totally turbulent. The flow being closer to the circumference, the caster ties up into a tight cross-section and the dimensionless coefficient of aerodynamic drag c_w decreases (Fig. 3-35). In an automobile, the Reynold's number spans the entire area between $Re=0$ up to more than $10,000,000$.

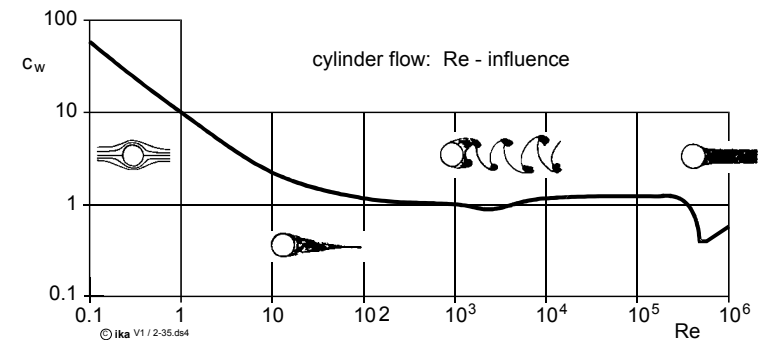


Fig. 2-35: Influence of Reynold's number in cylindrical flow (Source: Sorgatz, Fahrzeugaerodynamik)

If flow separation occurs in real body flow with friction, the pressure distribution in the emerging separated region ("dead water") on the leeward side of the flow considerably differs from the ideal one (Fig. 2-29). A vacuum is created in this area which induces a resistive force in the flow direction towards the body. This resistance is called the pressure resistance (e.g. on the rear end of a vehicle).

The previously mentioned shear stresses in the flowing media also cause frictional forces along the surface. The individual components of the frictional forces when summed up in the flowing direction results in the frictional resistance. In the absence of flow separation, the aerodynamic drag of the body is essentially results through friction. Apart from that, the frictional resistance has a significant effect only on long vehicles (e.g. railroad trains, aerodynamically optimized trucks with trailers).

In practice, these resistances are summed up by introducing the dimensionless coefficient c_W . The aerodynamic drag of a vehicle is described by:

$$F_{\text{drag}} = c_W \cdot A \cdot \frac{\rho_L}{2} \cdot v_{\infty}^2 \quad \text{Eq. 2-29}$$

where A =cross-sectional area exposed to flow²

ρ_L =density of fluid

The flow velocity v_{∞} is transferred to the moving vehicle, and is made up of the vectoral sum of the wind velocity v_W and the vehicle velocity V_{veh} which acts against the driving direction.

The influence of varying air density on the aerodynamic drag must not be forgotten. According to the ideal gas equation, air density is given by:

$$\rho_L = \frac{p}{R_A \cdot T} \quad \text{Eq. 2-30}$$

where: p =air pressure

R_A =gas constant of air (approx. 287 KJ/Kg K)

T =temperature

² projected frontal area of the vehicle

Realistic fluctuations of air density between 1.15 kg/m³ and 1.32 kg/m³ are noticed in habitable climatic zones with temperatures between -15°C (5°F) and +35°C (95°F) and air pressures between 980 and 1,020 mbar.

Since the main dimensions of a vehicle are to a large regulated by law, a reduction of the aerodynamic drag is possible only by means of a reduced c_W -value. Drag coefficients of $c_W=0.15$ achieved in prototypes, indicate the high development potential for the future compared to today's mass-produced passenger cars ($c_W=0.26-0.45$) and commercial vehicles ($c_W= 0.6-0.8$).

In order to achieve an aerodynamic body design, it is still necessary to undertake experimental tests in a wind tunnel despite the availability of computer-aided computational methods.

Based on the principle of guidance of air, one can distinguish between two designs of wind tunnels, the *Eiffel*- and the *Göttinger* design (Fig. 2-36, 2-37). A wind tunnel in the *Eiffel* design (Fig. 2-36) is open to the intake as well as to exit side, while the *Göttinger* design (Fig. 2-37) consists of a closed air circuit.

A wind tunnel with a closed design operates independent of the weather and is thus suitable for being converted into a climate tunnel. Heating up the air at a closed return requires a cooling device to protect the models mainly consisting of non-heat-resistant materials. The resulting pressure loss and the necessary cooling power reduces the advantage of the low propulsive power due to the design.

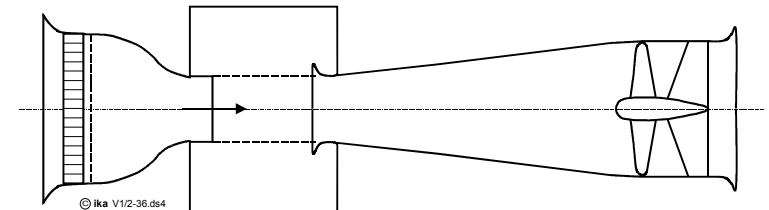


Fig. 2-36: Wind tunnel in open design (*Eiffel* design)

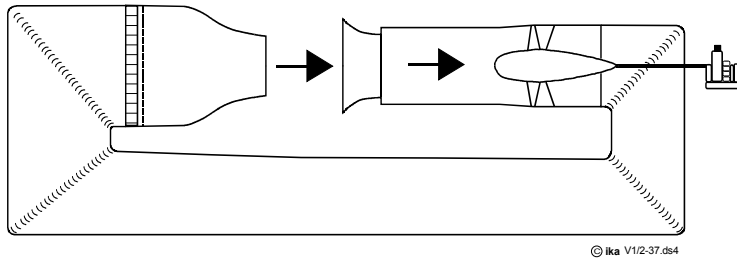


Fig. 2-37: Wind tunnel in closed design (*Göttinger* design)

The real advantage of the *Eiffel* tunnel consists in its cost-efficient design without a feedback channel and cooling. The influence of wind and ambient temperature upon the flow conditions within the measurement track only allows a limited use in Europe. Further disadvantages are the high noise pollution and contamination of the intake air. An additional installation of filter systems requires an increased blower output.

Measuring track, nozzle, and blower are decisive construction elements for a wind tunnel's performance.

The size of the measuring track, characterized by the relation of the frontal area of vehicle to the cross-sectional area of the stream (locking ratio), largely determines the dimensions of the entire wind tunnel. A locking ratio of 0.07 should not be exceeded in order to get as close as possible to free flow. The investment required for corresponding cross sections of the stream is normally too high for the automotive industry. By adjusting the measuring track to the flow conditions at the vehicle an attempt is made to minimize the size of the wind tunnel. Based on the type of wall, four kinds of measuring tracks may be classified (Fig. 2-38).

A closed measuring track offers the advantage of a large utility length. Pressure decrease as a result of frictional resistance on the walls can be compensated by a minor sectional reduction in flow direction. The simple design of the rim, however, requires larger measurement corrections. This led to the development of the streamlined wall-design in which the line guide is adjusted to the walls of the flow pattern of vehicles which varies only slightly for passenger cars.

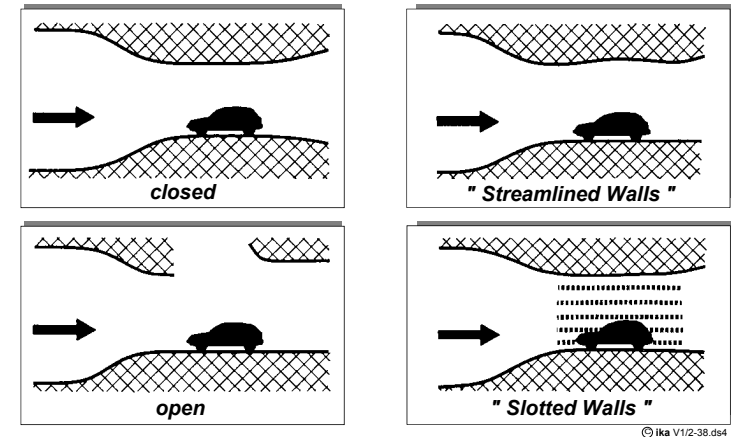


Fig. 2-38: Distinction between measuring tracks based on the kind of wall

The real advantage of the open measuring track is that the pressure roughly remains constant in flow direction. A low measurement falsification by locking and a good accessibility are further advantages. Disadvantages are the small utility length, higher losses, unhindered sound radiation, necessary air conditioning of the hall surrounding the measuring track when being used as a climate tunnel, and large catch trumpets required as a consequence of an extended stream. The measuring track with slotted walls comes pretty close to the demanded unhindered flow with small main dimensions. The layout of slots however turns out to be a problem.

The compression area ratio of the nozzle contour has an important influence on the regularity of flow. A large compression area ratio (quotient of the nozzle inlet and discharge cross section) signifies a uniform velocity distribution in the discharge at a low blower output. However, it requires a large constructional expense for a correspondingly large stream. The contour of the stream has to be designed in such a way that the flow within the stream does not separate and that no superfluous turbulences arise in the discharge.

In the past the regulation of flow velocity was largely carried out by rotor blade adjustments of the blower. Instantaneous changes in velocity and simple blower drives can be achieved this way. Serious disadvantages include the susceptibility to interference, difficult rotor implementation with correspondingly large supports, and high noise pollution in all operating conditions.

In the case of newer wind tunnels, electronic speed adjustment has gained acceptance as a method of control. The blower rotor can be designed lightweight.

Corresponding to the degrees of freedom in three-dimensional space, three forces (force of aerodynamic drag, lateral force, and lift) and three moments (pitching, rolling, and yawing moment) (Fig. 2-39) act on the vehicle. They are usually determined by measuring the three force coordinates on each wheel of the vehicle. By geometrically adding up the forces recorded in the wind tunnel, the aerodynamic forces and moments can be determined. This will not be considered here in more detail. Out of the measured forces, the aerodynamic coefficients, e.g. c_W -value, are computed by combining them with the remaining constants (Eq. 2-29). In this case the vehicle flow should not be changed and a nearly distance and friction-free measurement should be made possible. The latter can be achieved by using measurement boxes (DMS-pickups, pressure measuring boxes) and hydrostatic or pneumatic bearings.

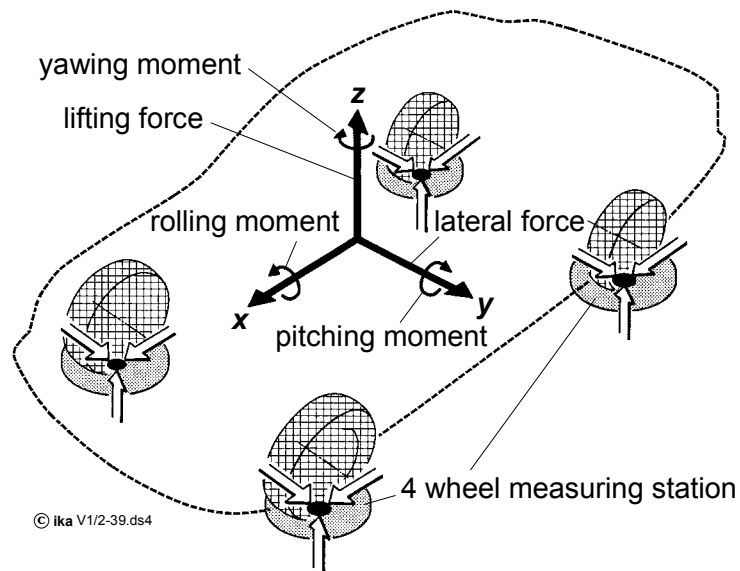


Fig. 2-39: Forces and moments in aerodynamic measurements

Modern wind tunnels are used not only determine the aerodynamic drag and their coefficients, but also for the acoustical testing of vehicles. The increasing demand for noise comfort in passenger cars, in this respect requires optimizations and tests. The

challenge involved in these tests lies in lowering the ambient noise level in so-called aeroacoustic-wind tunnels, to such a level that the actual noise measurement on the vehicle cannot be falsified very much by the tunnel's own service noise. Operating sounds have been lowered down to 60dB. Fig. 2-40 shows the wind tunnel at BMW AG Munich.

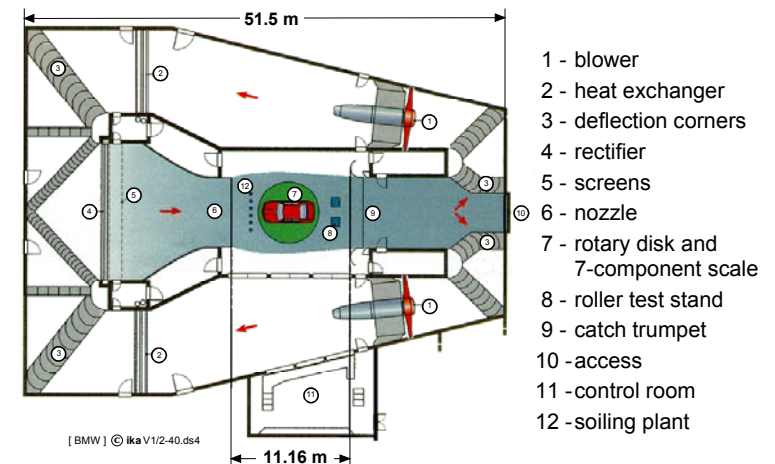


Fig. 2-40: Aerodynamic wind tunnel at BMW AG

2.2.2 Components of Aerodynamic Drag

Besides the already mentioned theoretical forms of resistance

I. Pressure resistance and

II. Frictional resistance

in the vehicle, there are also the

III. Induced resistance and the

IV. Internal resistance

Which constitute the overall resistance.

I. Pressure resistance and Induced resistance

Apart from smaller regions of flow separation, the size of the separation zone on the rear-end determines the pressure resistance to a large extent. As illustrated in

Fig. 2-41, for an equal vehicle frontal area A , differing flow directions of the distinct bodies result due to widely differing rear-end separation cross sections A_a .

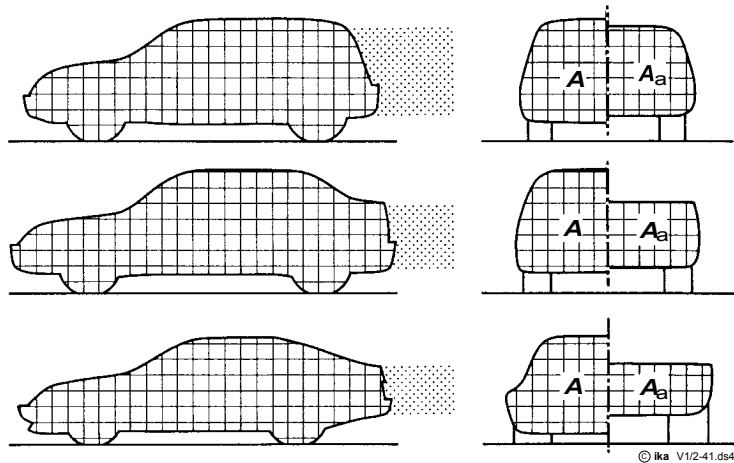


Fig. 2-41: Air separation cross sections of different bodies

Small separation zones and hence small vacuum zones on the vehicle are to be targeted. By specifically influencing the turbulence on the rear-end separation, a smaller vacuum and hence a smaller pressure resistance can be achieved. An “aspiration” of the boundary layer in the rear-end area can also bring in significant reduction in resistance.

The induced resistance is a part of the vehicle's pressure resistance. Air pressure differences between the vehicle's upper and lower side produce cross-flows (Fig. 2-42) that form two large longitudinal swirls together with the roof flow (Fig. 2-43).

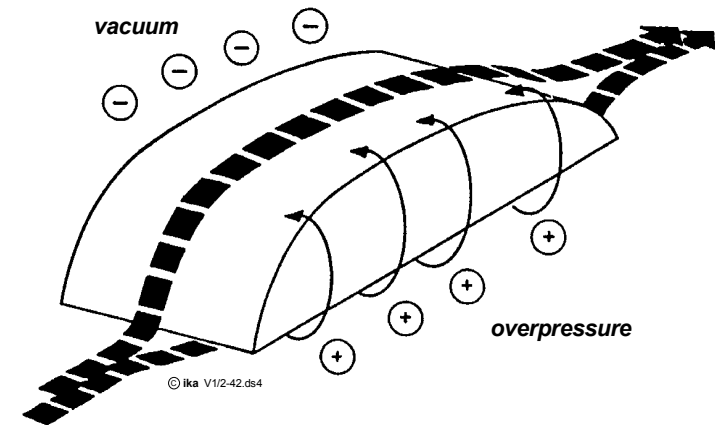


Fig. 2-42: Induced air flow at the vehicle body

In their immediate vicinity such swirls induce low pressures. The “dead water area” at the rear end is extended and thus leads to an increased pressure resistance.

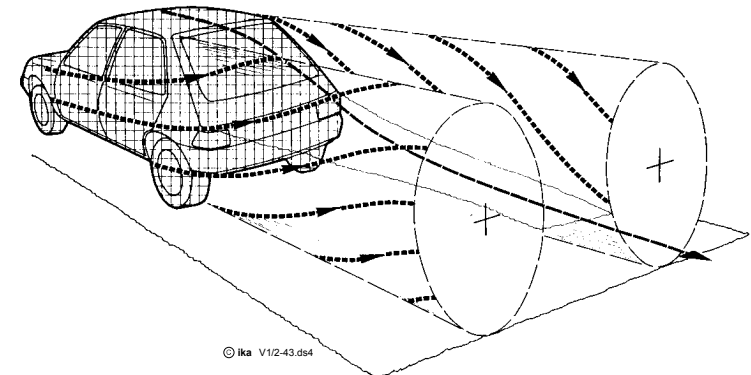


Fig. 2-43: Longitudinal swirls caused by induced draft

– Frictional resistance

The vehicle's surface resistance, in theory called frictional resistance, is of significance in long vehicles, e.g. busses. Fig. 2-44 shows the summing up of the resistance of an aerodynamically-efficient bus body with a negligible resistance at the vehicle front, relatively high resistance at the rear end, and body resistance

consisting basically of frictional resistance which steadily increases with the length of vehicle.

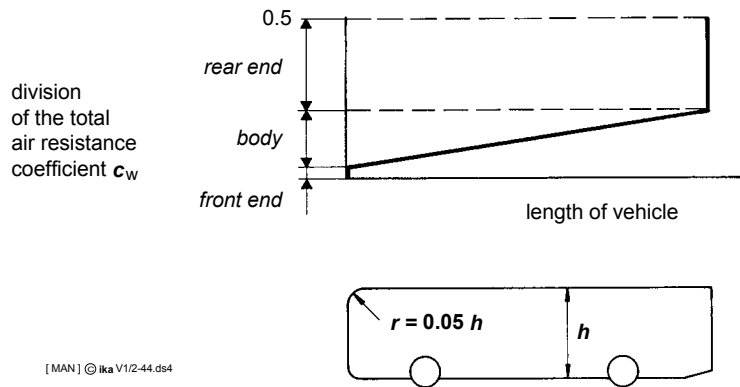


Fig. 2-44: Components of aerodynamic drag in an optimized bus body (MAN)

– Internal resistance

Air not only flows around a vehicle, but also through the vehicle in order to cool down the assembly and to ventilate the passenger compartment. When air flows through the cooler, engine compartment, wheelhouses, and passenger compartment (Fig. 2-45), losses of momentum arise from friction as well as turbulences and flow separation in the vehicle's interior. The resulting internal resistance which constitutes about 3-11% of the total aerodynamic drag only makes up a small part.

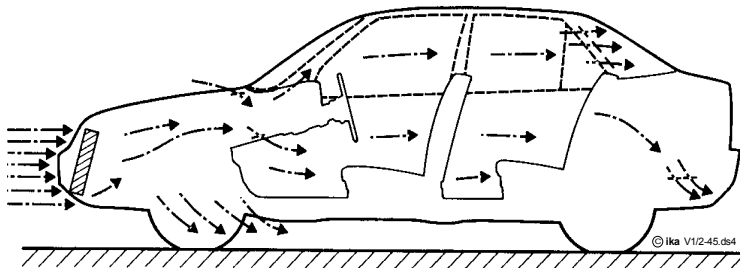


Fig. 2-45: Air flow through a motor vehicle

– Oblique air flow

When air flow attacks a vehicle at an angle, the drag coefficient significantly changes. Fig. 2-46 shows the influence of the angle of approach on the c_w -value for different passenger car designs.

Fig. 2-47 shows the corresponding dependence for a commercial vehicle with a 3.6 m (11.8 feet) high tarpaulin body ("Hamburg tarpaulin").

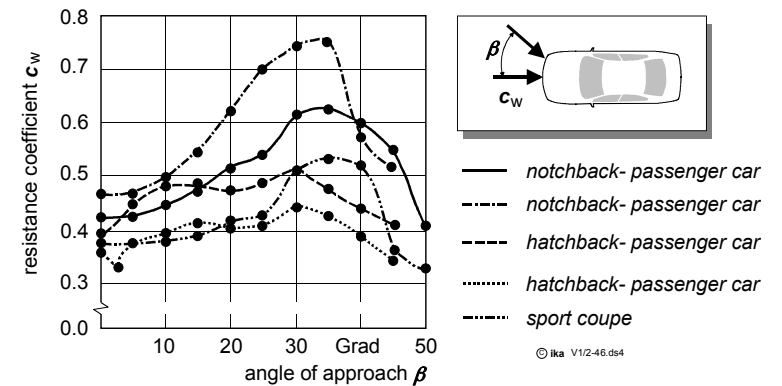


Fig. 2-46: Aerodynamic drag at oblique flow for different passenger cars

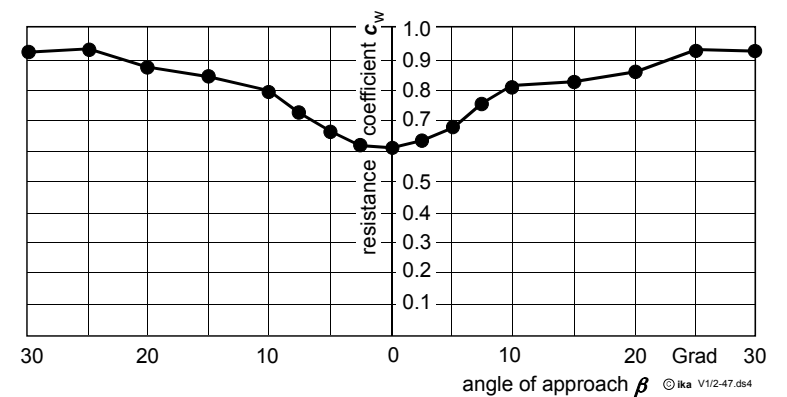


Fig. 2-47: Aerodynamic drag for a commercial vehicle with tarpaulin body approached at an angle

Consequently, for passenger cars as well as for commercial vehicles, there are considerable increases in the co-efficients of resistance when they are approached by a side wind. In this case the maxima of the angle of approach of $\beta=25^\circ$ - 35° can be achieved. When the vehicle is driven, angles of approach greater than 15° are however exceptional.

Subsequently the c_W -values of a few production vehicles are presented in Table 2-3. Fig. 2-48 shows the development of the c_W -values in recent years.

Vehicle	c_W -Value	Frontal Area	$c_W \cdot A$
Citroen 2CV	0.51	1.65	0.85
VW Beetle	0.48	1.80	0.87
VW Polo	0.32	1.90	0.61
VW Golf III	0.30	1.99	0.60
Ford Mondeo	0.32	2.00	0.64
BMW 5series (E39)	0.27	2.17	0.59
MB S-class	0.31	2.38	0.74
Opel Calibra	0.26	1.90	0.49

Table 2-2: c_W -values of different vehicle designs

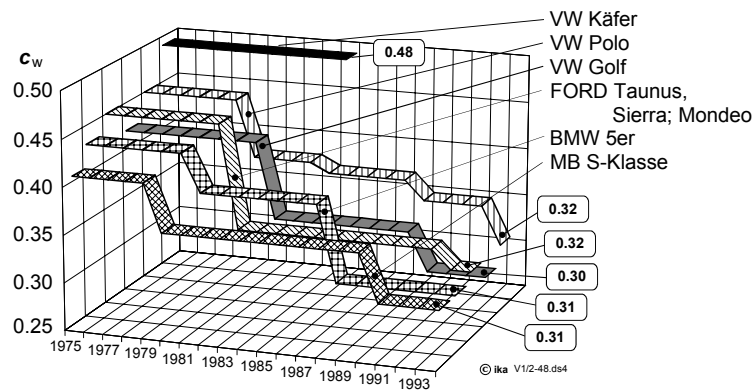


Fig. 2-48: Historical development of c_W -values

2.3 Resistance due to Gradients

A gradient p is defined as the quotient of vertical and horizontal road projection, which corresponds to the tangent of the angle of gradient α_{gr} . Traffic signs usually indicate the angle of gradient in percent.

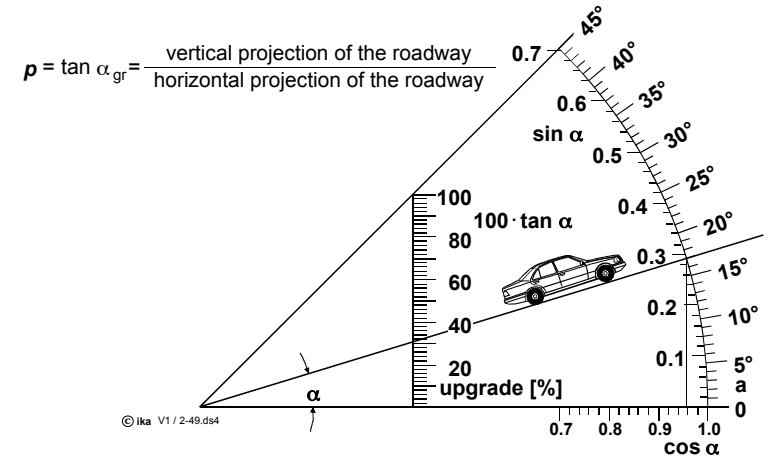


Fig. 2-49: Definition of gradients

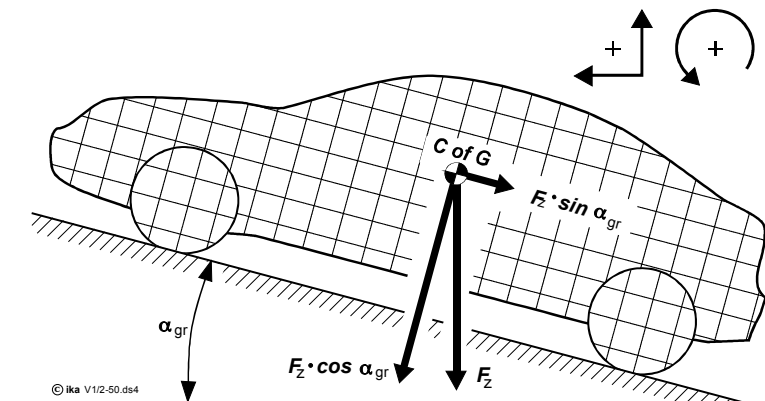


Fig. 2-50: Forces acting on a vehicle in a gradient

In a gradient, the weight of the vehicle shifts at the center of gravity according to Fig. 2-50. On a gradient, the normal force reduces to:

$$F_z = G \cdot \cos \alpha_{gr} \quad \text{Eq. 2-31}$$

while in addition a component parallel to the road, the negative force acting downhill of magnitude

$$G \cdot \sin \alpha_{gr} \quad \text{Eq. 2-32}$$

Thus for a vehicle on a gradient, the force that has to be overcome quantifies as upgrade resistance F_{gr}

$$F_{gr} = G \cdot \sin \alpha_{gr} \quad \text{Eq. 2-33}$$

where: $\alpha_{gr} = \arctan p$

Table 2-4 shows the maximum allowable gradients in Germany.

Road type	Design speed in km/h	Permissible gradients in %
1. Rural Roads		
District roads	40	10,0
Country roads	60	6,5
Inter-state highways	80	5,0
	100	4,5
German Federal Autobahns	100	4,5
	120	4,0
	140	4,0
2. Urban Roads		
Multi-lane main roads		5 - 6
Service roads		10
Residential roads		10
3. Alpine roads		
		30

Table 2-3: Permissible upgrades (Source: RAS-L-1)

2.4 Acceleration Resistance

In addition to the forces of resistance resulting from wheel, aerodynamic, and possibly climbing resistance, which arise in a stationary driving condition, in transient driving conditions ($v \neq \text{const}$), inertia forces arise as a result of occurring accelerations and decelerations. These inertia forces have to be overcome when the drivetrain of the vehicle is accelerated. The resistances are composed of a:

Translational component

which is produced by the transient motion of the vehicle mass and a

Rotational component

which results from the acceleration or deceleration of the vehicle rotating drivetrain components.

2.4.1 Translational component

The force resulting from a translational acceleration can be determined by the law of inertia forces according to d'Alembert.

$$F_T = -m \cdot a_x \quad \text{Eq. 2-34}$$

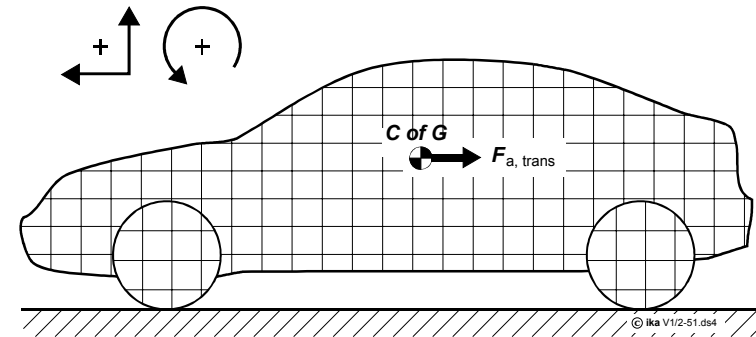


Fig. 2-51: Translational inertia force acting on the vehicle

As a result of the inertia force a translational resistance force $F_{a, trans}$ results as follows:

$$F_{a, trans} = -F_T = m \cdot a_x \quad \text{Eq. 2-35}$$

where: $m = m_V + m_{cap}$ Eq. 2-36

m_V =mass of the vehicle

m_{cap} =capacity weight

a_X =acceleration of vehicle

2.4.2 Rotational component

Since the vehicle's rotating parts are accelerated rotationally during its translational acceleration, in addition a rotational resistance force has to be overcome. In order to determine this force, the mass inertia moments of the rotating parts have to be reduced to the driven axle. Analogous to the calculation of the translational acceleration, we obtain:

$$M_T = -\Theta_{red} \cdot \ddot{\varphi} \quad \text{Eq. 2-37}$$

The rotational resistance force is given by:

$$F_{a,rot} = \frac{-M_T}{r_{dyn}} = \frac{+\Theta_{red} \cdot \ddot{\varphi}}{r_{dyn}} \quad \text{Eq. 2-38}$$

where: Θ_{red} =inertia moment of all rotating parts reduced to the drive shaft

φ =angular acceleration of the driven wheel

r_{dyn} =dynamic wheel radius

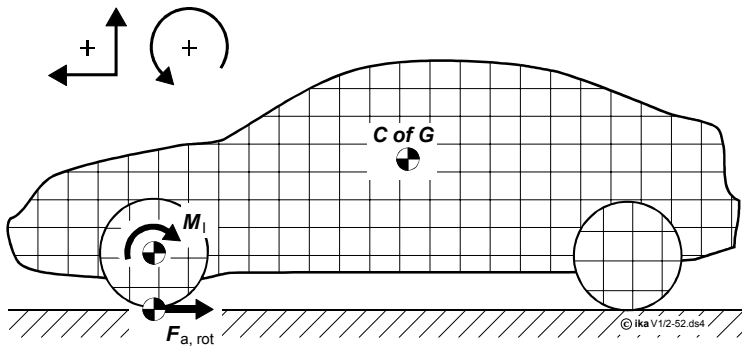


Fig. 2-52: Rotational inertia force acting on the drive wheel

From the relation

$$\varphi = \frac{x}{r_{dyn}} \quad \text{Eq. 2-39}$$

by differentiating it twice with respect to time

$$\ddot{\varphi} = \frac{\ddot{x}}{r_{dyn}} \quad \text{Eq. 2-40}$$

Applying $\ddot{x} = a_x$ it follows:

$$F_{a,rot} = \frac{\Theta_{red}}{r_{dyn}^2} \cdot a_x \quad \text{Eq. 2-41}$$

For the mass moment of inertia Θ_{red} , the following moments of inertia of the drive train have to be considered (Fig. 2-53):

motor, clutch : $\Theta_{eng} + \Theta_C$

transmission with particular multiplication i : Θ_{Ti}
(referred to the transmission input shaft)

drive shaft, differential : Θ_{dr}

wheels (mostly including brake drum or brake disk and axle shafts) : Θ_W

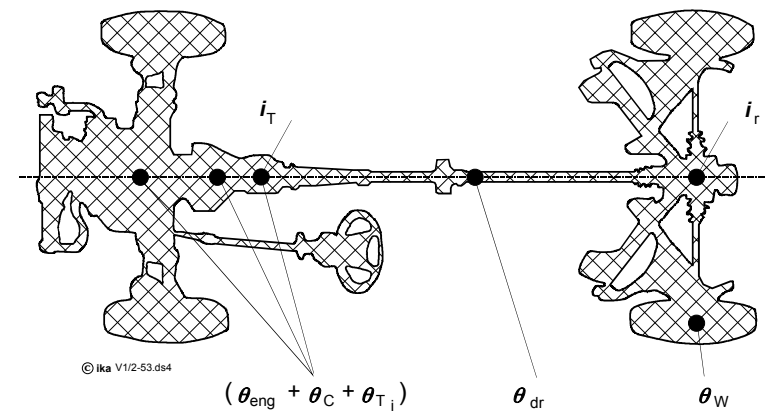


Fig. 2-53: Rotating masses that have to be considered at changing velocity

In the consideration of the mass moment of inertia of the wheels, attention has to be paid to the fact that all wheels of the vehicle have to be considered, regardless whether the front wheels, rear wheels, or all wheels are driven.

Considering the ratios in the transmission i_{Ti} (for the particular gear) and the axle transmission $i_{r(f)}$ (for rear- and front-wheel drive) we arrive at the mass moment of inertia reduced to the drive axle for a gear i . The dynamic equality of output and replacement system is to be maintained.

$$\Theta_{red\ i} = \Theta_R + i_{r(f)}^2 \cdot \Theta_{dr} + i_{r(f)}^2 \cdot i_{Ti}^2 \cdot (\Theta_{eng} + \Theta_C + \Theta_{Ti}) \quad \text{Eq. 2-42}$$

2.4.3 Summary of Acceleration components

The total acceleration resistance is given by the sum of the translational and rotational resistance forces as follows:

$$F_a = F_{a\ rot} + F_{a\ trans} \quad \text{Eq. 2-43}$$

$$F_a = \left(\frac{\Theta_{red\ i}}{r_{dyn}^2} + m_V + m_{cap} \right) \cdot a_x \quad \text{Eq. 2-44}$$

Introducing a mass factor:

$$e_i = \frac{\Theta_{red\ i}}{m_W \cdot r_{dyn}^2} + 1 \quad \text{Eq. 2-45}$$

introducing vehicle specific data, the total inertial resistance is given by:

$$F_a = (e_i \cdot m_W + m_{cap}) \cdot a_x \quad \text{Eq. 2-46}$$

Since the square of the transmission ratio is involved in the determination of the reduced mass moment of inertia, the mass factor can be distributed over a large area. For example in the case of off-road vehicles or commercial vehicles with an extremely high crawling gear ratio (crawler), a higher force is required to accelerate the rotating masses than for pure translational acceleration of the vehicle ($e_i > 2$).

Subsequently Table 2-5 lists mass factors for different vehicles in different gears.

Gear:	1	2	3	4	5
Ford Escort	1.32	1.15	1.10	1.07	1.06
BMW 730i A	1.21	1.10	1.05	1.03	
DB - bus	1.61	1.18	1.08	1.06	1.03
Gear:	crawler	2	4	6	8
DB 1632 K	10.12	3.18	1.65	1.21	1.08

Table 2-4: Mass factors for different vehicles in different gears

2.5 Overall Resistance

The individual road resistances deduced in the previous chapters can be clearly summed up as overall resistance. The resistances have to be overcome by propulsion power between the wheel and the road. They can also be presented in the form of driving torques acting on the wheels divided by the wheel radii:

$$F_{dem} = \frac{M_f + M_r}{r_{dyn}} = (e_i \cdot m_W + m_{cap}) \cdot a_x + F_Z \cdot (p + f_R) + c_w \cdot A \cdot \frac{\rho_L}{2} \cdot v^2 \quad \text{Eq. 2-47}$$

By entering the component resistances in a joint diagram, the following characteristic results:

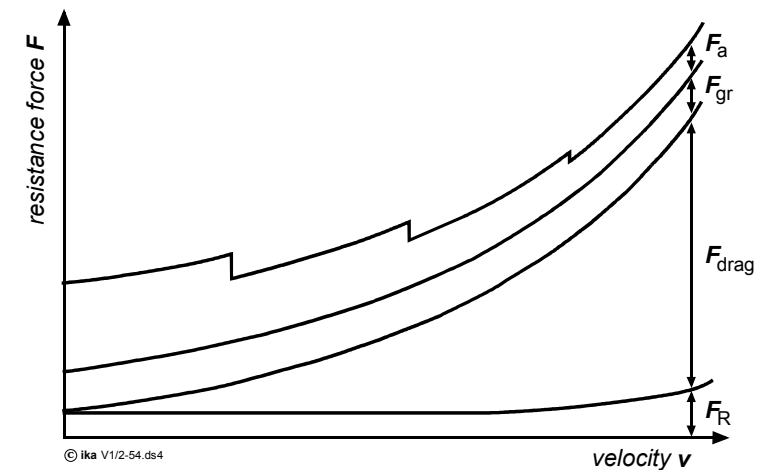


Fig. 2-54: Road resistances in a gradient

The influence of a particular component of resistance depends on the vehicle and the driving condition. It can generally be stated, that in passenger cars the aerodynamic drag has a dominant influence at high speeds. In commercial vehicles, however, this influence reduces (Fig. 2-55). It has to be taken into account that apart from the aerodynamic drag all road resistances proportionally depend on the vehicle weight³.

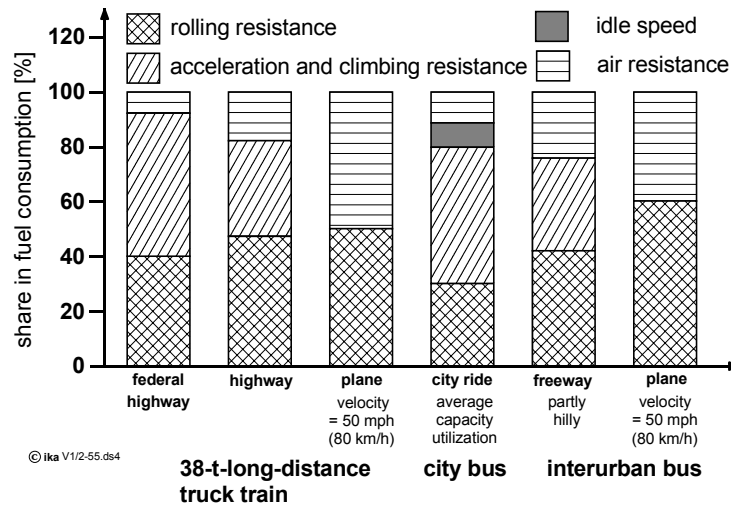


Fig. 2-55: Exemplary road resistance shares in commercial vehicles (Source: Lastauto & Omnibus 2/1996)

The road resistance curves are also called demand characteristics, since they reflect the driving torque that a vehicle requires for different driving conditions. If the power demand is required, in each case the curves shown in Fig. 2-54 have to be multiplied with the driving speed. The resulting characteristic diagram is shown in Fig. 2-56.

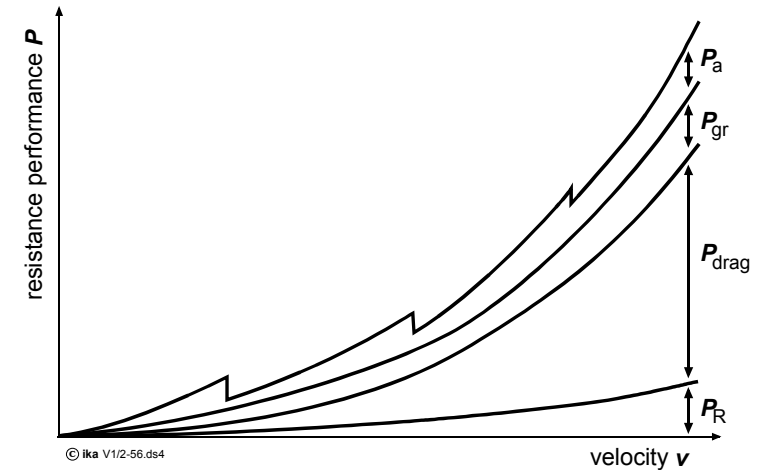


Fig. 2-56: Road resistance performances depending on speed

The “supply” from the vehicle powertrain has to correspond to these groups of demand curves in order to achieve the longitudinal dynamics corresponding to the required driving conditions.

³ a slight deviation results from the influence of the mass factors

3 Powertrain

Following the road resistances and the resulting power and energy demands of vehicles, this chapter deals with the components that are used to overcome the described resistances and make it possible to maintain a vehicle at the desired speed. In the last section, the interaction of the single components as a vibrating system is dealt with.

Based on their functions, the components of the powertrain may be classified into:

- Propulsion units (engines)
- Speed converters (clutches)
- Torque converters (transmissions)
- Transfer case (differential)
- Brakes

3.1 Energy Accumulators

Before considering the components in further detail, the fundamentals will be dealt with at first. It is necessary to contribute energy to the powertrain so that a vehicle may be set in motion. Only in a few cases, the vehicle receives this energy externally, e.g. electro trolley busses. Normally the vehicle requires its own energy source. Based on the propulsion concept it can be:

- Chemical energy,
- Electrical energy,
- Mechanical energy,
- Hydraulic energy,
- Thermal energy.

Table 3-1 compares the energy and performance densities of different energy sources. In case of fossil fuels it has to be taken into account that the weight of the tank reduces the listed values by approx. 15 %. The fuel cell, strictly speaking, is not an energy source, but rather an energy converter. Since electrical energy is produced, it is generally considered as an alternative to conventional sources of

electrical energy. The actual energy source is hydrogen which has to be carried along.

Energy sources	Energy density [Wh/kg]	Power density [W/kg]
Simple fuel		
Hydrogen	33,326	-1)
Carbon	9,101	-1)
Fossil fuel		
Gasoline	12,200	-1)
Diesel	12,050	-1)
Liquid Petroleum Gas	12,750	-1)
Methanol	5,450	-1)
Electrical energy sources		
Lead-Acid battery	25-35	80-100
Nickel-Cadmium battery	40-55	up to 200
Sodium-Nickel-Chloride battery	90	150
Supercapacitor	1-7	more than 800
Fuel Cell	more than 1,000	
Mechanical energy sources		
Flywheel	5-11	360-750
Hydraulic energy sources		
Pressure accumulator	up to 0.7	300
Thermal energy sources		
Salt-crystallization accumulator	more than 30	more than 3,000

¹⁾ maximum performance is not determined by accumulator

Table 3-1: Energy and Power density of different energy sources

It becomes clear that fossil fuels exceed by far all alternatives in the context of their energy density, except for pure hydrogen. Even in their highest stage of development, electrical energy sources produce poor conditions concerning their employment in motor vehicles, predominantly as a result of their difficult operating behavior. Only the fuel cell presently offers an essential development capacity here.

Mechanical flywheel sources have gained special attention recently. Their advantage essentially depends on the achievable speeds since the stored energy rises quadratically and the frictional losses at the mounts. In newer designs, the supply and withdrawal of energy takes place electrically (Fig. 3-1). Meanwhile there are

series applications of latent-heat energy sources. However, they are not used as primary energy sources. Fig. 3-2 shows the design of such an energy source. The principle involves bringing a salt to its molten state by heat supply and finally withdrawing the thermal energy in the form of latent heat during salt re-crystallization.

In electrical as well as in flywheel and thermal energy sources, the period of the heat storage is limited. Only chemical storage units do not present any problems here.

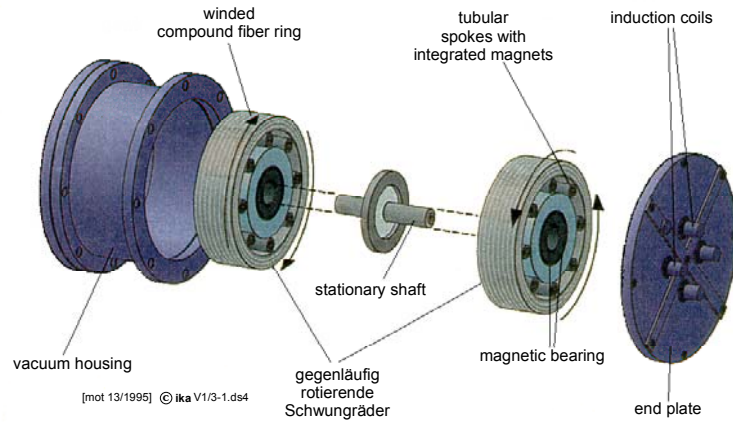


Fig. 3.1: Flywheel storage unit (Source: mot 13/1995)

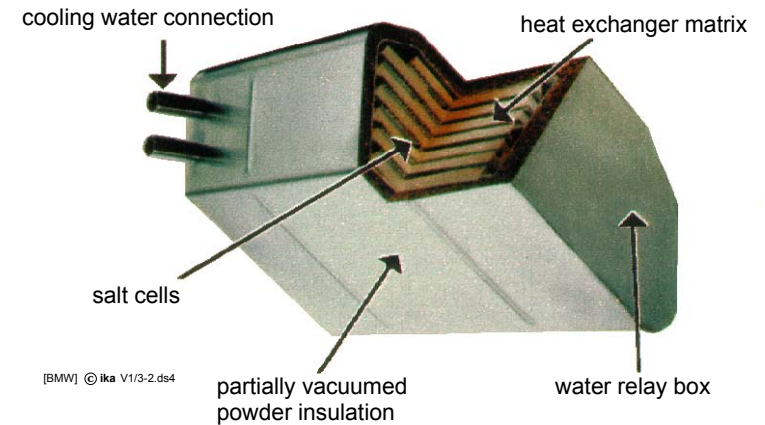


Fig. 3-2: Latent-heat storage unit (BMW)

It is important to consider the possibilities of energy sources since they essentially influence the total quality of a propulsion system. Hence the wide acceptance of internal-combustion engines is understandable.

3.2 Propulsion Units (Engines)

In more than 95 % of all motor vehicles, an internal-combustion reciprocating engine is used for propulsion. However, one should ignore the different propulsion concepts available for motor vehicles (Table 3-2).

Combustion Engine			Electric Drive
Internal Combustion		External combustion	Wire Autonomous
Single-plug Ignition	Continuous Combustion		
Otto engine	Gas turbine	Stirling engine	Electric motor with battery or fuel cell
Diesel engine	Axial-piston engine	Steam engine	
Wankel engine			
Hybrid Propulsion Systems			

Table: 3-2: Overview of motor vehicle propulsion concepts

So far the Stirling engine as well as the axial-piston engine, have not seen application, not only in the scope of automobiles, but also otherwise. The steam engine has to be considered as an antiquated form. In the following chapter the Compression Ignition (diesel) Engine, the Spark-Ignition (Otto) Engine will be considered in detail. As special cases, the Rotary (Wankel) Engine, the Gas Turbine, and the Electric Drive will be described in more detail. Finally, the different principles will be evaluated based on their suitability to motor vehicles.

3.2.1 Spark-Ignition (Otto) Engine

The Otto engine is the most widespread variation of the reciprocating engine with single ignition and internal combustion.

Here, the heated medium and the working medium are identical and represents the characteristic feature of internal combustion engines, meaning that mechanical power is directly extracted from the hot combustion air, which is in contrast to the steam engine.

In order to accomplish this, a combustible fuel-air-mixture is ignited inside the combustion chamber. The combustion energy released in the form of heat increases the pressure of the pre-compressed gases resulting in work being done in the subsequently following expansion cycle. Following each working cycle, the exhaust gas is replaced by a fresh combustible mixture which takes place according to one of two principles: the 2 or 4-stroke principle.

In the 4-stroke system, changes in volume of the combustion chamber are alternately used to produce work and to exchange gases. Gas exchange is achieved using valves. An entire 4-stroke cycle consists of the four phases: intake, compression, expansion, and exhaust strokes. Figure 3-3 shows the gas pressure curve of a 4-stroke reciprocating engine which depends on the working volume and the crankshaft angle, as presented in the $p_i(V)$ - or $p_i(\alpha)$ -diagram.

In the 2-stroke system, the exhaust gases are displaced by a fresh charge between the working cycles. The piston undertakes the control of gas exchange by blocking or opening the inlet and outlet ports. A working cycle is hence reduced to two strokes compression and expansion (Fig. 3-4). In contrast to the 4-stroke engine, no valve train is required for gas exchange. However, in order to manage a sufficient intake of scavenging air, the 2-stroke engines requires a blower. Either the bottom end of the piston - with the crankcase as compressor space - or a separate mechanical compressor takes over this task.

Due to emission norms which have to be fulfilled in the meanwhile, modern 2-stroke engines also require a considerable mechanical expense (valves, injection systems). Their advantage lies in the fact that work is produced in every rotation and not in every alternate rotation as is the case with 4-stroke engines. As a result, very low weight-to-power ratios can be realized. Currently, the chances for penetrating the market cannot be clearly seen.

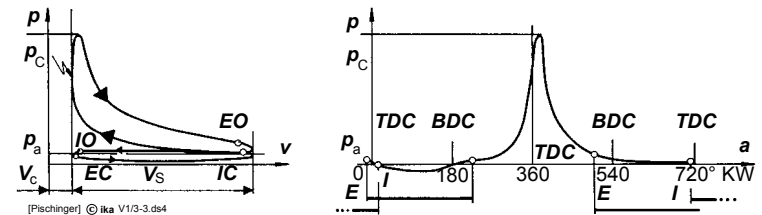


Fig. 3-3: The 4-stroke working cycle (Pischinger)

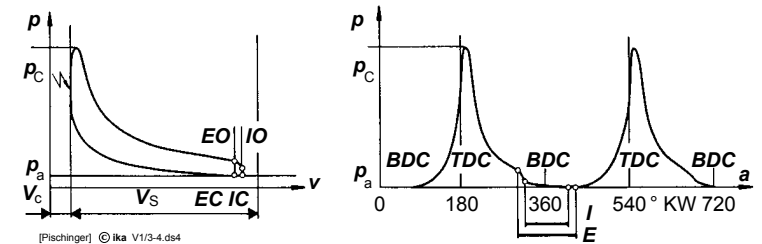
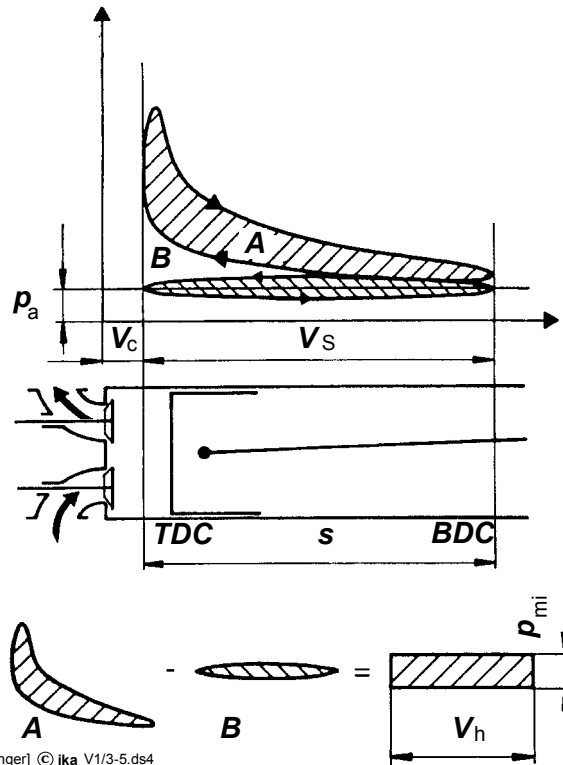


Fig. 3-4: The 2-stroke working cycle (Pischinger)

In the p_i -V-diagram, the positive work performed on the piston corresponds to the clockwise area enclosed (compression and expansion). An area enclosed counterclockwise indicates negative work (intake and exhaust). In the case of the 4-stroke system (Fig. 3-5), the internal work done per working stroke is obtained by computing and subtracting both enclosed areas. When the resulting area is replaced by a rectangle equal in area with the cylinder displacement V_H as a basis, the height corresponds to the indicated mean effective pressure p_{mi} .



2. [Pischinger] © Ika V1/3-5.ds4

Fig. 3-5: Determination of mean effective pressure in the 4-stroke engine (Pischinger)

The above mentioned variable is used for the computation of the indicated power of the engine. The product of p_{mi} and engine displacement describes the work acting on the piston. When associated with a time interval, the indicated power is obtained.

$$P_i = p_{mi} \cdot A_{pi} \cdot s_{pi} \cdot i \cdot n_{eng} \cdot z \quad \text{Eq. 3-1}$$

where: p_{mi} = indicated mean effective pressure

A_{ko} = piston surface

s_{ko} = piston stroke

i = working cycles per crankshaft revolution

3. ($i = 0.5$ for 4-stroke, $i = 1.0$ for 2-stroke)

n_{eng} = engine speed

z = number of cylinders

The effective power P_e at the shaft is smaller by the frictional power P_r caused by internal losses. The ratio of effective to indicated engine power defines the mechanical efficiency:

$$\eta_{mech} = \frac{P_e}{P_i} = \frac{P_i - P_r}{P_i} \quad \text{Eq. 3-2}$$

Through the effective power measured at the flywheel, the mean effective pressure p_{me} is determined analogous to p_{mi} . The mean effective pressure is proportional to the engine torque M_{eng} .

$$p_{me} = \frac{P_e}{V_S \cdot i \cdot n_{eng}} = \frac{M_{eng} \cdot 2 \cdot \pi}{V_S \cdot i} \quad \text{Eq. 3-3}$$

where: $V_S = A_{pi} \cdot s_{pi} \cdot z$

In the spark ignition engine, the indicated mean effective pressure p_{mi} mostly depends on the volumetric efficiency of the cylinders. By increasing the vacuum during intake, the cylinder charge improves as speed goes up in the lower speed range. At high engine speeds, the volumetric efficiency drops as a result of increasing flow losses in the inlet and exhaust channels.

At the same time mechanical losses increase with engine speed. This results in a peak for the mean effective pressure p_{me} and thus for the engine torque M_{mot} being achieved in the mid-speed range. In contrast, the peak effective power is reached at a higher speed, the so-called nominal speed. Below a minimum speed, the engine does not produce torque.

Figure 3-6 shows the power and torque characteristic of a reciprocating internal combustion engine.

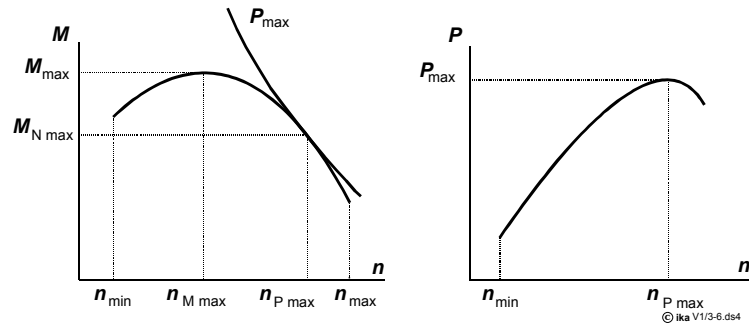


Fig. 3-6: Power and Torque characteristics of a reciprocating engine

Apart from the characteristic, the fuel consumption map (Fig. 3-7) is an important basis for matching the combustion engine, characteristic converter, and motor vehicle. In this case the mean effective pressure p_{me} is plotted as a function of the engine speed n_{Mot} .

The lines of constant power show the curve of hyperbolas. Apart from that, lines of constant specific fuel consumption, the so-called shell curves, are drawn in. The specific fuel consumption corresponds to the fuel mass flow relative to the effective engine power.

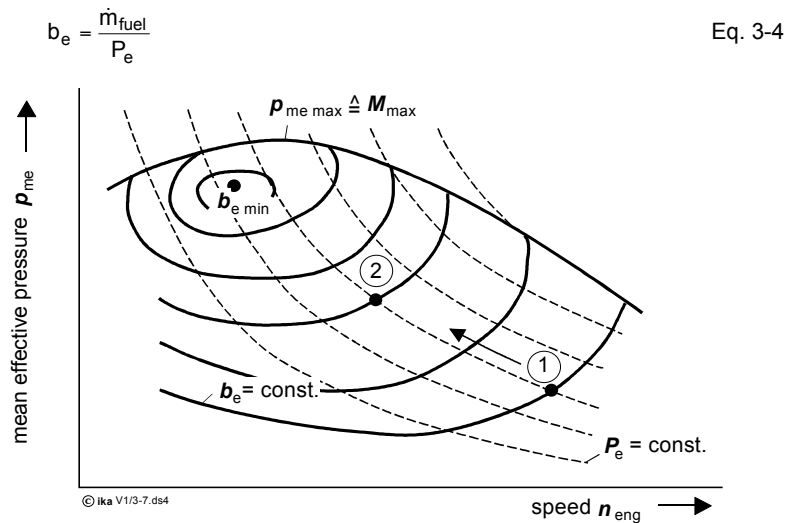


Fig. 3-7: Consumption map

It becomes clear that the fuel consumption of a motor vehicle is significantly influenced by the transmission ratio between engine and drive axle, since the same amount of power can be achieved at different speeds at a distinct specific fuel consumption. For example, by decreasing the speed from point 1 in Fig. 3-7 to point 2 at equal power it is possible to reduce consumption.

3.2.2 Compression Ignition (Diesel) Engine

The diesel engine today, is exclusively manufactured on a large scale as a reciprocating engine. Recently, discussions on the development of rotary piston engines working on the diesel principle have been again re-opened. So far, they have however failed as a result of the high manufacturing expenses involved.

The Diesel engine differs from the Otto engine, in that it compresses pure air which is followed by the injection of liquid fuel under high pressure at the end of the compression stroke. Thanks to the large compression ratio ($\epsilon^4 = 12$ to 22), the engine can also be fired at cold starts in the presence of sufficient hot air. To summarize the concept, these characteristics lead to an internal mixture formation and auto-ignition of a non-homogeneous mixture.

The operation with air-fuel-ratios that are higher compared to the Otto engine, meaning an extremely lean mixture (part load $\lambda^5 = 3$ to 6), implies lower heat transfer and dissociation losses (losses due to incomplete combustion).

In addition, the diesel engine being quality-controlled by fuel injection ($\lambda \neq \text{const.}$), the flow losses in the part-load area are low unlike in the case of the Otto engine where throttling occurs as a result of a quantity controlled strategy ($\lambda = \text{const.}$). All of this leads to an improved efficiency.

$$^4 \quad \epsilon = \frac{V_S + V_C}{V_C}$$

$$\text{where: } V_S = \frac{\pi}{4} \cdot D^2 \cdot s$$

V_C = compression/clearance volume
(remaining cylinder volume when piston is at OT)

D = piston diameter

s = piston stroke

$$^5 \quad \lambda = \frac{m_A}{m_{Astoich}}$$

- λ = air-fuel-ratio
 m_L = actual mass of air
 $m_{Lst\acute{o}ch}$ = stoichiometric mass of air (mass of air required for complete combustion)
 $\lambda > 1$ = excess of air $\Rightarrow$ "lean" mixture
 $\lambda < 1$ = lack of air $\Rightarrow$ "rich" mixture

Diesel engines can be classified into Direct-Injection Diesel Engines and Pre-Chamber or Turbulence-Chamber Engines. While direct-injection diesel engines have been dominating the commercial vehicle segment due to their low consumption for a long time, the high noise and low running culture has prevented its implementation in passenger cars. Meanwhile these problems have been largely solved in new designs.

The torque characteristics dependent on the engine speed are generally flatter compared to the Otto engine (more power at low speeds). As a consequence of higher pressures inside the cylinder, the diesel engine is made of more robust components. This results in a larger connecting-rod and piston mass as well as to a reduction of the permissible engine speed compared to the Otto engine.

In order to enhance the efficiency of diesel engines, turbochargers are often employed. For an additional improvement of the efficiency, they are used in combination with an intercooler. In contrast to Otto engines, a simultaneous improvement of the consumption and emission behavior is possible.

Apart from the Diesel and Otto engine described above, there are special forms of the piston engines which use hybrid combustion methods. This term summarizes methods that have the characteristics of both Diesel as well as Otto cycles. The stratified-charge engine is one example. Fundamentally it corresponds to the Otto engine but burns a non-homogeneous, partially very lean mixture. Its application is however essentially problematic since it does not allow for the use of a conventional 3-way catalyst. Earlier designs have not seen considerable application. Direct-injection gasoline engines have recently been developed to an extent where they can be produced in series. A new catalyst is used in these engines. This engine concept is applied in order to achieve identical fuel consumption values as the diesel engine (Mitsubishi).

Fig. 3-8 illustrates the torque characteristics of different Otto and Diesel engines. The engines shown have similar maximum power outputs between 95 and 110 kW but very different characteristics. The large variation in the design of truck engines compared to passenger cars engines becomes obvious.

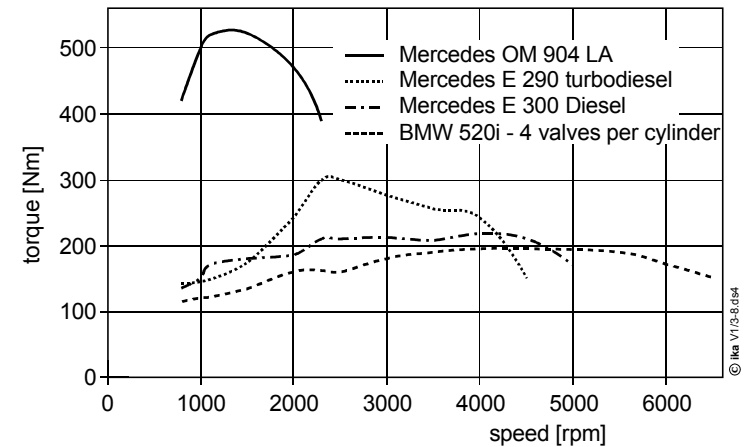


Fig. 3-8: Torque curves of Otto- and diesel engines

3.2.3 Wankel Engine

The previous discussion on internal combustion engines was restricted to reciprocating engines. The only rotary-piston engine produced in series was developed by Felix Wankel. Fig. 3-9 presents the geometry of the engine named after him.

The contour is achieved by superimposing two rotary motions. The traversing radius R rotates around the revolving eccentric point E . In doing so, its end points A as well as B and C , each located at an angle of 120° to R , describe a closed curve in the form of an epitrochoid.

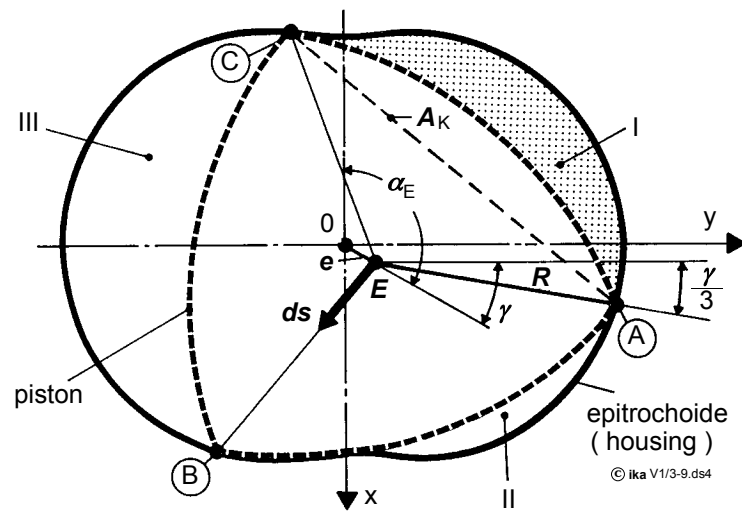


Fig. 3-9: Geometry of the Wankel engine

The points A, B and C, connected by convex sides, determine the outline of the piston. Three working chambers arise whose size changes periodically. The resulting so-called chamber piston displacement is computed:

$$V_S = 3 \cdot \sqrt{3} \cdot w \cdot R \cdot e \quad \text{Eq. 3-5}$$

where: w = chamber width
 e = eccentric radius

The kinematics of the piston are realized through a pinion firmly connected to the housing. At the pinion, the piston revolves over a wooden gear wheel. The ratio of radii of the wooden gear wheel to the pinion is fixed value of 3:2. The piston drives an eccentric shaft that represents the equivalent of the crankshaft in a reciprocating engine. The translational vibrations generated by this motion can be balanced without problems.

The Wankel engine works on the 4-stroke Otto process. In the course of a piston revolution (corresponds to three rotations of the eccentric shaft) the working cycles intake, compression, expansion, and exhaust take place in each of the three chambers. Mixture formation and ignition correspond to the Otto cycle.

The piston is responsible for charge exchange in the Wankel engine. While in nearly all designs the exhaust port flows into the circumference of the housing, the intake port is designed either as a peripheral or side intake. The peripheral intake, in contrast to the side intake, is always open for at least one combustion chamber. As a result, the unhindered flow of the gas column in the inlet port provides low flow losses and a good volumetric efficiency at high speeds leading in turn, to a high maximum output. At low speeds, however, the long overlap periods of the peripheral intake cause scavenging losses since it allows the exhaust gases to flow into the fresh intake air. The consequence is a loss of torque in the low speed range. This torque loss can be prevented by a side intake. In addition, by modifying the position and shape, it offers the possibility to vary the operating times within a wide range and hence influence the power output and efficiency.

In order to achieve a good torque characteristic and low fuel consumption in the part-load area with a simultaneously high maximum output at full-load, systems that consist of several intake ports per piston have to be developed on each side of the housing. The intake ports can be operated individually or in combination according to the operating condition (Mazda).

3.2.4 Gas Turbine

Gas turbines developed for the propulsion of motor vehicles, operate on an open cycle with constant-pressure internal combustion. In a turbo charger, atmospheric air is compressed to 5 bar or more and then preheated by hot exhaust gases in a heat exchanger in order to improve the thermal efficiency. In the combustion chamber, liquid injected fuel is burned with the compressed hot air at constant pressure. Finally the combustion gases expand inside a turbine before they are released over the heat exchanger. One can distinguish between one and two-shaft gas turbines.

Based on current developmental trends, the two-shaft gas turbine is the predominant design. Here, the combustion gases expand in two mechanically independent gas turbines which are positioned in series (Fig. 3-10).

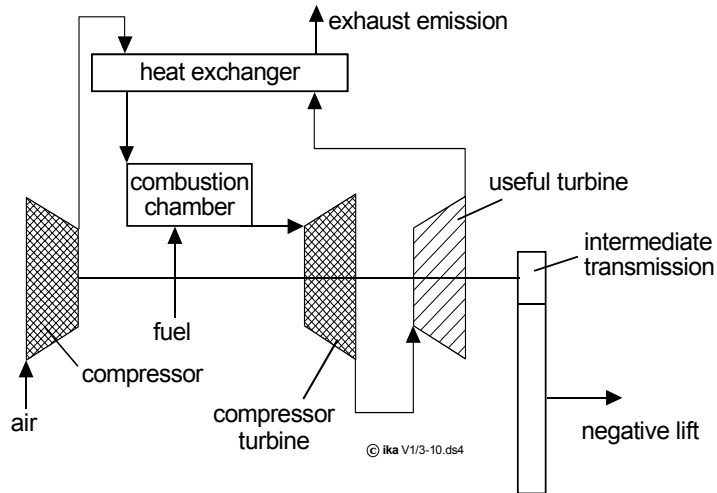


Fig. 3-10: Diagram of a two-shaft gas turbine

The compressor and accessories are driven by a high-pressure turbine. The power output derived from the adjoining low-pressure turbine is exclusively available as useful output for vehicle propulsion.

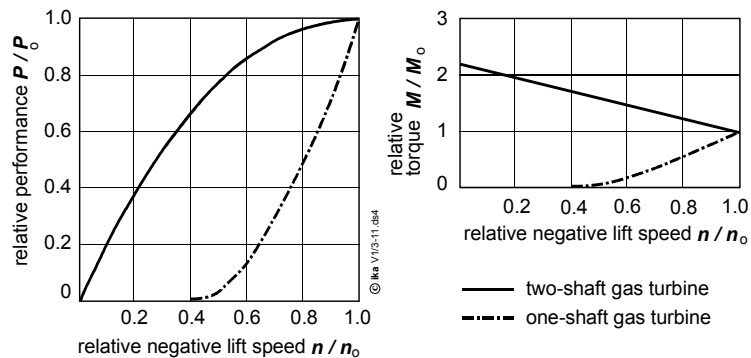
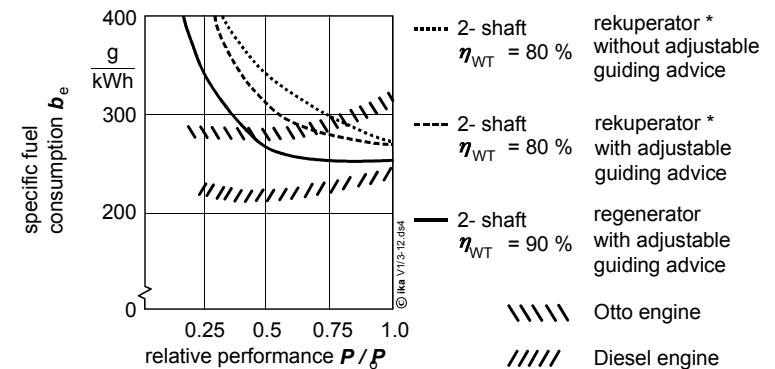


Fig. 3-11: Power- and torque curves of one and two-shaft gas turbines

As Fig. 3-11 shows that, in contrast to the one-shaft gas turbine, the two-shaft gas turbine has a more suitable map for vehicle propulsion. Due to their operating principle being based on fluid mechanics, gas turbines have an unsatisfactory part-load efficiency. Fig. 3-12 shows in a comparison diagram, that the part-load

consumption is definitely higher than traditional reciprocating engines. A significant improvement of the efficiency is possible by increasing the turbine inlet temperature. The continuous thermal load limit for metallic materials, however, lies at a temperature of approx. 1,300 K. As a result, unburned secondary air has to be mixed into the combustion gases (bypass) for cooling. If cooling is to be withdrawn in order to optimize consumption, ceramic materials, which are stable up to gas temperatures of approx. 1,600 K, have to be used in the combustion chamber, inlet cone, turbine blade ring, and turbine blade wheel. The use of ceramic materials, in addition allows for a considerably smaller operating gap between turbine cover ring and blades as a result of lower thermal dilatation. Since losses due to leakage decrease in doing so, a further additional improvement in efficiency can be achieved. In the entire operating area, these measures signify a fuel consumption reduction of 20-30 %. The emission behavior of gas turbines is better than the one of comparable reciprocating engines. Due to the high air flow rate, catalytic after-treatment is not possible.



* The recuperator is a special design of the heat exchanger.

Fig. 3-12: Specific fuel consumption of Gas Turbine, Otto and Diesel Engine

3.2.5 Electric Drives

The attempt to equip motor vehicles with electric motors dates back to the past century. Around the turn of century for example, 15,000 electric-passengers cars were already registered in the United States. The fundamental design of an electric powertrain is shown in Fig. 3-13.

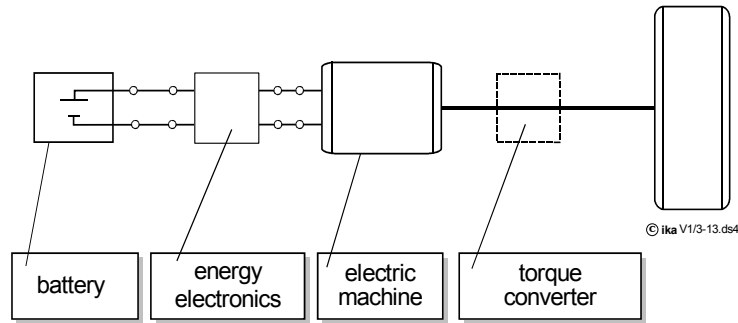


Fig. 3-13: Components of an electric drive

Energy electronics which guarantee the control of the drive is interconnected between the energy source, the battery and the energy converter, the electric motor. In electric vehicles, the Direct-Current (DC) drive is traditionally the most widely used standard drive. The energy supply for an electric vehicle is generally based on direct current. The possibility of simple adjustment by cost-efficient power electronics establishes the direct-current motor as an economically suitable solution. Three different types of connections are available for the direct-current motor (Fig. 3-14).

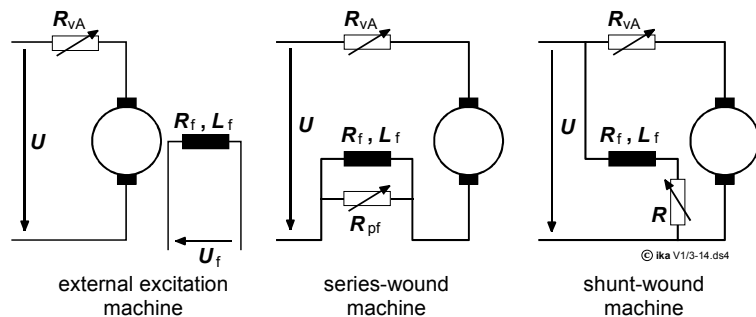
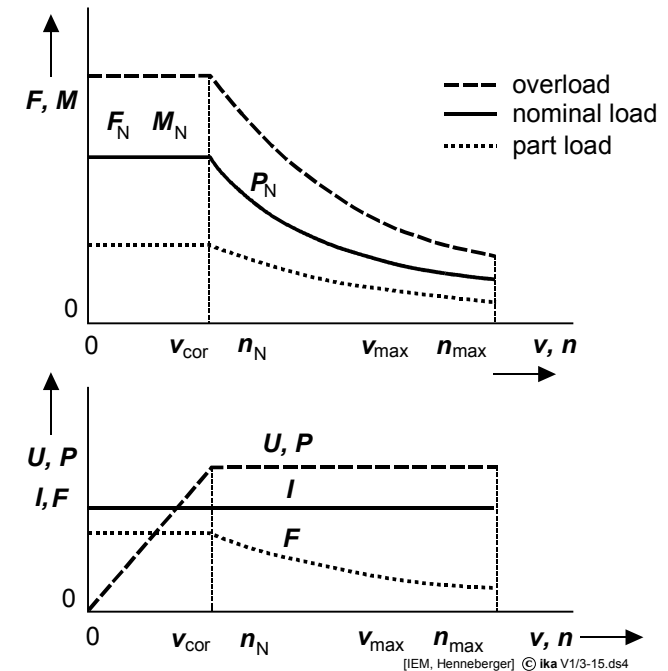


Fig. 3-14: Connecting types of direct-current machines

For electric drives, the externally excited or shunt DC motor is most suitable as a drive with regenerative braking and variable speed. By means of suitable control electronics, these motors can be easily set to any operating point with small losses. Fig. 3-15 shows the motor characteristic of an externally-excited direct-current motor.

In the low speed range, the motor is driven with a variable armature voltage and a constant torque. Above the corner speed, field weakening with a constant power

output is employed. Theoretically a starting clutch could be left out. In practice, however, due to the required large current-control area, clutches are used without exception. Series-wound machines distinguish themselves through their robustness and simplicity but are disadvantageous from the point of view of their control since the entire working flow has to be led over the control equipment in order to vary the speed. Regenerative braking and the reversal of the running-direction require expensive mechanical alterations.



3-15: Fig Motor characteristic of externally-excited direct-current machines

The permanently-excited dc motor has no field weakening and is hence less suitable as a traction motor.

Due to the further development of power electronics, today three-phase alternating-current drives are seeing wider application. The α -controlled asynchronous motor is suitable as a variable speed drive not only in industry but also in automotive applications. Only the bypass rotor is a possibility because it is maintenance free and robust. By employing an expensive field-oriented control, an operating behavior with basic setting and field weakness areas can be achieved which corresponds to the

behavior of dc motors. Particularly for a good efficiency, relatively large areas of field weakness can be presented. The power output is achieved at high speeds but not at high torques. This results in smaller machines. The efficiency is higher compared to direct-current motors but lower than synchronous motors.

The externally-excited synchronous motor consists of an excitation winding on the rotor which can be supplied with the necessary direct current either through commutator rings or using a brushless design. This motor achieves a field weakening which leads to a favorable torque curve. Here, on the one hand it is a disadvantage that the commutator rings are not maintenance-free while on the other hand a contact-free supply is very expensive. Permanently-excited synchronous machines are often used as drives. In this case rare earth magnets are employed that are available at acceptable prices today. In case of constant excitation using permanent magnets, a field weakening can only be realized at additional expense. The advantages of these machines besides the compact design and low power weights are the high efficiency.

A diagram concerning the control of a direct-current motor (GM) and asynchronous (ASM) or synchronous motor (SM) is shown in Fig. 3-16.

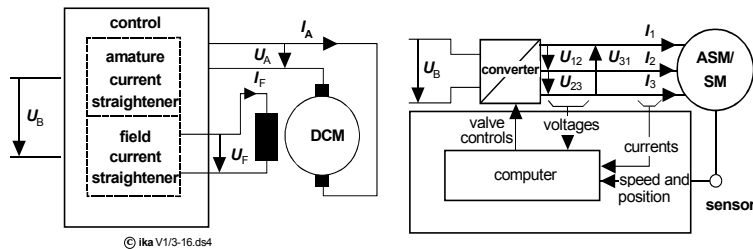


Fig. 3-16: Control of electric machines

As a result of its characteristics, the electric motor fulfills most requirements associated with an automobile drive. In addition, it can be overloaded up to 100% for a short time, operates with low vibrations and noise and is emission-free. Operation and maintenance are relatively simple and along with the constructional possibility of regenerative braking. Its competitiveness has essentially failed so far because of insufficient energy storage. This was already presented in Chapter 3.1. The electric drive becomes more interesting within the framework of hybrid propulsion concepts.

Fig. 3-17 shows an asynchronous machine as wheel-hub motor for a hybrid bus

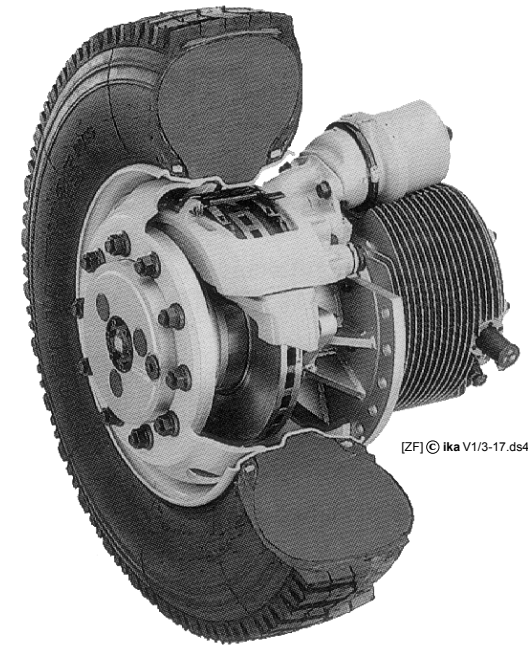


Fig. 3-17: Wheel-hub motor (ZF)

3.2.6 Hybrid Propulsion

Drive concepts which involve at least two different energy sources are called hybrid propulsions or hybrid drives. The objective of such drives is to combine the conceptual advantages of one energy source with another in order to avoid the conceptual disadvantages of individual drives (e.g. environmental influence on the combustion engine, insufficient operating range of electric drives).

From the multitude of realized or theoretically conceivable concepts, only two will be presented more in detail here.

a) Parallel hybrid

Fig. 3-18 shows the design of a parallel hybrid consisting of a combustion engine with fuel tank, electric motor with battery, and transmission.

The characteristic feature of this concept involves a coupling of the torques of both machines ahead of the transmission so that the transmission ratios can be used for both engines.

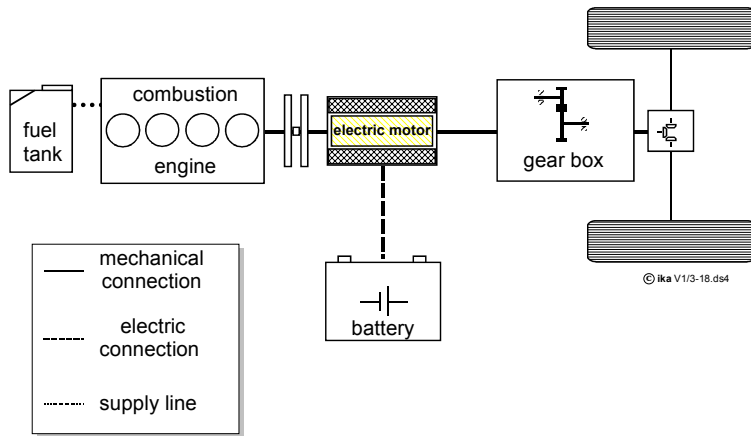


Fig. 3-18: Design of a parallel hybrid

The concept offers three operating conditions. Apart from pure battery operation, the vehicle can also be exclusively driven by the combustion engine.

The third operating condition involves the hybrid drive in which one of the two engines or both as the case may be, are activated by the drive management depending on power requirements and speed. Generally the thing to do would be to cover small power requirements exclusively by the electric motor, while higher speeds and larger power requirements would warrant the need for the combustion engine. When driving at full speed, both engines are activated such that the individual power outputs add up. Minor decelerations serve for regenerative braking. In this case the electric motor operates as a generator. The resulting energy is supplied back to the batteries.

The ranges of power output covered by the individual propulsion systems are presented in Fig. 3-19 using an example. The advantage of this concept lies in its relative simplicity apart from its economical, structurally-simple and light electric motor, small battery relative to the pure electric vehicle and the possibility of partially emission-free operation. In contrast, the potential for regenerative braking reduces as a result of the small electric motor, apart from the additional weight as a result of batteries and electric motor when compared to a pure combustion engine drive.

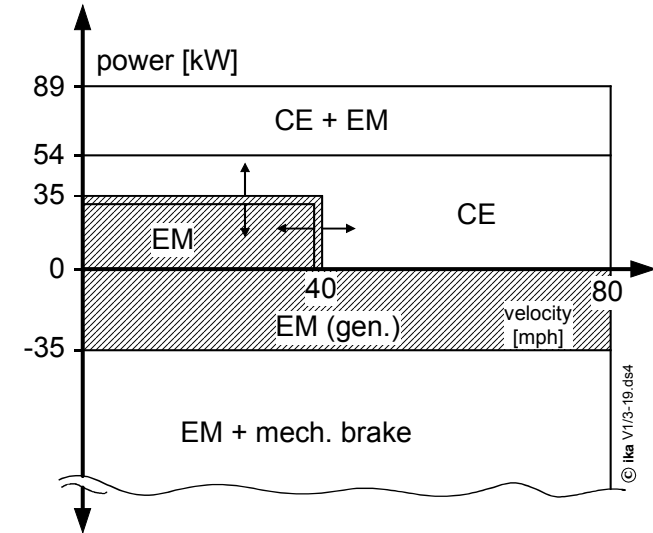


Fig. 3-19: Power splitter in hybrid propulsion

b) Serial hybrid

Fig. 3-20 indicates the structure of this concept. The characteristic feature of this concept lies in the serial positioning of its components and the double power conversion of the combustion engine. The disadvantage being that the efficiency of the electric machine appears as a product in the energy demand is offset against the advantages resulting from the mechanical decoupling of the combustion engine from the drive train and the existence of a short-term storage unit. The power output of a combustion engine is largely independent of the vehicle's power requirement. As a result, the combustion engine can be operated at steady-state and more or less stationary. By designing a combustion engine for a small operating area, an additional improvement of efficiency can be achieved.

The characteristics of the short-term storage unit essentially influences the advantages of this concept. In traditional batteries, the power density presents a problem (Table 3-1). The amount of instantaneous energy available is limited and the efficiency sharply drops at quick discharges. In other words, the battery would have to be designed rather large and bulky. This is the reason behind the use of electro-mechanical flywheels or super-condensers as a short-term storage units in alternative concepts.

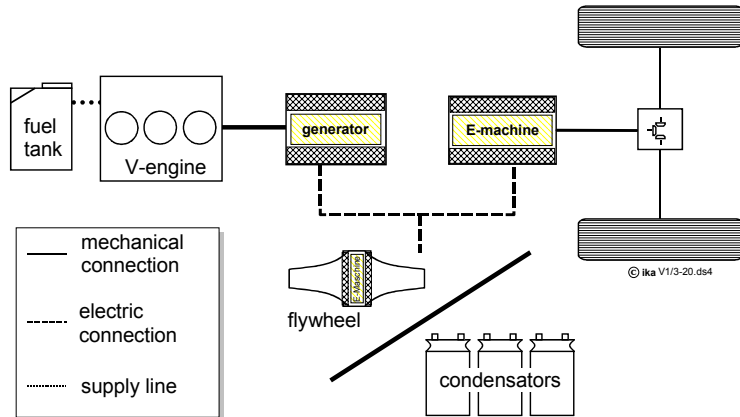


Fig. 3-20: Structure of a serial hybrid

In choosing a suitable operating strategy, attention has to be paid in order that, on the one hand, the combustion engine operates along a favorable characteristic, while on the other hand, as little energy as possible is stored in between, in order to minimize the associated losses.

Compared to the parallel concept, the advantage of the serial concept lies in its better emission behavior as a consequence of its more or less stationary engine operation, higher potential for regenerative braking when using flywheels or condensers, better driving comfort, and lower excess weight. However, the complex control strategy and short-term storage unit technology which has not yet matured, still poses problems.

3.2.7 Comparison of Propulsion Units

3.2.7.1 Selection and Assessment Criteria

In order to choose a suitable propulsion system, the required power output for the respective vehicle and the torque characteristics against speed are of primary importance. A number of other factors are also to be considered. They can be summarized into the following main groups:

- Operating behavior,
- Economy, and
- Environmental influence.

Initially, these criteria will be explained in more detail and then applied to different drive concepts.

a) Operating behavior

The demand characteristics for the motion of a vehicle which is given by the sum of the individual road resistances, is set against the delivery map in a drive-end way. Initially, it is determined by the characteristics of the drive concept. A general presentation describes the possible power or torque characteristic of the propulsion aggregate against speed. This area is limited at its upper end by a characteristic line, the so-called full-load curve. Beyond this curve, no higher power output or higher torque can be generated. Since:

$$M \sim \frac{P}{n} \quad \text{Eq. 3-6}$$

the torque delivery takes place along a characteristic in the form of a hyperbola for the respective full-load curve of the delivery map.

Due to a limited traction between tires and road (Chapter 4.2), only a finite torque can be transferred on to the driven axle of a vehicle. As a result, the torque of the propulsion engine can also be limited to a corresponding maximum value.

The maximum speed represents a further limit which is either dependent on the engine or the maximum speed of the vehicle. A delivery map results from these limits, as shown in Fig. 3-21.

It is also an advantage for a motor vehicle to use a relatively high share of the engine power for so-called engine braking. This means that the delivery map should be expandable in the negative range of power output and torque, possibly a mirror-image against the speed axis. Ideally, the kinetic energy that is set free can be completely regained and returned to the vehicle's energy sources.

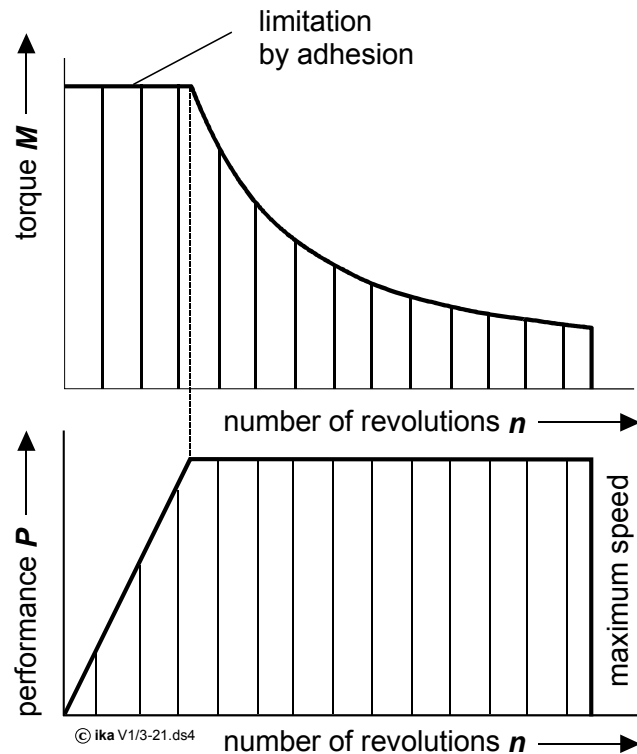


Fig. 3-21: Delivery map limited by traction and maximum speed

By anticipating the concluding comparison of different propulsions, Fig. 3-22 compares some delivery maps.

The steam engine has an almost ideal delivery map. An additional map conversion is superfluous here. This explains the significant role played by the steam engine as a drive concept for a long time. Locomotive drives have to fulfill identical requirements as in the case of motor vehicles. Around the turn of century, automobiles were also equipped with steam engines. Electrical drives also indicate a favorable delivery map. The map of a diesel engine, however, seems to be only slightly suitable for motor vehicle applications. The following chapters will deal in detail with necessary torque and speed conversion.

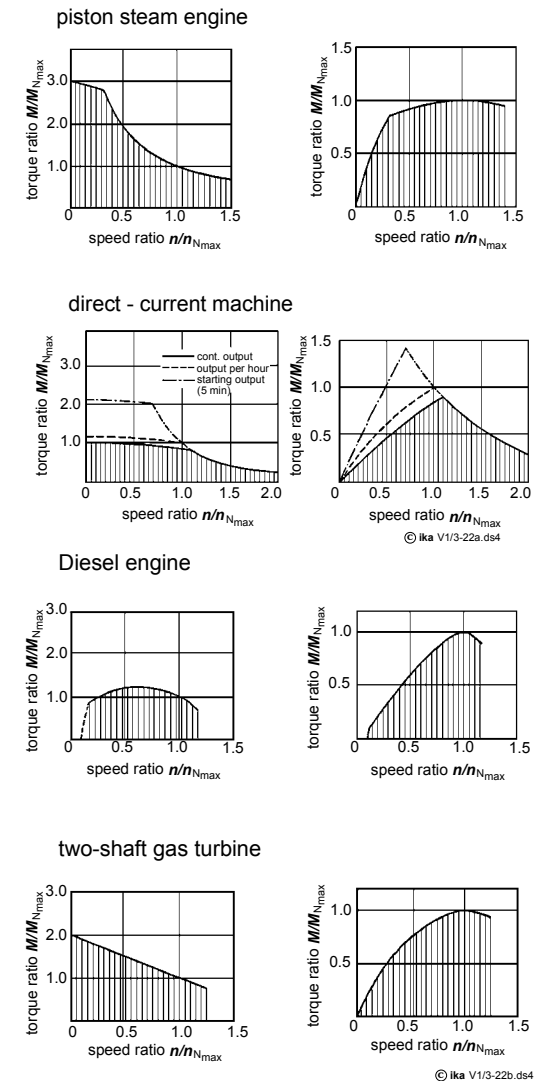


Fig. 3-22: Delivery maps of different motor vehicle propulsions

Since the combustion engine represents the conventional vehicle propulsion, it is of considerable interest to compare the maps of different combustion engines in more

accurate detail. The creation of indices is a simple way to do so. The following characteristic points on the full-load curve are defined for the sake of comparison:

Point	torque	speed	power
1 permitted speed	M _{n max}	n _{max}	P _{n max}
2 maximum power	M _{P max}	n _{P max}	P _{max}
3 maximum torque	M _{max}	n _{M max}	P _{M max}

Table 3-3: Characteristic points in the engine map

Fig. 3-23 illustrates the position of points in the map.

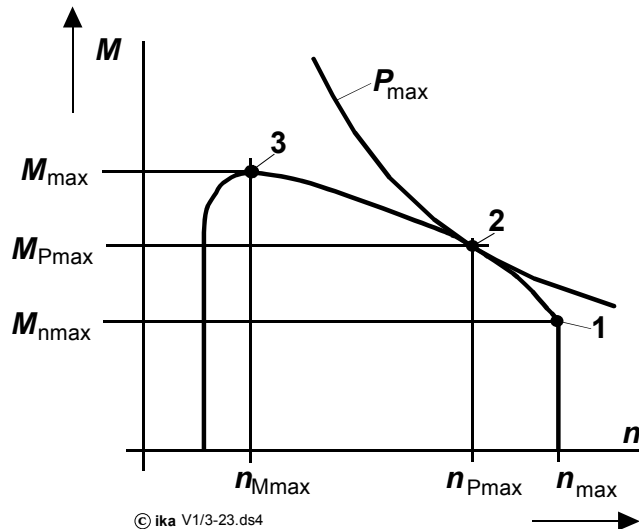


Fig. 3-23: Characteristic points in the engine map

Theoretically a number of indices can be used. In particular, the description of the engine elasticity is interesting. For passenger cars this means that a high traction force can already be transferred at low speeds. This reduces the need for frequent gear changes and hence increases driveability. The speed elasticity represents a measure of elasticity. It is defined as:

$$e_n = \frac{n_{P \max}}{n_{M \max}}$$

Eq. 3-9

In trucks, however, at decreasing speeds or driving velocity, an increasing traction force is required, e.g. when driving on gradients. The torque elasticity serves as a measure here. It is defined as:

$$e_m = \frac{M_{\max}}{M_{P \max}}$$

Eq. 3-8

Fig. 3-24 shows a compilation of indices of different vehicle engines. As it can be noticed, the computation of the total elasticity $E = e_m \cdot e_n$ does not make sense.

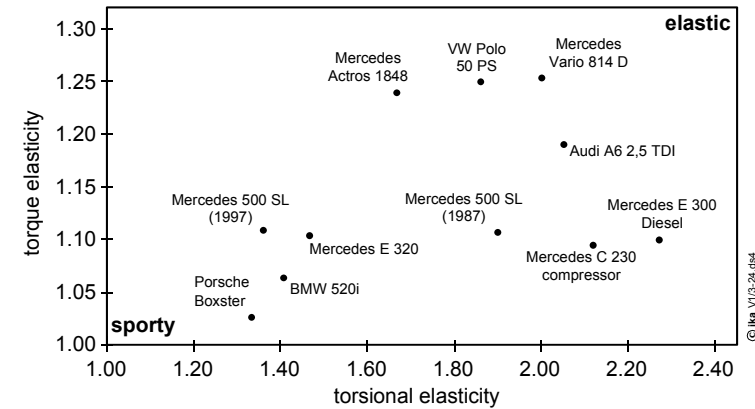


Fig. 3-24: Speed- and torque elasticity of selected aggregates

b) Economy

Economic criteria play an outstanding role in the evaluation of vehicle components (Chapter 1.2). This is particularly applicable to the powerplant since it repeatedly influences the economy of the entire vehicle:

- The operating costs as a result of fuel consumption and maintenance costs,
- The fixed vehicle costs (depreciation and interest) as a result of production costs and the proportional sales price,

- The payload as a result of the mass density (the engine mass relative to power), since the possible capacity weight in

a commercial vehicle can be reduced by the mass of the powertrain.

Since a vehicle constantly experiences varying driving conditions, the engine should operate at a favorable efficiency over a broad power band between part and full load.

Fuel consumption is a decisive factor, most of all in the commercial vehicle sector, since the annual mileage here is approx. ten times that of passenger cars. Therefore higher specific production costs of the drive concept are acceptable. If, for example, the annual mileage of a heavy truck is 150,000 km (93,210 miles), a reduction in the average specific fuel consumption from 225 g/kWh to 200 g/kWh would result in a saving of approx. 7,000 DM per year for a fuel price of 1.15 DM/l.

The economy of a passenger car, however, depends to a large extent on the production costs (Chapter 1.2).

The significance of the mass and volume of the of the entire drivetrain, consisting of the engine with auxiliaries, speed and torque converter, and energy sources, relative to power is common to all motor vehicles. These parameters should be kept as small as possible since the pay load and useful volume would otherwise be reduced. Particularly in future vehicles with increased passive safety, compact and lightweight propulsions are a real advantage from the point of view of the required deformation zones and additional installations. This among others, explains the shift of trend from 6-cylinder inline engines to the V-engine (inspite of unfavorable vibrational behavior).

Moreover factors such as maintenance costs, expected life-span, and reliability influence the economy of a vehicle's powerplant. In this context one refers to the "cost of ownership" which is normally determined for the entire vehicle.

In the context of increasing raw material shortages or taxation on raw materials, possibility to use multiple energy sources and the recycling of engine materials during its disposal could be significant from the point of view of the total economy of a vehicle.

c) Environmental influences

In order to examine the the motor vehicle's influence on the environment in its entirety, attempts have been made to draw up so-called ecological balance sheets. These balance sheets include vehicle production and disposal as well as vehicle operation.

As far as individual components are concerned, the powerplant has a significantly larger influence. In this respect, exhaust emissions are the focus of interest. The automobile makes a significant contribution to air pollution, as shown in Fig. 3-4.

	annual mass emission	share [%]
Traffic	6.3 Mio t	46
Power Stations	2.7 Mio t	19.7
Households and Trade	1.9 Mio t	13.9
Industry	2.8 Mio t	20.4

Table 3-4: Share in total air pollution in the Federal Republic of Germany (Source: Verkehr in Zahlen 1996)

Through statutory measures, the permissible values of the certain pollutants have been reduced step by step.

The four significant toxic components of combustion engine exhaust are: carbon monoxide (CO), unburned hydrocarbons (HC), nitrogen oxides (NO_x), and sulphur dioxide (SO₂).

Table 3-5 shows the proportion of toxic emissions related to vehicular traffic in the total emission of specific toxic substances in West-Germany in 1990.

Emission		mass share [%]
Carbon monoxide	CO	61.4
Carbon dioxide	CO ₂	20.2
Unburnt hydrocarbons	HC	34.7
Nitrogen oxides	NO _x	58.0
Sulphur dioxide	SO ₂	2.1

Table 3-5: Contribution of traffic in total exhaust emission (Source: Verkehr in Zahlen 1994)

The absolute amount of pollutants classified based on passenger cars and commercial vehicles emitted by motor vehicles in the Federal Republic of Germany in 1990, is shown in Fig. 3-25,.

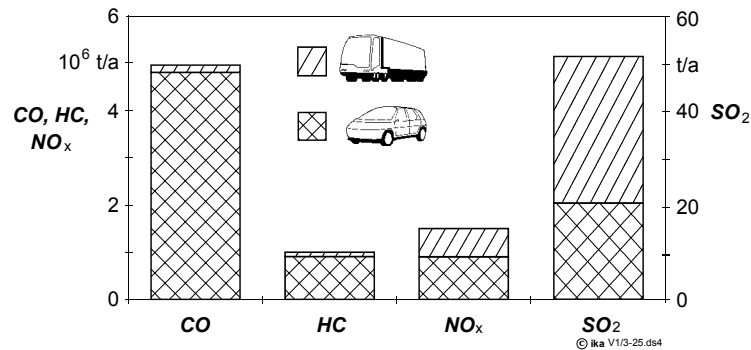


Fig. 3-25: Annual emission of motor vehicles in the Federal Republic of Germany in 1990

Legislation on the permissible composition of exhaust is being continuously tightened. Fig. 3-26 shows the periodic change in the limits of motor vehicle emissions. Apart from a progressive decrease of the permissible carbon monoxide emissions, since 1977, it has been required by law that the sums of HC and NO_x are maintained.

The new legislation applicable for Europe since 1988, no longer uses the vehicle mass as a basis for evaluation, instead the engine displacement. From that point of view, the numerical values can only be compared with each other to a limited extent.

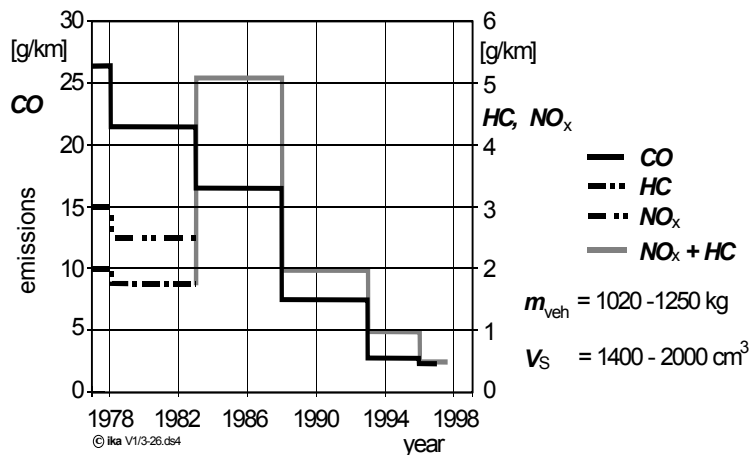


Fig. 3-26: Limits on exhaust emissions in Europe

In the context of the indicated regulations, questions arise as to what direct effect the mentioned exhaust components have on human beings and at what concentrations serious consequences on health have to be taken into account.

Carbon monoxide is a severe blood poison. Its affinity to hemoglobin is about 300 times larger than oxygen. Half an hour exposure to air with 0.3 Vol-% CO can lead to fatal internal asphyxiation. Due to the additive effect, a long exposure to air with a low CO-concentration can be even more dangerous than a short one in a high concentration. Tests have shown that exposure to a CO-concentration of 30 ppm for a couple of hours, can already lead to respiratory difficulties.

Unburned hydrocarbons have adverse effects on health only at high concentrations. However, partially burned hydrocarbons are undesirable due to their annoying smoke and irritation to the eyes and breathing.

Oxides of nitrogen, similar to carbon monoxide, are severe blood poisons. A concentration of 700 ppm leads to fatal lung oedema. The MAK-value (maximum concentration at workplace) amounts to 25 ppm.

In addition, compounds of lead in engine emission resulting from the use of leaded fuel are poisonous and are transferred into the human body mainly through inhaled air and also through the stomach and intestine.

Particulate emissions are also poisonous for human beings, most of all due to substances present in the particles which have been found to be carcinogenic.

In contrast to the direct effects on human health, the influence of indirect effects upon human living space is increasingly gaining in significance in discussions related to pollution. In this case one should mention the responsibility of vehicle emissions in forest depletion which has not entirely been investigated. Moreover, the atmospheric temperature increases as a result of the increasing concentrations of carbon dioxide in the air due to the high consumption of fossil fuels (Greenhouse effect). Vehicles contribute considerably to this effect.

Repeated attempts have been made in the past in order to use renewable fuels instead of fossil fuels to power vehicles. Fuels made out of a bio-mass are theoretically CO₂ neutral and their application in combustion engines technically does not present a problem. On the other hand, there are a number of disadvantages. Almost half of the energy generated is re-used for production and processing. The surface area required in order to produce energy worth mentioning is considerable. The production is inevitably connected with the emission of new pollutants (fertilizers, pesticides). At the moment biological fuels are generally uneconomical compared to

crude oil. In the 70s and 80s Brazil is the only country that took essential steps to promote alternative fuels (methanol derived from sugarcane), but has today given up.

An additional aspect of vehicular environmental pollution is noise emission. The noise emission limits enforced by law, are partially shown in Fig. 3-27. The frequency-evaluated sound-pressure level in dB(A) in a definite driving condition (accelerated drive-by) serves as a measure for the disturbance resulting from noise.

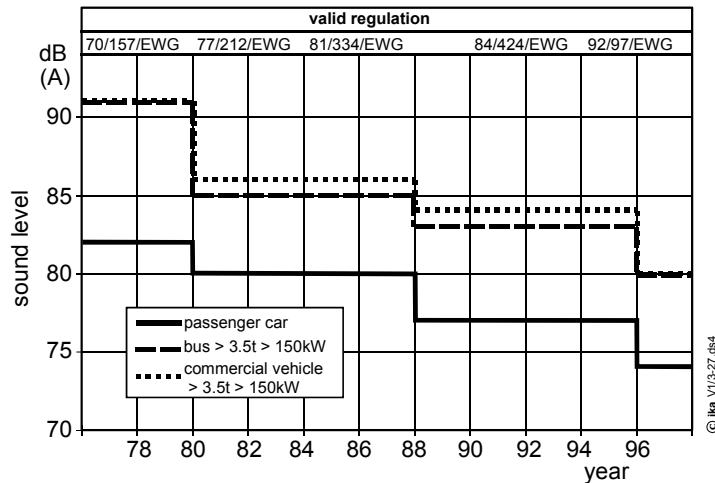


Fig. 3-27: Sound-level limits for motor vehicle registration in the EC

One final selection criterion that still belong to the area of environmental influences is the vibratory characteristic of vehicle propulsion. Besides the noise strain, vibrations of the engine that are too strong can annoy the vehicle's passengers and cause material fatigue of the power train components or the engine mounts.

Finally, the assessment criteria can be summarized as follows:

Operating behavior

- Driving characteristic
- Start-up behavior
- Control behavior and startability

Economy

- Specific energy or fuel costs
- Specific production costs
- Power-to-weight ratio and power-to-volume ratio
- Maintenance costs
- Capability to use multiple fuels
- Durability and reliability

Environmental influences

- Exhaust emission
- Noise emission
- Vibratory behavior

3.2.7.2 Assessment of Propulsion Units

The above-mentioned assessment criteria are applied to the following propulsions:

- Otto engine in naturally aspirated version (OI)
- Otto engine with exhaust-gas turbocharging (OT)
- Diesel engine in naturally aspirated (DI)
- Diesel engine with exhaust-gas turbocharging (DT)
- Wankel engine (WE)
- Gas turbine with internal combustion (GT)
- Electric motor with conventional lead battery (EL)

An assessment of 10 points implies: very good, with 0 points: very unfavorable.

a) Operating behavior

The delivery map of the different propulsions is shown in Fig. 3-28. A common characteristic line is drawn in for all reciprocating internal combustion engines. This characteristic line, like the other characteristic lines, represents the typical behavior.

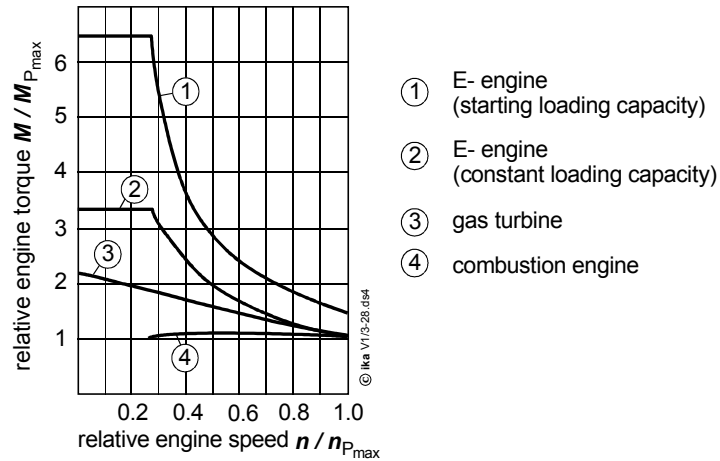


Fig. 3-28: Delivery map of different vehicle propulsions

The electric drive turns out to be significantly superior both in permanent operation as well as during the start phase, i.e., for short-term overload. The characteristic of the two-shaft gas turbine can be used rather well for vehicle propulsion. Reciprocating engines have a satisfactory behavior only when they are connected with characteristic converters.

Without going into further detail here, the operating behavior of the various propulsion units can be presented by the following assessment:

Propulsion	OI	OT	DI	DT	WE	GT	EL
Torque characteristic	4	3	4	4	4	8	10
Starting behavior	2	2	2	2	2	7	10
Control response and startability	9	8	8	9	9	4	10

Table 3-6: Assessment of operating behavior

The torque lag of engines with exhaust-gas turbocharger in the lower speed range is explained by the (smaller) difference in points between supercharged and naturally aspirated versions. As a result of the usual employment of characteristic converters in all propulsions, the torque characteristic and start-up behavior are of limited significance.

When it comes to control behavior and startability, the electric drive is most advantageous. The control behavior of naturally aspirated engines just slightly differs. The characteristic sluggish response of the turbocharged versions leads to their low assessment. The overall control behavior of the gas turbine when compared with today's economically feasible technology is poor. In the start phase, only the pre-chamber diesel engine and the gas turbine engine require a preheating time.

b) Economy

In Table 3-7, the above-mentioned economy criteria have been applied to different engine designs and assessed.

Among the competition, the assessment of specific operating costs is poor in the electric motor because of the generally low power output. The diesel engine with exhaust-gas turbocharging is assessed more favorably.

Propulsion	OI	OT	DI	DT	WE	GT	EL
Specific operating costs	7	7	8	10	7	7	2
Specific manufacturing costs	10	10	8	10	10	5	0
Power-to-mass ratio	9	10	6	8	9	8	0
Capability to use multiple fuels	1	1	1	1	1	7	10
Maintenance effort	8	8	8	8	9	9	7
Reliability and service life	9	9	10	10	9	8	5

Table 3-7: Assessment of the economy

For the purpose of better comparability, the production costs are also referred to the power output. The underlying costs are average values for the propulsion units produced on large-scale. In the case of gas turbines and electric drives, however, estimate values are used for larger volumes.

Average values for the power-to-weight ratio for reciprocating engines were also determined. The turbocharged Otto engine has the smallest power-to-weight ratio, followed by the Wankel engine and the Otto engine. In the diesel power plant, the high operating pressures lead to a bulky design. A power-to-volume ratio comparison would show nearly corresponding relations with the power-to-weight ratio.

When considering the capability of propulsion units to use multiple-fuels, piston power plants can only be operated with alternative fuels following partly large and constructive modifications. In the gas turbine, however, for a large number of liquid fuels this is possible without additional measures. The electric motor can, at least indirectly, fall back on nearly all energy types, since wind, solar, and hydro-energy are used to produce electricity.

Maintenance costs are relatively small for all propulsion types. In case of piston power plants, the Wankel engine has a slight advantage due to the absence of a valvetrain. However, the increasing employment of valves with hydraulic compensation of clearance for the most part levels out this supposed disadvantage of other piston engines. Gas turbines and electric motors similarly require small maintenance costs as in the case of the Wankel engine, though today's lead accumulators still require regular maintenance.

When comparing the life-span of the various power plants, the diesel engine gets the highest values due to its robust design and the low speed level. The gas turbine is assessed lower because of its lower stage of development. It is true that the electric motor has the longer life, but as a total system it is assessed lowest because of its short battery life.

c) Environmental influence

In order to assess the environmental influence, in Table 3-8 assessments of average exhaust emissions, noise, and vibrations of the different propulsion systems were compared.

Propulsion	OI	OT	DI	DT	WE	GT	EL
Pollution emission	3	3	8	8	2	9	10
Noise emission	5	5	3	3	7	6	10
Vibrational behavior	8	8	6	6	10	10	10

Table 3-8: Assessment of environmental influence on propulsions

The distribution of points for exhaust emissions was averaged by individually assessing different pollutants. In this case, the employment of exhaust catalysts which leads to a significant shift in favor of the Otto, stratified-charge, and Wankel engine, is not considered. The electric drive does not produce exhaust emission on a local level, but it should not be ignored that pollutants emerge when electricity is produced in conventional power plants.

When it comes to noise development, the electric drive is lower on average by 10 dB(A) when compared to conventional power plants. Also the gas turbine functions more quietly than piston power plants due to continuous combustion. The high frequencies of compressor and turbine however have a negative effect. Among piston engines, the Wankel engine has slight advantages compared to the Otto and stratified-charge engine due to the absence of a valvetrain. The diesel engine is the loudest. Finally, one has to distinguish between supercharged and naturally aspirated version: supercharged engines work insignificantly more quiet since the turbocharger causes a damping of the intake noise.

Considering the vibratory behavior, Wankel engine, gas turbine, and electric drive are better than the reciprocating power plants. High translational accelerations occur in reciprocating engines as a result of the piston kinematics. These accelerations depend on the number of cylinders and their arrangement and can only be partly compensated.

It has to be taken into consideration that the presented distribution of points can only demonstrate tendencies that do not have to be valid for every isolated case. However, they serve as an overview and also for orientation.

3.2.7.3 Summary

The degree of sophistication and the reasonable production costs will ensure the fact that the reciprocating power plant will maintain its dominant role in the vehicle powertrain in the future. Meanwhile the Wankel engine has reached a high stage of development and is hence capable of competing with the reciprocating engine. Its disadvantage in reference to environmental influences can be remedied by the use of secondary emission control systems.

For the moment, the large-scale production of gas turbines seems improbable due to the high production costs involved even in the absence of ceramic components and the poor part-load efficiency. The fundamental problem of all internal combustion engines which is the unfavorable delivery map, can be solved with moderate effort by the use of appropriate speed and torque converters. The application of electric drives in large-scale seems to make sense only for special applications in the medium-term as a result of serious disadvantages in all decisive assessment criteria for economy. Its future will depend on the development of efficient storage systems. It seems to be possible that it will be employed within the scope of hybrid drives.

3.3 Speed Converters (Clutches)

Speed converters (clutches) are used in the drivetrain of vehicles in order to level out speed differences between the input and output. This is valid on the one hand during the starting phase in which the speed difference between standstill and the lowest engine operating speed have to be bridged, while on the other hand during the shift phases of commonly used manual transmissions. Speed converters are almost exclusively based on adherent or frictional coupling methods. Fig. 3-29 shows the power balance of a speed converter with the corresponding input and output parameters.

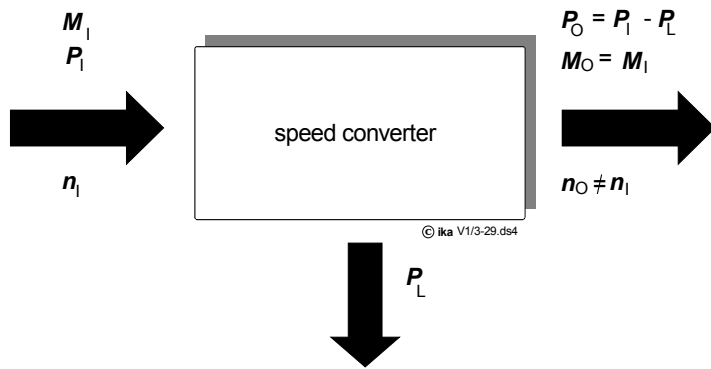


Fig. 3-29: Input- and output parameters of a speed converter

A constant torque is characteristic for a speed converter. The power output P_A is smaller by the power dissipation P_{diss} when compared to the input power P_I . From the ratio of power output to the power input, the clutch efficiency η_C can be calculated.

$$\eta_C = \frac{P_O}{P_I} \quad \text{Eq. 3-9}$$

implying:

$$\eta_C = \frac{P_O}{P_I} = \frac{M_O \cdot 2 \cdot \pi \cdot n_O}{M_I \cdot 2 \cdot \pi \cdot n_I} = \frac{M_O \cdot n_O}{M_I \cdot n_I} \quad \text{Eq. 3-10}$$

considering $M_O = M_I$:

$$\eta_C = \frac{n_O}{n_I} \quad \text{Eq. 3-11}$$

The slip s , defined as the difference of input and output speeds relative to the input speed, is a standardized dimension used to represent the operating condition of a speed converter.

$$s = \frac{n_I - n_O}{n_I} \quad \text{Eq. 3-12}$$

From the formula for efficiency:

$$s = 1 - \frac{n_O}{n_I} = 1 - \eta_C \quad \text{Eq. 3-13}$$

Based on the definitions of efficiency and slip, during a coupling process a relationship totally dependent on speed can be deduced for the power dissipated in the form of heat.

$$P_{diss} = P_I - P_O = (1 - \eta_C) \cdot P_I = s \cdot P_I \quad \text{Eq. 3-14}$$

The clutches used in different fields of application in a motor vehicle, can be classified based on the type of power transmission into:

- Mechanical clutches
- Hydrodynamic clutches
- Visco-hydraulic clutches

3.3.1 Mechanical Clutch

Mechanical clutches in combination with mechanical stepped transmissions are widely used in passenger cars and are normally realised in the form of dry single-plate clutches. In this case, the friction between input and output shafts is produced over circular disks which rub against each other.

Fig. 3-30 illustrates the conversion of the engine characteristic graph by a mechanical clutch.

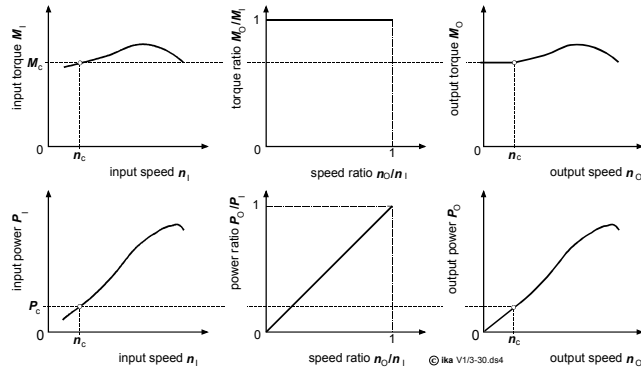


Fig. 3-30: Conversion of the engine characteristic by a mechanical clutch (Source: Mitschke)

As the vehicle starts, the engine in the operating point C produces the torque M_C at speed n_C . This input torque is transferred to the output side at the same magnitude as long as n_O is smaller than n_C . The starting vehicle accelerates until a speed equality is achieved, resulting in: $n_i = n_O = n_C$. Once the clutch is entirely closed, it is be driven according to the engine graph.

In the case of the disk clutch, the magnitude of the transferable torque depends on the applied force F_C between the disks, the frictional coefficient μ_C of the friction pairing, the effective frictional radius r_{mC} and the number of frictional pairs z_C :

$$M_C = F_C \cdot \mu_C \cdot r_{mC} \cdot z_C \quad \text{Eq. 3-15}$$

$$\mu_C \approx 0.2 - 0.3$$

The mean frictional radius (new condition) is calculated as follows:

$$r_{mC} = \frac{2}{3} \cdot \frac{r_{eC}^3 - r_{iC}^3}{r_{eC}^2 - r_{iC}^2} \quad \text{Eq. 3-16}$$

where: r_{eC} = external radius of clutch disk
 r_{iC} = internal radius of clutch disk

It can also be approximately calculated using a simplified formula:

$$r_{mC} \approx \frac{r_{eC} + r_{iC}}{2} \quad \text{Eq. 3-17}$$

The axial force is generally produced by a preloaded spring. By varying the spring travel, in the vehicle the transferred moment can be adjusted as one likes between “zero” and “maximum” using the clutch pedal.

Due to poor heat dissipation, the mechanical dry clutch can only absorb a limited loss of heat. As a result of this, the speed conversion is timewise limited. Today's linings consist of so-called Fillers. Metallic splinters are sometimes part of it. They are held together by an organic bonding agent. They can deal with high temperatures and are abrasion and wear-resistant. Their life often corresponds to the entire vehicle life.

Based on the type of spring used to create the axial force, clutches may be classified into coil spring clutch and diaphragm spring clutch as shown in Fig. 3-31. Due to the low production costs and the favorable spring characteristic, diaphragm springs or disk springs and are nearly exclusively used today.

Fig. 3-32 shows the design of a diaphragm spring clutch in the clutched and de-clutched conditions. The spring force brings the clutch disk into adherent contact with the flywheel over the pressure disk. The clutch disk transfers the engine torque into the transmission. In the declutched condition, the pressure disk is withdrawn from the clutch disk. The torque transmission from engine to transmission is disconnected.

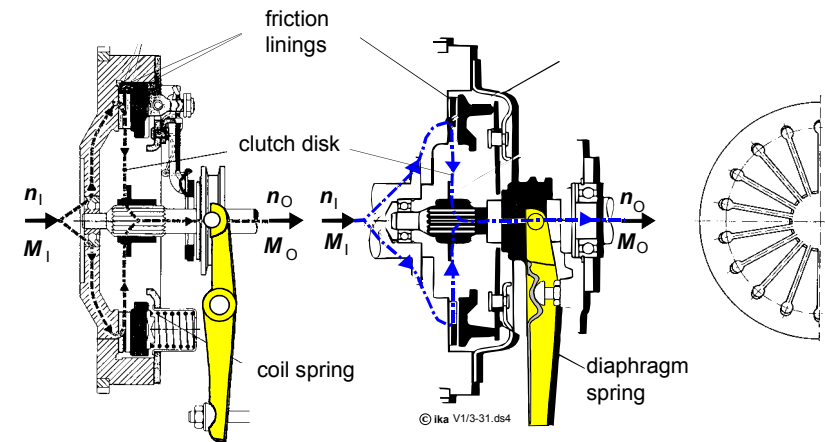


Fig. 3-31: Sectional drawings of coil and diaphragm spring clutch

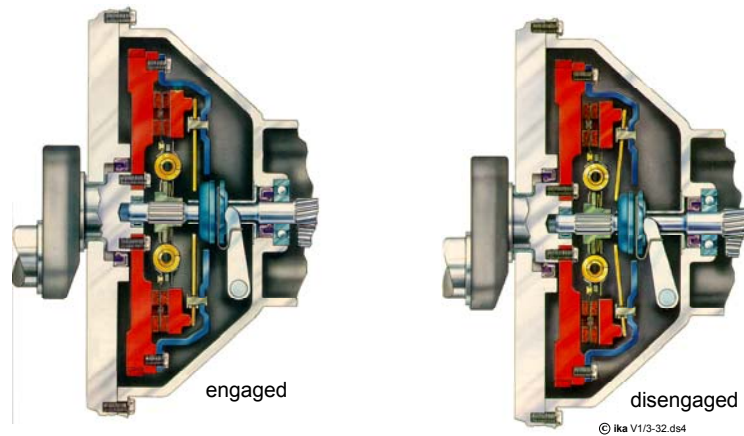


Fig. 3-32: Diaphragm spring clutch in inside and outside condition (Source: mot - Spezial, Antrieb)

In case of a rigid design of the operating mechanism, a torque peak would result during clutch engagement. In order to prevent this, nearly all clutch disks today are equipped with a lining springs. Furthermore the axial springs between the clutch linings lead to a uniform load transfer and thus to a lower lining wear.

Fig. 3-33 shows the air gap, axial force, and de-clutching force dependent on the clutch displacement for a declutching process.

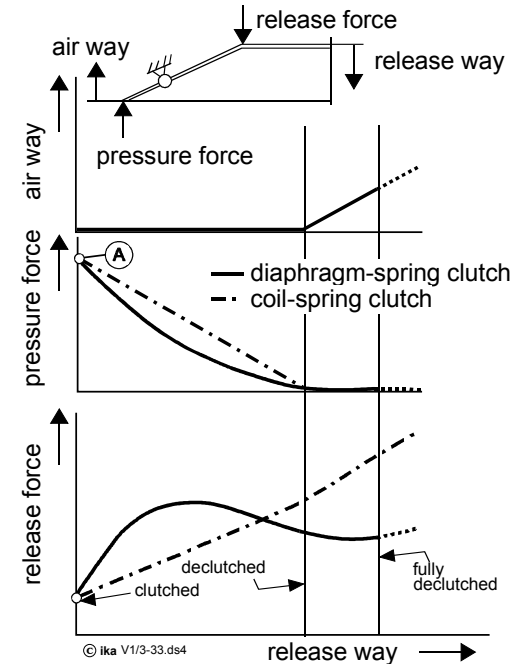


Fig. 3-33: Air gap, axial force and de-clutching force over clutch displacement for mechanical clutches

Considering equal displacements up to the formation of an air gap, the coil and diaphragm spring clutch have a similar characteristic of the axial force over the clutch displacement.

In the case of the diaphragm spring, the de-clutching force reaches a maximum. Thus the applied pedal force for coupling is limited and in the decoupled condition smaller than the steadily rising force of the coil spring clutch over clutch displacement. For complete de-clutching, an additional displacement between the linings (air gap) has to be adjusted. In the case of the coil spring clutch, it leads to a rise in the pedal force whereas the pedal force decreases in the case of the diaphragm spring clutch.

In addition, the different spring characteristics have an effect in determining which axial forces become effective in clutch wear. Fig. 3-34 shows the force-displacement-diagram on the friction pairs of coil and diaphragm springs over the lining wear.

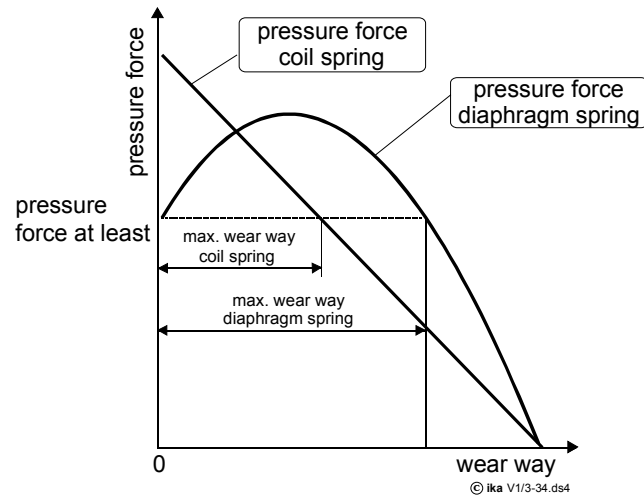


Fig. 3-34: Force-displacement-diagram of coil and diaphragm spring

While in the coil spring clutch, the axial force drops linearly over the wear displacement, the diaphragm spring's axial force shows a distinct maximum in the operating area. At an approximately similar maximum axial force, this curved characteristic of the diaphragm spring results in a significantly larger wear displacement compared to the coil spring without falling short of a pre-determined minimum axial force.

Handling the clutch today still demands a certain experience from the driver and does not exclude misuse. As a result, a number of attempts have been made to automate the clutch control. Fig. 3-35 shows the functional principle of an automatic clutch. The clutch itself is built conventionally. It is driven by an electronic system over a control motor. This electronic system records the engine and transmission speeds using corresponding pickups. It recognizes when it is shifted into which gear and closes the clutch more or less quickly depending on the driving style. Shifting gears takes place manually. Such automatic clutches are finding wider application in trucks.

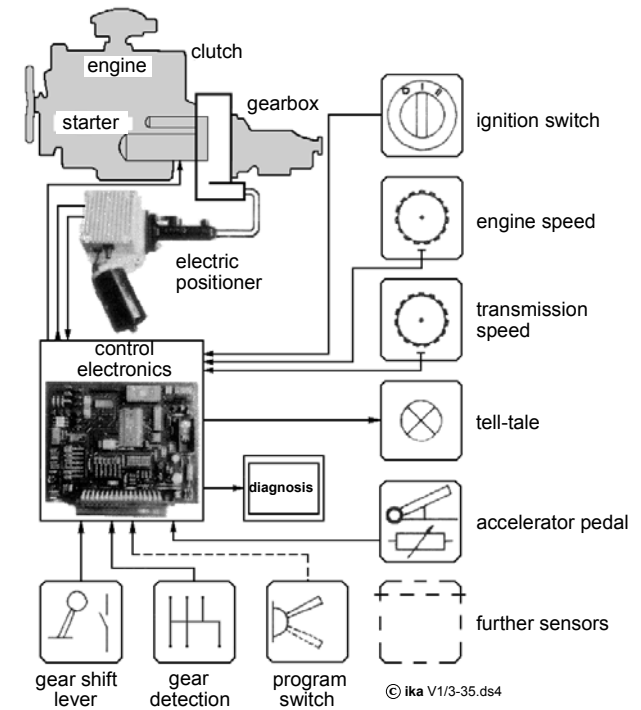


Fig. 3-35: Functional principle of an automatic clutch (Fichtel & Sachs)

Apart from speed conversion, the clutch also plays an important role in the absorption of noise and vibration from the engine and drivetrain. Chapter 3.7 deals with this in more detail.

3.3.2 Hydrodynamic Clutch

The hydrodynamic clutch, also called *Foettinger* clutch, generally consists of a pump wheel connected to the engine and a clutch-ended turbine that is equipped with uniform, radial vanes. A torque is transferred between the pump wheel and the turbine over a fluid which is accelerated by the pump and decelerated in the turbine (Coriolis effect). The hydrodynamic clutch can partially dissipate thermal losses over the working fluid. Consequently the speed conversion is not limited over time. Fig. 3-36 shows the hydrodynamic clutch in a design without external cooling.

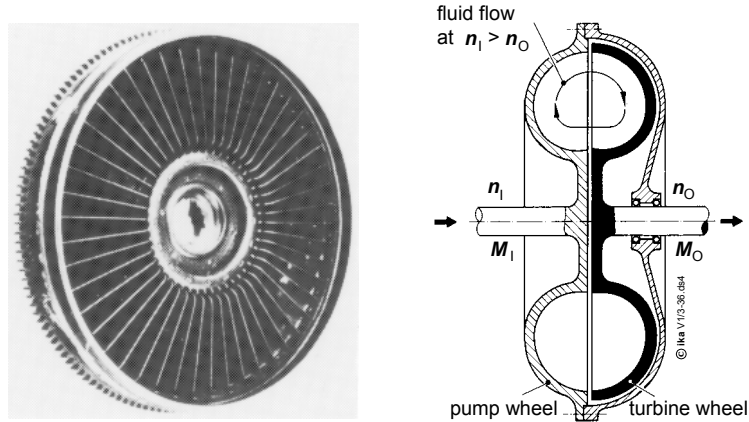


Fig. 3-36: Hydrodynamic clutch

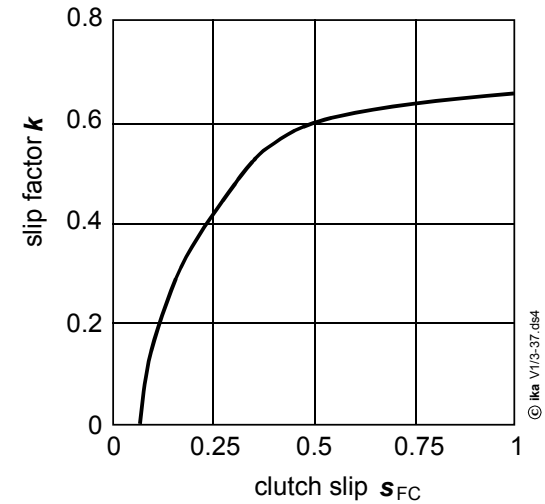
The *Foettinger* clutch can transfer a torque only in the presence of slip. When the output speed is larger than the input speed, the pump and turbine wheel reverse their functions.

The transferred torque can be calculated according to *Foettinger's* law using the following formula:

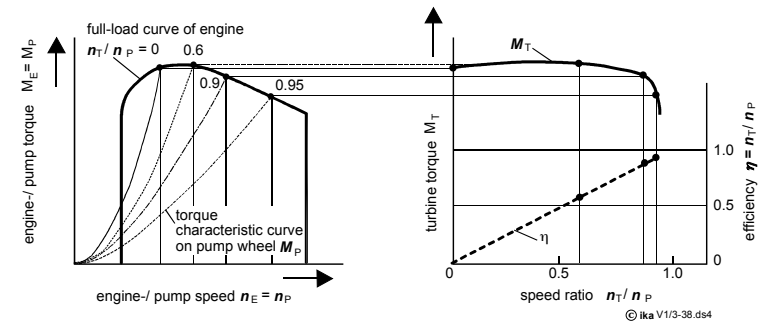
$$M_{FC} = k \cdot \rho \cdot n_p^2 \cdot D^5 \quad \text{Eq. 3-18}$$

where: k = slip factor
 ρ = density of fluid (oil $\approx 870 \text{ kg/m}^3$)
 n_p = pump speed
 D = clutch diameter

The slip factor k depends on the design and the clutch slip. Fig. 3-37 shows an exemplary characteristic.

Fig. 3-37: Slip factor of a *Foettinger* clutch (torque-slip-characteristic)

The interaction between a combustion engine and a *Foettinger* clutch is illustrated in Fig. 3-38.

Fig. 3-38: Interaction between combustion engine and *Foettinger* clutch (Source: Mitschke: „Dynamik der Kraftfahrzeuge“)

The pump characteristic represents the torque delivered by the pump wheel. Since the torque at the pump is determined by the combustion engine, the specific operating point can be obtained by the intersection of the pump characteristic and engine torque. The speed n_p is hence defined. According to the instantaneous characteristic of driving resistance, the torque at the pump can be converted into a

driving velocity and hence into a turbine speed. Since the pump characteristic changes depending on the pressure ratio between the turbine and pump, the operating point also moves on the curve of the engine torque.

When the vehicle is stationary, the clutch transfers a moment that results from the point of intersection of pump characteristic "0" and the engine characteristic. When the vehicle accelerates, the turbine or output speed n_T increases. As a result, the k -factor changes resulting in a change in the slope of the pump characteristic. The new point of intersection provides the new clutch torque. This process takes place continuously until the coupling is achieved and n_T/n_P is constant.

Moreover, the slip of a *Foettinger* clutch cannot become zero since otherwise no torque would then be transferred. The maximum efficiency is 98 %. With the aid of mechanical converter lockup clutches efficiencies up to 100 % can be achieved.

3.3.3 Visco-Hydraulic Clutch

The visco-hydraulic clutch, generally called "visco clutch", holds special significance among the clutch designs discussed here. So far it has not been applied in vehicles as a starting or shifting clutch but as a self-regulating axle or transfer case (Chapter 3.5), i.e., as a locking device for these components. If used as a main clutch in its present design, slip would have to occur permanently in order to transfer a torque, which would be disadvantageous.

Fig. 3-39 shows the fundamental design of a visco-hydraulic clutch, which basically consists of a number of axially-displaceable lamellae. Inner and outer lamellae are each respectively connected to a clutch outlet. The clearances are filled with a highly viscous liquid (generally silicon oil). By the visco-hydraulic principle, torque transmission takes place as a result of the fluid dynamic forces acting between the individual lamellae.

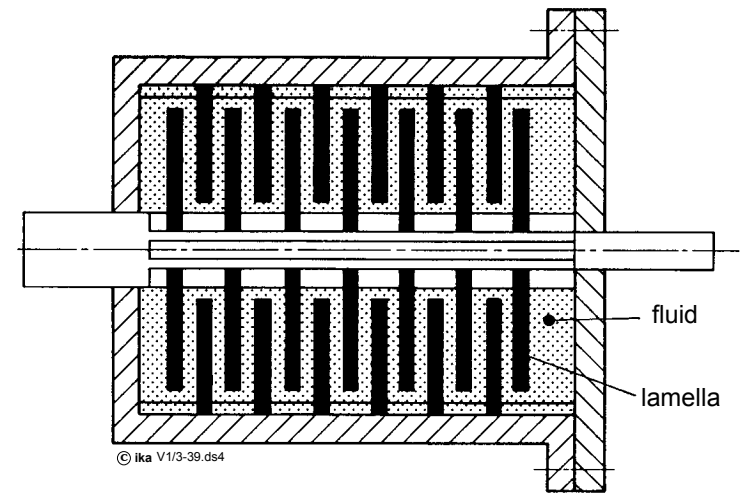


Fig. 3-39: Fundamental design of a visco-hydraulic clutch

The tensile stresses resulting from the shearing of the fluid film, linearly depend on the dynamic viscosity η_{dyn} and the difference in velocity relative to the film thickness according to Newton's Law of Tensile Stresses.

When applied to the visco-hydraulic clutch, the transferred torque is given by:

$$M_{VC_{id}} = z_{VC} \cdot \eta_{dyn} \cdot \frac{\Delta\omega}{c} \cdot \frac{1}{2} \cdot \pi \cdot (r_o^4 - r_i^4) \quad \text{Eq. 3-19}$$

where z_{VC} = number of active areas
 $\Delta\omega$ = differential velocity between outer and inner lamellae
 r_o = outer radius
 r_i = inner radius
 c = clearance

In a visco-hydraulic clutch, this approach for laminar flow is generally applicable to smooth lamellae since, the critical Reynold's Number can be hardly be achieved as a consequence of high dynamic viscosity.

By the appropriate selection of the following parameters:

- silicone oil
- degree of filling

- clearance
- lamella shapes (Fig. 3-40)

the torque characteristic of a visco-hydraulic clutch can be varied in a within a wide range from an almost linear to an extremely degressive course over the differential speed.

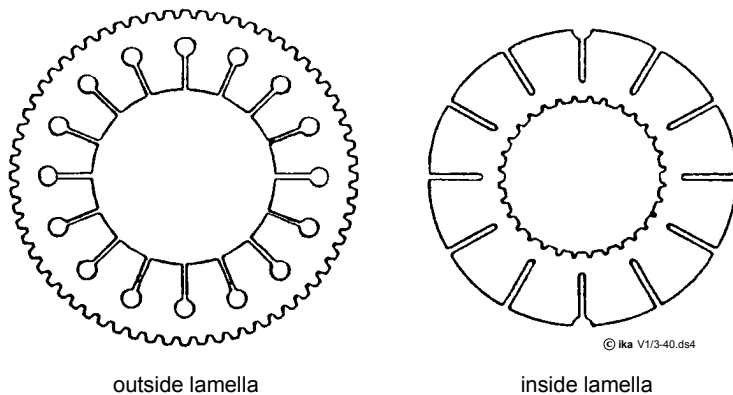


Fig. 3-40: Clutch lamellae of a visco-hydraulic clutch

Since the physical processes occurring in a visco-hydraulic clutch cannot be accurately determined when slit lamellae are used, the following general equation proved in practice is used to mathematically describe a degressive clutch characteristic:

$$M_{VC} = M_0 + j \cdot \Delta n^q \quad \text{Eq. 3-20}$$

where: M_0, j, q = clutch-specific parameters

At high differential speeds and longer slip phases, the temperature and pressure inside the clutch sharply increase. As a consequence, the transferable torque also rises steeply (Hump effect). Among other things, this steep rise results from the mechanical contact between certain clutch lamellae. As a consequence of the resulting friction, the transferable torque increases many fold over the transferable visco-hydraulic torque. This constructionally influenced torque increase acts a self-protection against over-heating. If the transferable torque is insufficient, the lamellae generally seize. An increased in the heat-input to the clutch overloads the sealing washers.

Fig. 3-41 illustrates the characteristic of torque, pressure, and temperature at a constant speed difference.

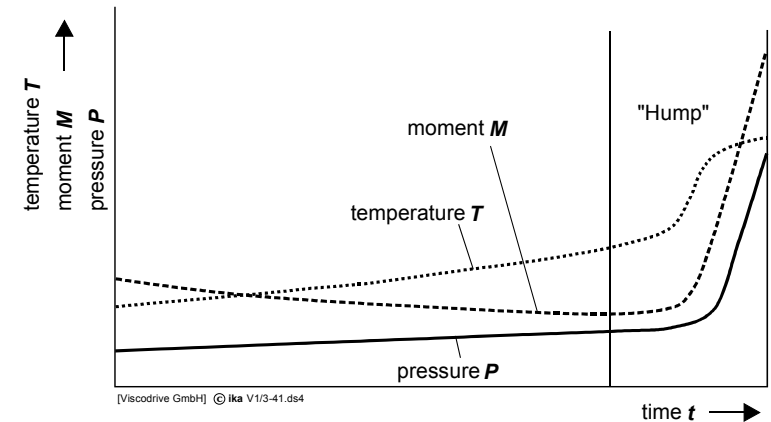


Fig. 3.41: Hump effect of visco-hydraulic clutches (GKN Viscodrive GmbH)

3.4 Torque Converters (Transmission)

While clutches only convert the speed, the output and input torques however remain the same. Torque converters (transmissions) are capable of additionally varying the output torque against the input torque. This way, the engine characteristic can be approximated to the desired performance hyperbola $P = \text{const.}$ (Fig. 3-21 shows the ideal characteristic, in which the desired characteristic curves for torque and power are included) (Fig. 3-42).

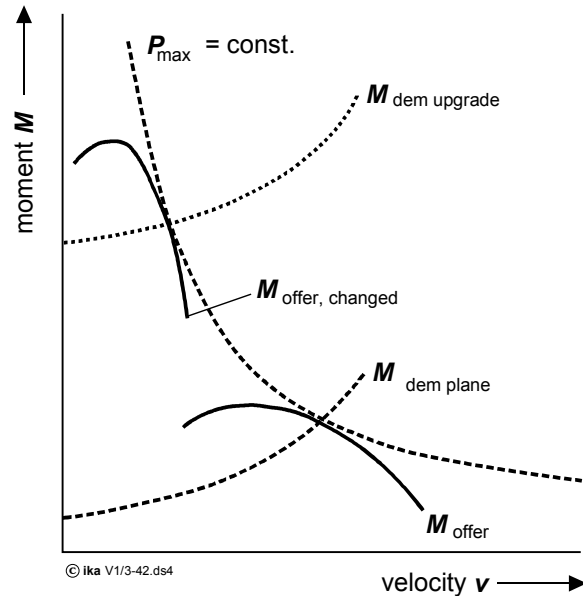


Fig. 3-42: Characteristic conversion by a torque converter

Fig. 3-43 shows the input and output parameters for a transmission. Torque conversion requires a torque which supports the differential moment between output and input torque. This is a characteristic of transmissions.

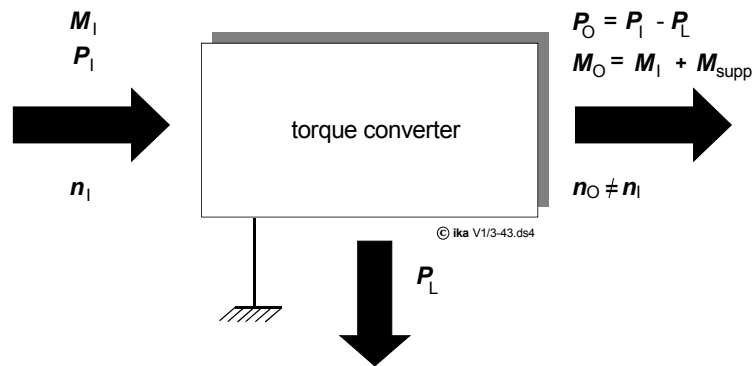


Fig. 3-43: Input- and output parameters of a torque converter

Similar to speed converters, in the case of torque converters, the overall power input which goes into the transmission is not available at the output. The definition of efficiency

$$\eta_T = \frac{P_O}{P_I} = \frac{M_O \cdot 2 \cdot \pi \cdot n_O}{M_I \cdot 2 \cdot \pi \cdot n_I} = \frac{M_O \cdot n_O}{M_I \cdot n_I} \quad \text{Eq. 3-21}$$

results in the equation for power dissipation:

$$P_{\text{diss}} = P_I - P_O = (1 - \eta_T) \cdot P_I \quad \text{Eq. 3-22}$$

Apart from the classical mechanical stepped transmission widely used in vehicles today, there are a number of alternative transmission concepts. Table 3-9 shows an overview of the different possibilities for torque conversion:

Stepped Transmission		Continuously Variable Transmission		
Mechanical		Mechanical	Hydraulic	Electric
Not shiftable under load	Shiftable under Load	Belt element Rolling body	Hydrostatic Hydrodynamic	
Countershaft Planetary	Countershaft Planetary			
Automatic Transmission High Power Transmission				

Table 3-9: Overview of different torque converters

3.4.1 Mechanical Stepped Transmissions

Torque conversion of mechanical transmissions is determined by the effective radii of the point of application of forces.

In case of mechanical stepped transmissions, a positively-engaged force or torque conversion normally takes place over gear wheels. The conversion characteristic hence depends on the gear ratio of the gears engaged together. As a result of the positively-engaged transmission, a conversion of the torque is only possible at definite levels. Each level represents a definite transmission ratio i which is the ratio of input speed to output speed:

$$i = \frac{n_l}{n_o} = \frac{M_o}{M_l \cdot \eta_T}$$

Eq. 3-

23

Fig. 3-44 shows the characteristic conversion using the example of a 4-speed transmission.

In this case four output maps emerge from the engine's delivery map. At the same time gaps remain between the transmission maps and the power hyperbola for $P = \text{const.}$ which cannot be driven.

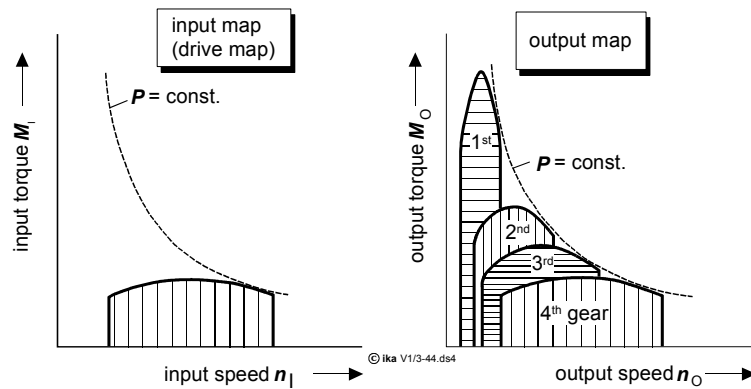


Fig. 3-44: Conversion of a torque map by a four-speed transmission

A stepped transmission should be designed in such a way that these transmission gaps have nearly no disturbing effect on driving. This can be achieved by increasing the number of gear levels or an appropriate choice of the individual ratios.

In today's vehicles the required conversion of characteristics between engine and driven wheels generally takes place in several stages. An axle ratio, which is a fixed ratio is connected behind the transmission which represents the torque converter with variable ratios. In commercial vehicles, the transmission with variable ratios can be further classified into the main auxiliary unit and rear-mounted or front-mounted auxiliary units. The axle multiplication can be followed by an additional wheel multiplication which is represented by a planetary transmission in the hub.

Fig. 3-45 illustrates in a diagram the arrangement of the individual components of the transmission for a passenger car and commercial vehicle.

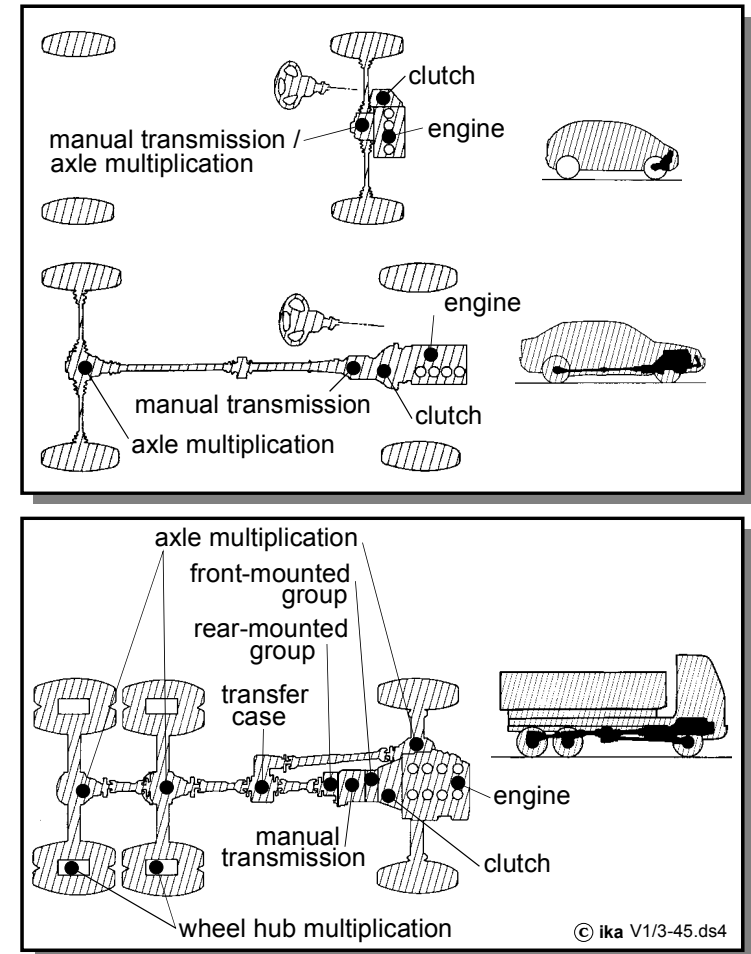


Fig. 3-45: Transmission components of a passenger car and commercial vehicle

Assuming the most common set-up consisting of a stepped transmission and drive axle transmission (front or rear axle), the relationship:

$$n_W = \frac{n_{\text{eng}}}{i_T \cdot i_{r(f)}}$$

Eq. 3-24

results for the engine speed n_{eng} and the speed of the driven wheels n_W .

In this case, i_T describes the manual transmission multiplication and $i_{r(f)}$ the axle multiplication of the driven rear or front axle.

Thus we receive for the vehicle speed v :

$$v = \frac{n_{\text{eng}} \cdot 2 \cdot \pi \cdot r_{\text{dyn}}}{i_T \cdot i_{r(f)}} \quad \text{Eq. 3-25}$$

Fig. 3-46 shows the linear dependence between vehicle velocity and engine speed for different stepped transmission ratios.

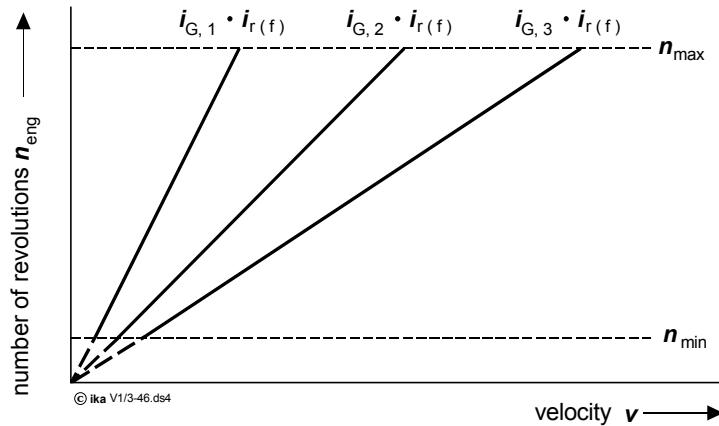


Fig. 3-46: Relationship between engine speed and driving velocity for different gear ratios

3.4.1.1 Transmission Layout

The size of a specific gear ratio depends on different criteria. The largest gear (the gear with the smallest ratio) is generally designed with respect to the maximum attainable vehicle speed (Chapter 4.1). The axle multiplication is determined in such a way that the gear which is statistically most frequently used in driving, can be designed as a direct gear ($i=1$). This implies that the input and output of the stepped transmission are directly connected with each other by a positively-engaged clutch provided that this is made possible by the transmission design. Due to the direct engagement, mechanical losses drop to a minimum.

The ratio of the smallest gear is determined by the maximum desired torque required at the driven axle, e.g. for the vehicle's climbing ability and the smallest possible driving speed with an engaged clutch.

In the selection of the intermediate gear ratios, the stability criterion has to be considered at first (Fig. 3-47).

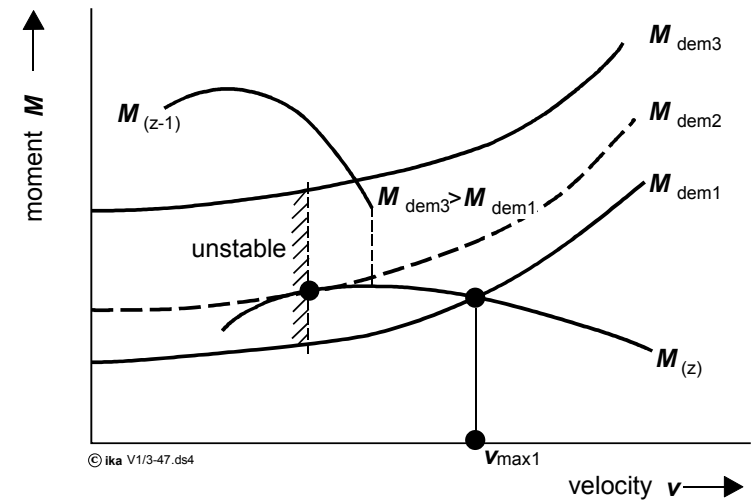


Fig. 3-47: Gear ratios for steady driving conditions

It has to be guaranteed here, that the transmission ratios are selected in such a way that in the limiting case, i.e., when the maximum torque on the engine characteristic line in the gear (z) is reached, it should be possible to shift to the preceding smaller gear ($z-1$). This should be possible without exceeding the permissible maximum speed n_{perm} of the engine.

For example, if a vehicles moves in the gear (z) with the maximum possible speed v_{max1} (point of intersection of the demand curve M_{dem1} with the supply curve $M_{(z)}$), a higher engine torque can be provided in order to cover the increasing torque demand, e.g. on a gradient, by decreasing the engine speed.

If the demanded torque exceeds the maximum transferable torque at the wheel, it should be possible to shift into the next smaller gear ($z-1$) before the unstable area at the contact point of the delivery characteristic $M_{(z)}$ with the demand characteristic M_{dem2} begins. In the unstable area, at a decreasing speed, the drop in speed or velocity is amplified by a simultaneously dropping torque supply.

The stability condition is thus called:

$$v_{Mmax(z)} \leq v_{nperm(z-1)} \quad \text{Eq. 3-26}$$

For the ratio of the multiplications i of two neighboring gears (z) and ($z-1$), the so-called step jump α_G , is given by:

$$\alpha_G = \frac{i_{(z-1)}}{i_{(z)}} \quad \left(= \frac{n_{(z-1)}}{n_{(z)}} \quad \text{für } v = \text{konst.} \right) \quad \text{Eq. 3-27}$$

The maximum possible step jump results from the stability condition:

$$\alpha_{Gmax} = \frac{n_{perm}}{n_{Mmax}} \quad \text{Eq. 3-28}$$

In case of engines that reach their maximum torque at low speeds, a large step jump and hence a low number of gears can be realized. In today's engines (reciprocating internal combustion engines) α_{Gmax} is within the range of 1.5 to 2. Assuming that the maximum speed is just slightly larger than the nominal speed, this value corresponds more or less to the speed elasticity (see Chapter 3.2.7.1). Conversely, it can be deduced from this, that engines with a high speed elasticity enable large step jumps and thus a smaller number of gears. It has to be considered, however, that the stability criterion is only insignificantly important in the case of highly motorized passenger cars. With commercial vehicles it is often desirable to choose considerably smaller gradations.

In order to calculate the ratios of intermediate gears there are two formal methods which are defined according to the type of layout:

- Geometric layout
- Progressive layout

- Geometric Layout

The geometric layout is characterised by a constant speed jump for shifting in all gears. The resulting speed-velocity-diagram is shown in Fig. 3-48.

The shifting speed n_u at which shifting into the next smaller gear results in the maximum speed n_{perm} being achieved, remains the same for all transmission ratios.

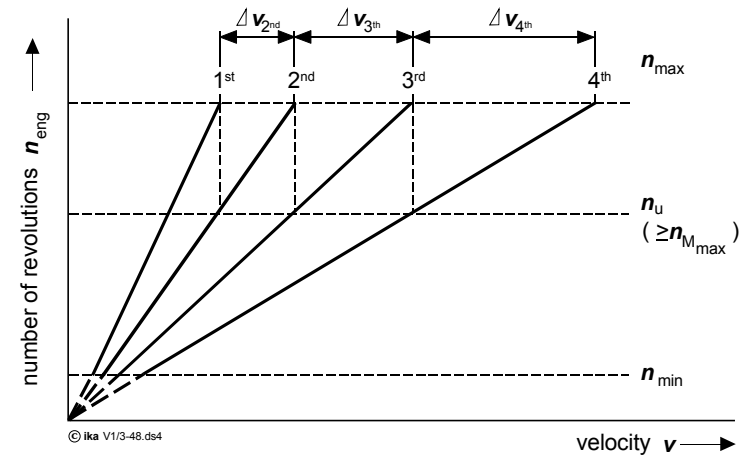


Fig. 3-48: Speed-velocity-diagram for geometric transmission stepping

From

$$\alpha_G = \frac{i_{(z-1)}}{i_{(z)}} = \frac{i_{(z-2)}}{i_{(z-1)}} = \frac{i_{(z-3)}}{i_{(z-2)}} = \dots \quad \text{Eq. 3-29}$$

and

$$i_{(z)} = \frac{n_{perm} \cdot 2 \cdot \pi \cdot r_{dyn}}{i_{r(f)} \cdot v_{perm(z)}} \quad \text{Eq. 3-30}$$

results

$$\alpha_G = \frac{v_{perm(z)}}{v_{perm(z-1)}} = \frac{v_{perm(z-1)}}{v_{perm(z-2)}} = \dots \quad \text{Eq. 3-31}$$

From Fig. 3-48 using the theorem of intersecting lines:

$$\frac{v_{perm(z)}}{v_{perm(z-1)}} = \frac{n_{perm}}{n_u} = \text{const.} \quad \text{Eq. 3-32}$$

The step jump of the geometrically stepped transmission is given by:

$$\alpha_G = \frac{n_{perm}}{n_u} = \text{const.} \quad (\leq \alpha_{Gmax}) \quad \text{Eq. 3-33}$$

The step jump α_G remains the same when shifting in all gears. In a transmission with z gear levels, its size can be determined by the total step jump α_{Gtot} , which is also called the setting range or the spread of the transmission.

$$\alpha_{Gtot} = \frac{i_1}{i_{(z)}} = \frac{i_1}{i_2} \cdot \frac{i_2}{i_3} \cdot \dots \cdot \frac{i_{(z-1)}}{i_{(z)}} = \alpha_G \cdot \alpha_G \cdot \dots \cdot \alpha_G = \alpha_G^{(z-1)} \quad \text{Eq. 3-34}$$

As a result, the step jump is given by:

$$\alpha_G = \sqrt[z-1]{\alpha_{Gtot}} = \sqrt[z-1]{\frac{i_1}{i_{(z)}}} \quad \text{Eq. 3-35}$$

Starting with the ratio of the largest gear $i_{(z)}$, the remaining ratios can be determined by the formation of a geometric series as follows:

$$i_{(z-1)} = i_{(z)} \cdot \alpha_G \quad \text{Eq. 3-36}$$

$$i_{(z-2)} = i_{(z-1)} \cdot \alpha_G = i_{(z)} \cdot \alpha_G^2 \quad \text{Eq. 3-37}$$

$$i_{(z-3)} = i_{(z-2)} \cdot \alpha_G = i_{(z)} \cdot \alpha_G^3 \quad \text{etc.} \quad \text{Eq. 3-38}$$

While the difference Δn between the permissible maximum speed n_{perm} and shift speed n_u remains constant, the difference of maximum velocities of successive gear levels Δv gets larger with a decreasing ratio or increasing gear number:

$$\Delta v = \frac{\Delta n \cdot 2 \cdot \pi \cdot r_{dyn}}{i_{r(f)} \cdot i_G} \quad \text{Eq. 3-39}$$

Fig. 3-49 shows the torque delivery map of a reciprocating engine, coupled with a geometrically stepped four-speed transmission. The gaps between the torque transfer of two successive gears and the power hyperbola for maximum engine output which cannot be driven as operating points, are of the same size for all gear ratios.

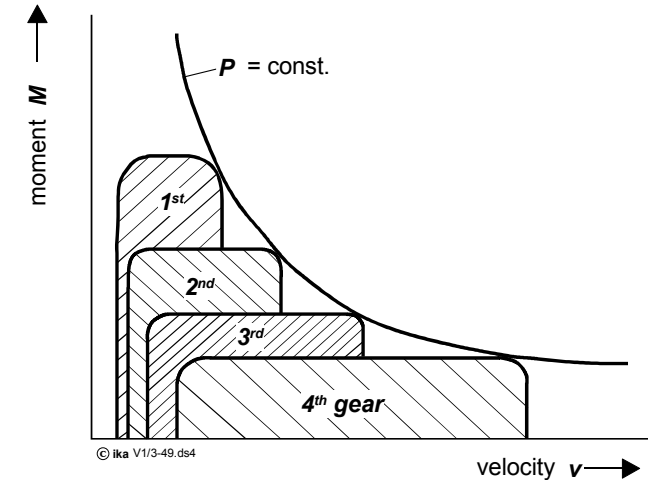


Fig. 3-49: Torque offer in geometric transmission layout

The geometric transmission is widely used in commercial vehicles. Due to the comparatively unfavorable weight-to-power ratio, equal velocity ranges have to be considered for all gears for the transmission layout.

- Progressive Layout

In the case of passenger cars, where engines produce their maximum torques at relatively larger engine speeds, a geometric transmission layout would lead to

- An unacceptably high number of gears which would be detrimental to driving comfort
- An unacceptably low overall step jump, resulting in too small a starting torque or
- A reduced maximum speed.

A progressive layout is common in the transmissions of today's passenger cars. The associated speed-velocity-diagram is shown in Fig. 3-50.

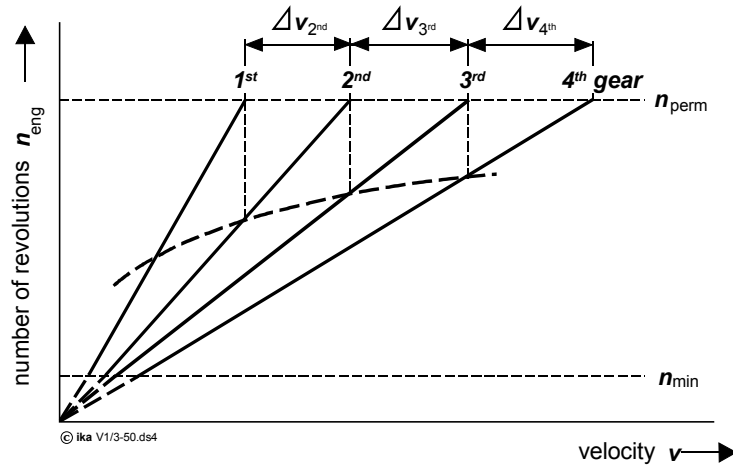


Fig. 3-50: Speed-velocity diagram for progressive transmission layout

A characteristic for the progressive transmission layout is the speed margin Δn between maximum speed and associated shifting speed that increases with a rising gear. This takes place with equally small differences in the maximum velocity change Δv_i between successive gears.

Hence in comparison with the geometric layout, gaps in the maps are reduced in the upper velocity range and enlarged in the lower velocity ranges (Fig. 3-51).

In the lower velocity range, larger transmission gaps can be tolerated in passenger cars due to the high available surplus power and low **power to mass** ratio as a result of small road resistances.

The step jump of a progressively stepped transmission with z gears is formed by the factors α_{G1} and α_{G2} :

$$\alpha_G = \alpha_{G1} \cdot \alpha_{G2}^n \quad \text{Eq. 3-40}$$

where: $\alpha_{G2} > 1$; $n = 0, 1, 2, 3, \dots, (z-2)$

α_{G1} and α_{G2} are constants while the exponent n depends on the particular step.

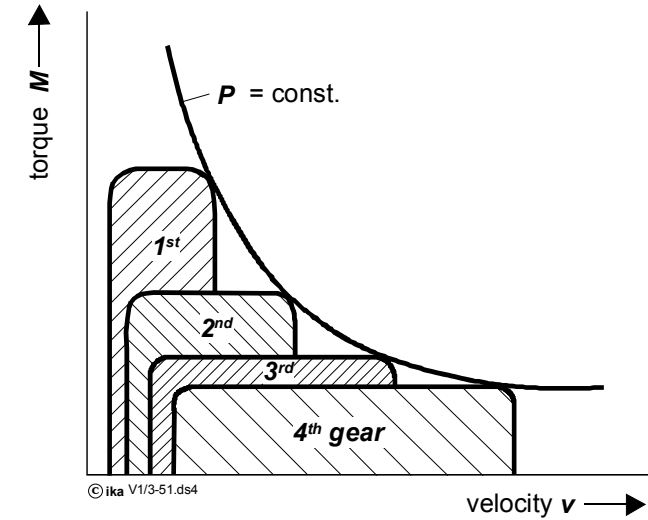


Fig. 3-51: Torque offer at progressive transmission layout

For example, for a progressively designed five-speed-transmission the following individual step jumps result:

Gear Step jump

$$5 \rightarrow 4 \quad \alpha_{G(z) \rightarrow (z-1)} = \frac{i_{(z-1)}}{i_{(z)}} = \alpha_{G1} \cdot \alpha_{G2}^0 \quad \text{Eq. 3-41a}$$

$$4 \rightarrow 3 \quad \alpha_{G(z-1) \rightarrow (z-2)} = \frac{i_{(z-2)}}{i_{(z-1)}} = \alpha_{G1} \cdot \alpha_{G2}^1 \quad \text{Eq. 3-41b}$$

$$3 \rightarrow 2 \quad \alpha_{G(z-2) \rightarrow (z-3)} = \frac{i_{(z-3)}}{i_{(z-2)}} = \alpha_{G1} \cdot \alpha_{G2}^2 \quad \text{Eq. 3-41c}$$

$$2 \rightarrow 1 \quad \alpha_{G(z-3) \rightarrow (z-4)} = \frac{i_{(z-4)}}{i_{(z-3)}} = \alpha_{G1} \cdot \alpha_{G2}^3 \quad \text{Eq. 3-41d}$$

This results in the following ratios for individual gears:

$$i_{(z-1)} = i_{(z)} \cdot \alpha_{G1} \cdot \alpha_{G2}^0 = i_{(z)} \cdot \alpha_{G1} \quad \text{Eq. 3-42a}$$

$$i_{(z-2)} = \frac{i_{(z-1)} \cdot \alpha_{G1} \cdot \alpha_{G2}^1}{i_{(z)} \cdot \alpha_{G1}^2 \cdot \alpha_{G2}} = \quad \text{Eq. 3-42b}$$

$$i_{(z-3)} = \frac{i_{(z-2)} \cdot \alpha_{G1} \cdot \alpha_{G2}^2}{i_{(z)} \cdot \alpha_{G1}^3 \cdot \alpha_{G2}^3} = \quad \text{Eq. 3-42c}$$

$$i_{(z-4)} = \frac{i_{(z-3)} \cdot \alpha_{G1} \cdot \alpha_{G2}^3}{i_{(z)} \cdot \alpha_{G1}^4 \cdot \alpha_{G2}^6} = \quad \text{Eq. 3-42d}$$

This corresponds to the general formula:

$$i_{(x)} = i_{(z)} \cdot \alpha_{G1}^{(z-x)} \cdot \alpha_{G2}^{0.5 \cdot (z-x) \cdot (z-x-1)} \quad \text{Eq. 3-43}$$

where x = gear number

z = number of gears

The overall step jump is given by:

$$\alpha_{Gtot} = \frac{i_1}{i_{(z)}} = \alpha_{G1}^{(z-1)} \cdot \alpha_{G2}^{0.5 \cdot (z-1) \cdot (z-2)} \quad \text{Eq. 3-44}$$

3.4.1.2 Transmission Designs

Mechanical stepped transmissions are distinguished into two designs:

- Dual shaft transmission
- Epicyclic transmission (planetary gearbox)
- Dual shaft transmission

The dual shaft transmission, also called countershaft transmission, is the generic term applicable to all conventional stepped transmissions. In this case, the transfer of torque in the non-direct gear levels takes place over externally toothed spur gear pairs mounted on two parallel shafts.

The spur gear pairs for the forward gears are permanently meshed with each other. They consist of a fixed gear wheel connected to a shaft and a free gear wheel. The latter is positioned on the second shaft and can freely rotate but cannot be axially displaced. Positively-engaging clutch elements are used to engage the freely rotating gears to the shaft, as per demand.

Dual shaft transmissions can be classified according to the position of the transmission input and output shafts into:

- Coaxial two-shaft transmission
- Deaxial two-shaft transmission
- Coaxial dual-shaft transmission

In the case of standard-driven vehicles (front engine with rear wheel drive), the dual-shaft mechanical stepped transmission in coaxial design is predominantly used. The coaxial position of input and output shaft which can normally be connected rigidly in the direct gear, is characteristic to this design. The power flow for the remaining gears is led over a countershaft (Fig. 3-52).

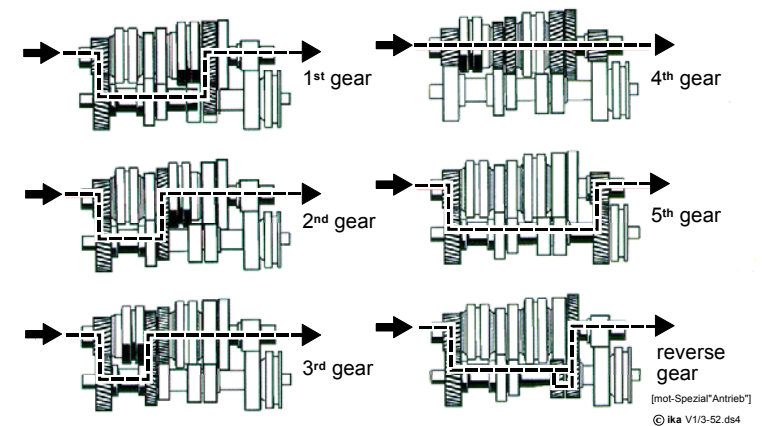


Fig. 3-52: Design and power flow in a coaxial dual-shaft transmission (Source: mot - Spezial, Antrieb)

As a rule, the coaxial countershaft transmission is directly coupled with the engine along with the clutch assembly. The output is led over the cardan shaft. This leads to

an unfavorable axle load distribution. This is the reason behind the transmission being located at the rear axle in some vehicles (transaxle design). The cardan shaft then rotates at the engine speed.

Fig. 3-53 shows an example for a coaxial five-speed transmission of a passenger car

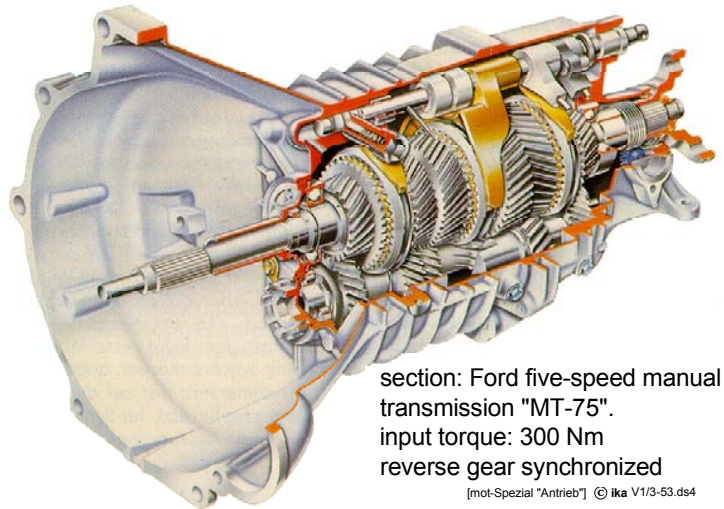


Fig. 3-53: Five-speed manual transmission (Ford)

- Deaxial dual-shaft transmission

In dual-shaft transmissions for vehicles with front-wheel drive or rear wheel driven vehicles with rear engines, the transmission input and output are axially offset and normally arranged on the same side of the transmission. Close to the output is the differential, which is integrated into the transmission housing. The power flow in dual-shaft transmissions in deaxial design is indicated in the functional diagram shown in Fig. 3-54. In each gear, power is transferred over a gear wheel pair. Such a transmission is advantageous for front wheel driven vehicles, however, since an axial offset between the crankshaft position and the position of the drive shaft must always be bridged.

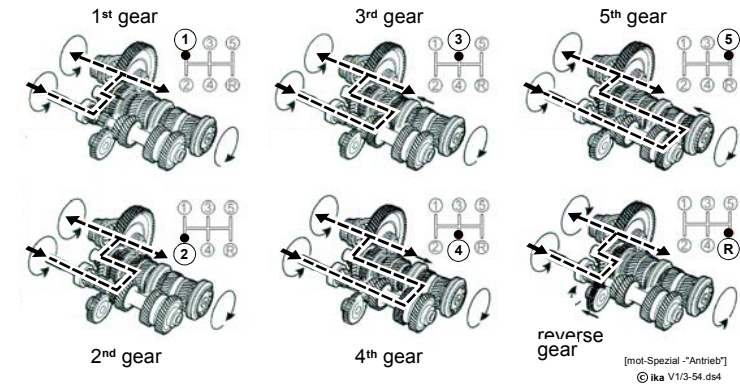


Fig. 3-54: Design and power flow of a deaxial two-shaft transmission (Source: mot - Spezial, Antrieb)

Fig. 3-55 shows an example of a deaxial dual-shaft transmission with front wheel drive and transverse-mounted engine.

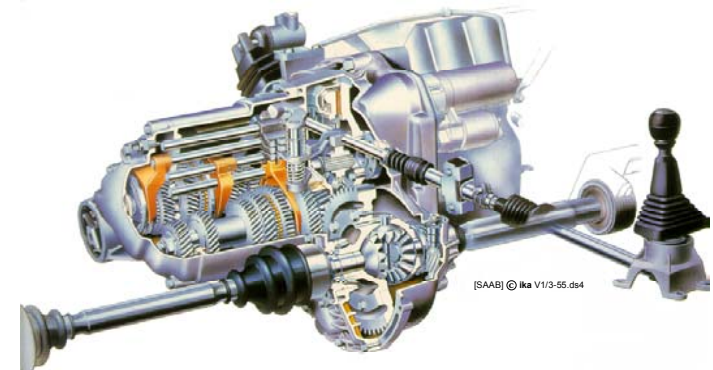


Fig. 3-55: Deaxial 5-speed-manual transmission (Saab)

- Gearshift

In dual-shaft transmissions, shifting into different gears takes place by means of an appropriate shifting clutch agent. In this case one differentiates between:

- Positively-engaged shifting clutch agents and
- Adherent shifting clutch agents

In manual transmissions, shifting gears takes place over gearshift rods. By displacing sleeves, they establish a positively-engaged link between gear wheel and shaft. A speed synchronization of shaft and gear wheel is required. The so called non-synchromesh transmissions were used for a long time, where the speed equality was achieved by double declutching on the part of the driver. The correct handling of non-synchromesh transmissions requires a certain skill, the result being that they are only occasionally used in trucks today. In order to achieve convenient operation at high shift speeds, low noise and low wear, the forward gears of stepped transmissions are synchronized by ensuring the speed equality of the rotating parts which have to be engaged during gear shifts, over frictional elements. Fig. 3-56 shows elements involved in speed synchronization.

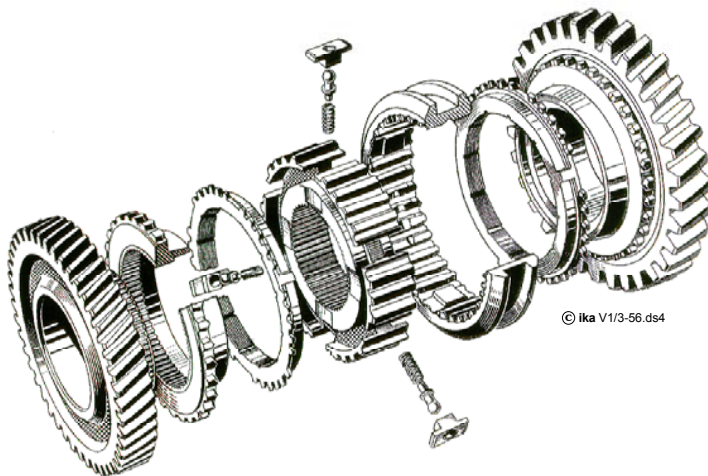


Fig. 3-56: Internal cone synchromesh (ZF)

Fig. 3-57 shows the functional principle of synchromesh.

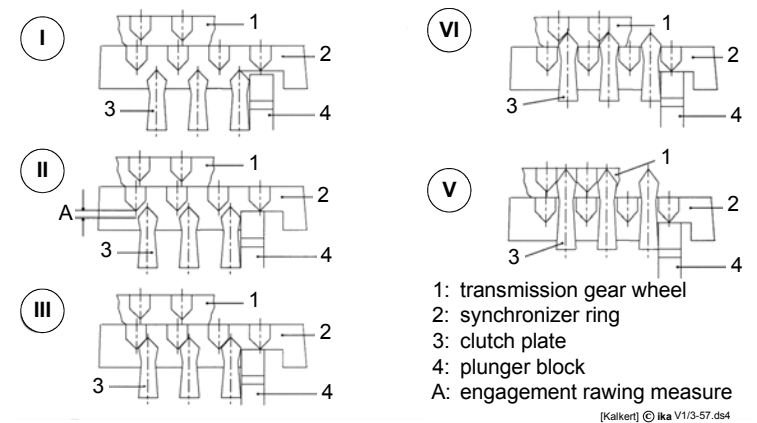


Fig. 3-57: Functional principle of Synchromesh (Kalkert)

Point I:

As a result of the gearshifting force, the clutch plate (3) along with the plunger block (4) moves out of the neutral position in the direction of the gear that has to be engaged. The gearshift play amounts to zero.

Point II:

The synchronizer ring and the gear wheel are loaded by the plunger blocks and the synchronizer spring (pre-synchronization).

The cone clutch goes into operation. This leads to the synchronizer ring (2) moving into the initial locking position. The engagement measure (A) is run through. So far no significant gearshift force can be measured.

Point III:

After having run through the engagement measure, the **roof slants** of synchronizer-ring and the interlocking clutch (sliding sleeve) come into contact with each other and the gearshift force increases quickly. The main synchronization starts off.

As a result of the gearshift force, the following torques develop at the synchronizer elements:

- The synchronizing torque between gear wheel cone and synchronizer ring,

- the plunger moment between the roof slants of synchronizer ring and sliding sleeve,
- the supporting torque between synchronizer ring and plunger block.

During the main synchronization, the synchronizing torque is larger than the plunger torque. Early shifting through is prevented. During this phase, there is no axial displacement of the clutch ring.

Point IV:

When the synchronizing speed is achieved, meaning that the clutch ring (3) and the gear wheel (1) are at the same speed, the clutch ring can rotate relative to the synchronizer ring (2) at the **roof slants**. The teeth of the clutch ring can now pass through the synchronizer ring.

Point V:

In the last phase, a positively-engaged connection with the synchronized gear is obtained.

The gearshift is terminated by closing the main clutch.

This process of synchronization however requires a finite time. In order to shorten this synchronization-specific shifting time, Porsche worked many years on the so called “double clutch transmission” which can be shifted under load. When driving in a particular gear, the next higher or lower gear can already be shifted. The real gearshift takes place by opening one clutch and closing another simultaneously. This means that the gearshift in this case is friction based. Fig. 3-58 shows the design of a double clutch transmission in deaxial design.

The second clutch is connected over a hollow shaft with the gear wheels of the gears 2 and 4, the first clutch over the inner shaft with the gear wheels of the gears 1, 3, and 5. In this way, two successive gears can be shifted simultaneously and in a positively-engaged manner. It is however not possible to shift between the gears on one shaft without a torque interruption, as is the case in regular dual-shaft transmissions.

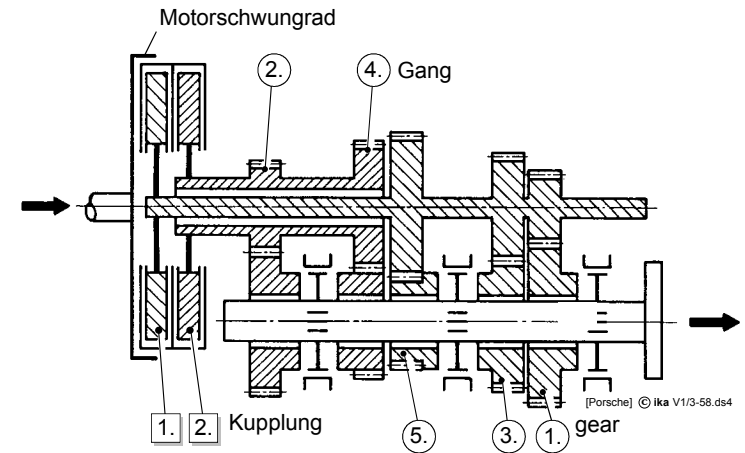


Fig. 3-58: 5-speed double clutch transmission (Porsche)

For sufficient speed and comfort in operating such transmissions, electronic regulation of clutch control and gearshift is necessary. The driver can in addition externally influence this arrangement as far as he wants.

- Epicyclic Transmission (Planetary Gearbox)

Epicyclic transmissions are characterized by at least three coaxial center shafts that permanently engaged with each other over gear wheels. Fig. 3-59 shows the different gear wheel positions in an epicyclic transmission with three center shafts.

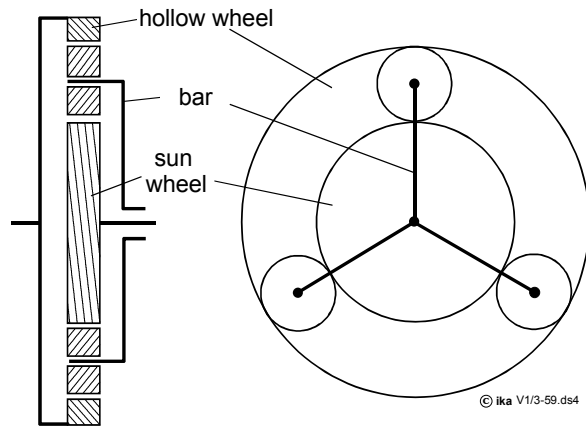


Fig. 3-59: Epicyclic transmission

Two of the shafts involved are connected with the central wheels, the sun wheel and the hollow wheel. The third one is coupled to a bar which is connected to the so-called planetary wheels which mesh with both central wheels. This design is chosen when possible as a result of its low constructional length, simplicity of the gear wheels and the point of action of loads which are in plane. This design of a circulation transmission is called spur-gear planetary transmission.

The characteristic parameter of an epicyclic transmission is the **nominal ratio** i_0 which is defined by the ratio of hollow wheel radius to sun wheel radius.

$$i_0 = \frac{r_H}{r_S} \quad \text{Eq. 3-45}$$

The speed ratios of a planetary train result from the basic equation for epicyclic transmissions:

$$\omega_B \cdot (1 + i_0) = \omega_H \cdot i_0 + \omega_S \quad \text{Eq. 3-46}$$

where:

- ω_B = angular velocity of the bar
- ω_H = angular velocity of the hollow wheel
- ω_S = angular velocity of the sun wheel
- i_0 = **nominal ratio**

In order to determine the torque ratios, Fig. 3-60 shows the forces on a planetary wheel that is in a constant state of motion.

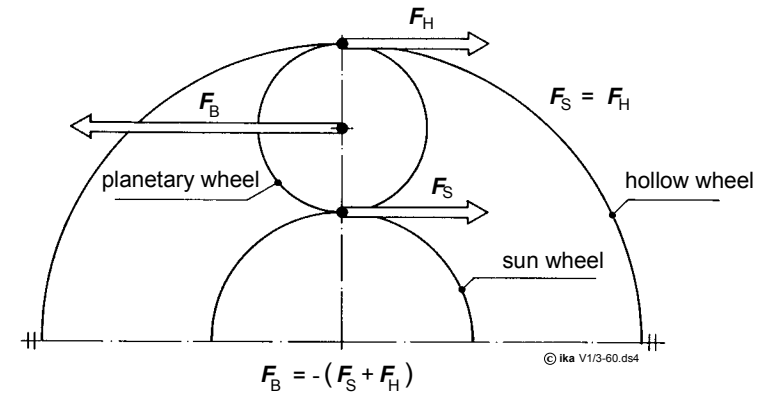


Fig. 3-60: Forces acting on a planetary wheel

The balance of forces and moments on a planetary wheel results in:

$$F_B = -(F_S + F_H) \quad \text{Eq. 3-47}$$

$$F_S = F_H \quad \text{Eq. 3-48}$$

where: $M_H = F_H \cdot r_H$ $M_S = F_S \cdot r_S$ $M_B = F_B \cdot r_B$

in magnitude it results in:

$$\frac{M_H}{M_S} = \frac{F_H \cdot r_H}{F_S \cdot r_S} = i_0 \quad \text{Eq. 3-49a}$$

Since: $r_B = \frac{r_S + r_H}{2}$

the remaining torque ratios can be calculated similarly:

$$\frac{M_B}{M_H} = \frac{1}{i_0} + 1 \quad \text{Eq. 3-49b}$$

$$\frac{M_B}{M_S} = 1 + i_0 \quad \text{Eq. 3-49c}$$

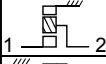
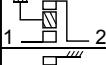
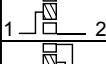
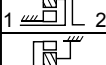
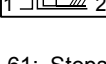
Thus, when losses in transferring moments are neglected, the ratio of torques on the three center shafts can be clearly determined only by geometric data of the planetary train.

For equilibrium $\Sigma M = 0$, we obtain

$$M_B = -(M_S + M_H) \quad \text{Eq. 3-50}$$

Hence the torques are either split up or added. In this case the direction of the bar moment M_B is opposite to the torques M_S and M_H acting in the same direction. When all shafts rotate in the same direction, the same is applicable to the power output of a planetary transmission.

In contrast to usual stepped transmissions, free-wheeling epicyclic transmissions have no free bearing forces. If a shaft is rigidly braked, it acts as a torque support. The other two shafts move relative to each other in a firm speed ratio, which can be determined from the basic equation. Based on coupling of the individual center shafts with the engine, output, and rigid transmission housing, a planetary train allows for six different ratios, two of which reverse the direction between the engine and output sides (Fig. 3-61).

scheme	positioning			multiplication	
	propulsion (1)	negative lift(2)	fixed ///	$i = n_1 / n_2$	range
	sun wheel	bar	hollow wheel	$1 + i_0$	$2 < i < \infty$
	sun wheel	hollow wheel	bar	$-i_0$	$-\infty < i < -1$
	bar	sun wheel	hollow wheel	$\frac{1}{1 + i_0}$	$0 < i < 0,5$
	bar	hollow wheel	sun wheel	$\frac{1}{1 + \frac{1}{i_0}}$	$0,5 < i < 1$
	hollow wheel	sun wheel	bar	$-\frac{1}{i_0}$	$-1 < i < 0$
	hollow wheel	bar	sun wheel	$1 + \frac{1}{i_0}$	$1 < i < 2$

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Fig. 3-61: Steps of a planetary train with theoretical ranges of ratios

In addition, a direct gear can be realized if the third shaft is linked with the engine or the output. The entire planetary train then rotates as a block. The ratios $i=2$, $i=0.5$ or $i=-1$ cannot be realized by a simple spur-gear planetary train. In order to achieve this, the number of teeth on the hollow gear has to correspond with that of the sun gear.

The situation can be set right using conical gear wheels or by assembled epicyclic transmissions.

The favorable, symmetrical design as well as a low construction volume and low weight with a simultaneously high maximum transferable torque, as a result of the distribution of forces over several meshings (several planetary wheels), led to a large number of applications of planetary transmissions in motor vehicles (automatic transmission, transfer cases in all-wheel drive systems, wheel-hub transmission).

In designs which involve several shiftable ratios, the planetary transmission can be shifted under load by employing adherent shift clutch agents. In realized transmissions, these shift elements consisting of multi-disk brakes and clutches as well as band brakes (Fig. 3-62) for the most part are hydraulically operated.

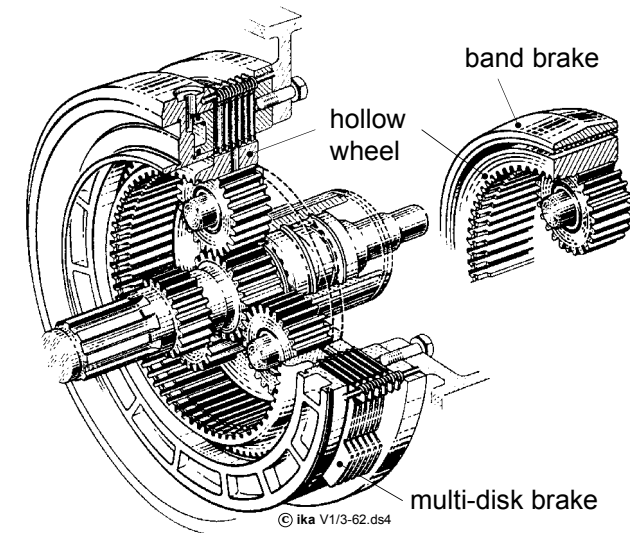


Fig. 3-62: Adherent shift clutch agents in epicyclic transmissions for braking the hollow wheel

In order to keep the number of shift clutch agents low in gearboxes with high number of gear ratios and to freely design the relationships of the individual gear ratios against each other, coupled planetary transmissions are used.

Fig. 3-63 shows the principle sketch of a 4-speed transmission with 2 planetary sets and the appropriate shift clutch agents.

Drive is transferred over one sun gear, or in the case of a blocked second planetary train, over two sun wheels. The output takes place over the planetary carrier of the rear planetary train. The particular gear ratios result from the repeated use of the basic equation, according to the number of planetary trains involved.

In automatic transmissions, often special designs of coupled planetary transmissions are used. Fig. 3-64 shows a Ravigneaux-transmission that distinguishes itself through a simple design of favorable gear ratios and a low constructional space. The design and shifting diagrams are shown in Fig. 3-65.

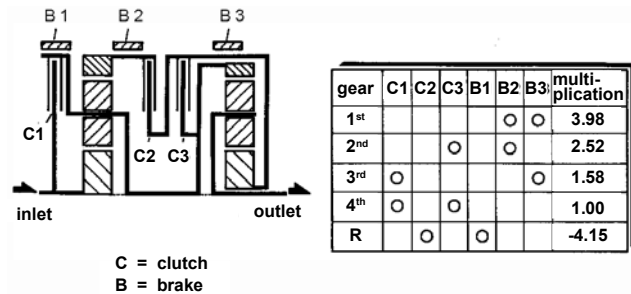


Fig. 3-63: Design and shifting diagram of a coupled planetary train (Daimler Benz)

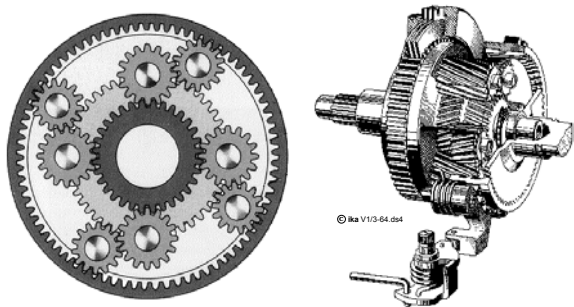


Fig. 3-64: Ravigneaux-transmission (Source: mot - Spezial, Antrieb)

Drive takes place over the sun wheels, while the output is derived over the hollow gear in the transmission shown in Fig. 3-64. The short and long planetary gears are connected by a common bar.

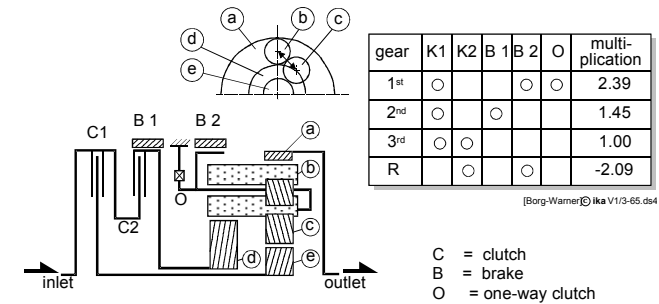


Fig. 3-65: Design and shifting diagram of the Ravigneaux-transmission (Borg Warner)

In non-shiftable designs with a fixed gear ratio, spur-gear planetary trains are used as wheel hub transmissions. In this way, the torque that acts on the drive shafts of the driven axles can be lowered and the radius of the shaft can be reduced. Despite the transmission being positioned in the wheel area, this leads to a reduction of the unsprung masses (Script „VT II“). Using the two-level ratio, distinctly reduced “short” axles can be realized, e.g. for construction site vehicles. Furthermore this leads to economy of space as a consequence of the smaller differential ring gear due to the smaller ratio required on the differential transmission. A disadvantage is the unavoidable reduction of efficiency due to the multi-stage reduction. The spur-gear transmission is only used in commercial vehicles (Fig. 3-66).

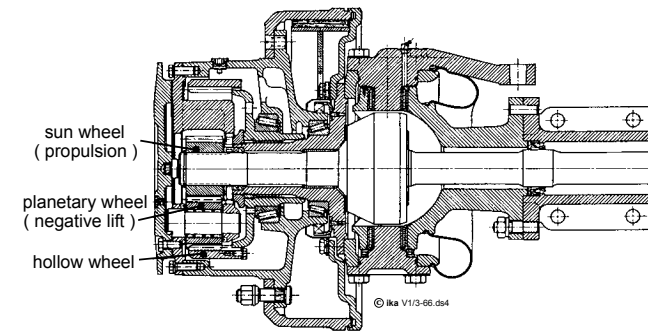


Fig. 3-66: Spur gear with planetary transmission

The planetary transmission is also used as a transfer case (Chapter 3.5) or add-on transmission (in performance-extended transmission designs). None of the center shafts of the epicyclic transmission are fixed. In the transfer case, the input torque is distributed into two outputs. In an add-on transmission, an input torque can be added on to an output torque. However, in this case, the epicyclic transmission is no longer a torque converter since the supporting torque is absent.

- Multi-group transmissions

In order to cope with the widely varying freight and loading conditions, a tighter transmission layout and thus a higher number of gears is required in commercial vehicles when compared to passenger cars. There are generally two possibilities in order to raise the number of gears:

- An additional gear wheel pair is intended for each additional gear in the main transmission,
- Front-mounted or rear-mounted groups are coupled to the main transmission.

The second possibility results in multi-group transmissions whose advantage lies in multiplying the number of gears without a corresponding increasing in the number of gear wheel pairs and shifting elements. This implies a saving in construction space, weight, and costs.

Fig. 3-67 shows a two-group transmission with a two-speed planetary train as rear-mounted group.

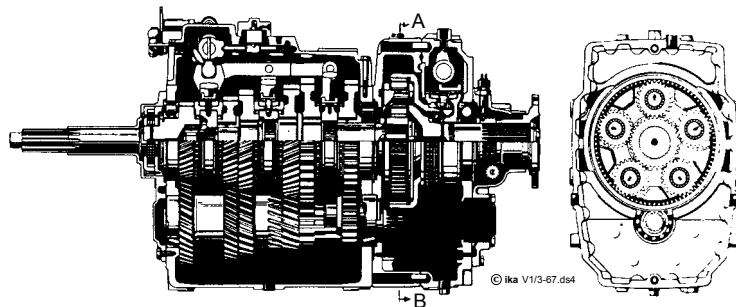


Fig. 3-67: 9-speed commercial vehicle transmission with a two-speed planetary train as rear-mounted group

In a three-group transmission, up to 16 different gear levels can be obtained from a four-speed main transmission at a relatively low construction expense. An example with 16 gears is shown in Fig. 3-68. The shifting diagram of this three-group transmission is presented in Fig. 3-69.

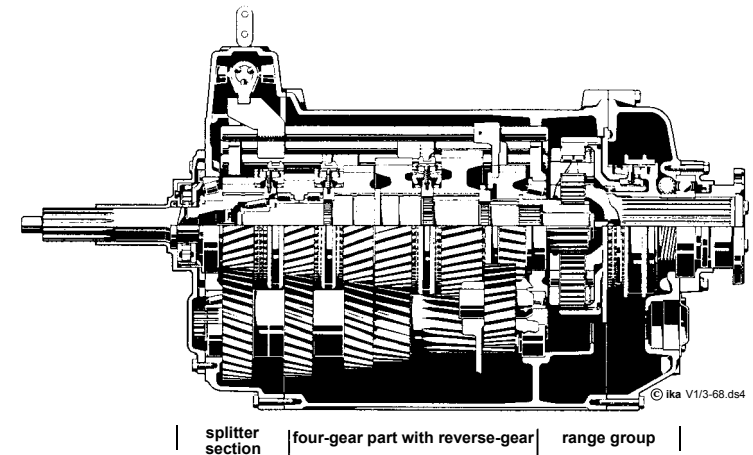


Fig. 3-68: Commercial vehicle three-group transmission (ZF)

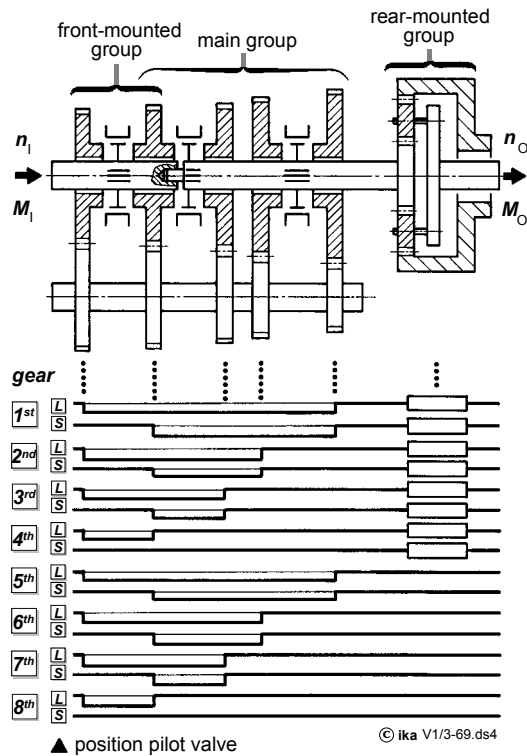


Fig. 3-69: Design and power flow of a three-group transmission

3.4.2 Mechanical Continuously Variable Transmissions

In contrast to the mechanical stepped transmissions so far used in motor vehicles, mechanical continuously variable transmissions, also called CVTs, have certain advantages based on their principle. The power characteristic is shown in Fig. 3-70. As a result of the continuously variable torque conversion the presented power curve (center) represents the converter characteristic curve. In order to produce a delivery map (right) which adjusts to the different driving conditions, the CVT only requires a demand characteristic line (left) from the engine. Except for the starting area, which requires an additional clutch, the (for a sufficient range of transmission ratios) the entire demand map can be covered in this way.

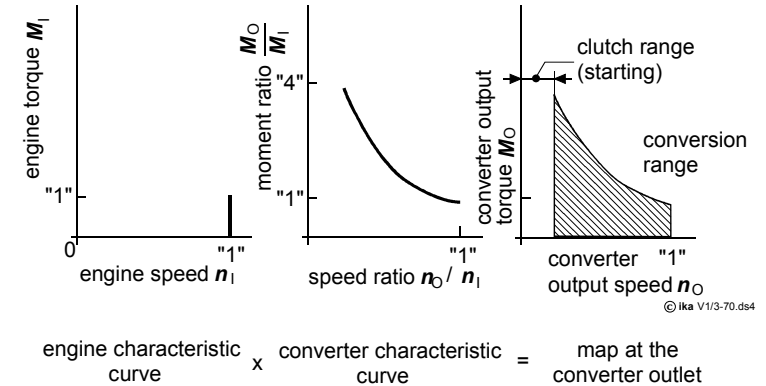


Fig. 3-70: Characteristic conversion by a CVT

Since the CVT allows for a free selection of the engine parameters torque and speed, the operation of the combustion engine can be optimized based on different criteria. When the optimum points on the engine map are combined over the entire performance area of the engine, a curve which is parameterized with respect to power, also called “control line”, results. Fig. 3-71 shows the control lines for the optimization criteria “noise”, “fuel consumption”, and “driving dynamics”.

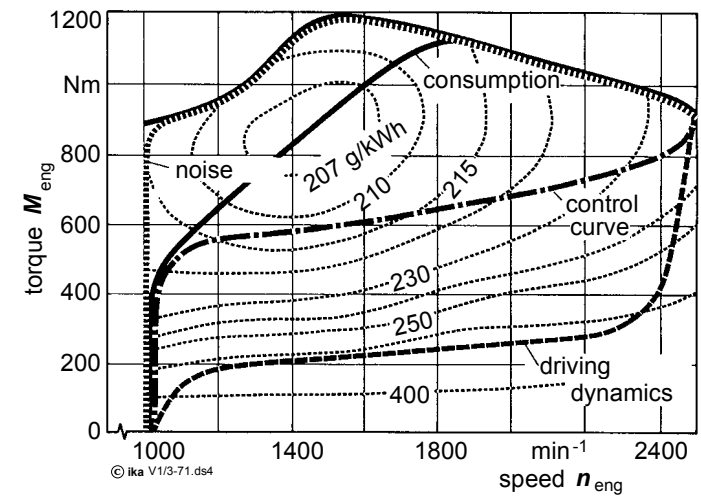


Fig. 3-71: Control lines for different optimization criteria

When realizing only one control line for transmission operation using a relatively simple mechanical control, the drawn-in curve “RL” also represents a good compromise. In the future, we can expect an increasing number of electronically regulated CVTs. In this case, control lines that correspond to the demand as well as the driver's needs can then be realized. This already implies that continuously variable transmissions can be exclusively realized as automatic transmissions.

In mechanical CVTs, the variation of the transmission ratio is achieved by varying the radius of the point of action of forces. The torque is transferred frictionally. Based on the type of torque transmission, mechanical CVTs can be classified into:

- U transmission,
- Pitch transmission.

a) U transmission

In U transmissions, force transmission is achieved over belts, bands, and chains which are frictionally clamped between disk pairs. By varying the rolling radii on the disks, the transmission ratio can be varied infinitely. As a result, the disk pairs along with the accompanying adjusting mechanism are also called variator. For the force transmission that is in the center of development work, so far different belt, band, and chain concepts have been proposed. Fig. 3-72 shows the current force transmission systems.

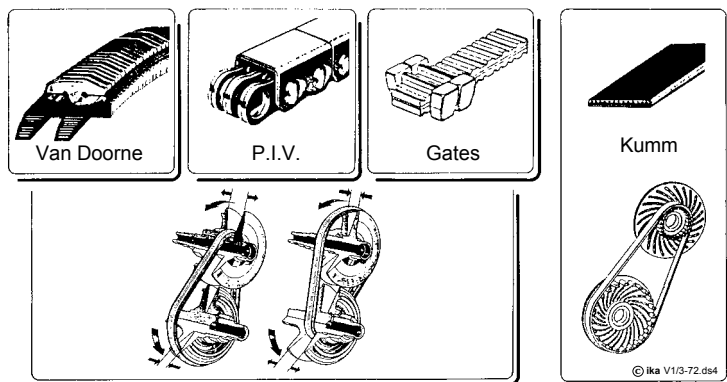


Fig. 3-72: Force transmission in modern U transmissions

In contrast to rubber belts used in variomatic transmissions in the 50s by Van Doorne, Holland, the transferable torque (up to 210 Nm) and efficiency were, in particular, significantly improved by changing over to a steel-link band in current CVTs. The transmission element, also called thrust link band due to the kind of loading, consists of approx. three hundred steel segments that are held together on both sides by steel band packages (band size 0.1 mm) piled up onto each other.

In contrast to the thrust link band, transmissions manufactured by PIV-Antriebe Werner Reimers KG, a chain loaded by tension is used for force transmission. The single chain's shackle packages are connected with each other by cradle compression parts. Force transmission takes place by friction in the contact areas of the cradle compression parts and disk wheels. U-shaped metallic clips that enclose the shackle bandage from the outside take over the chain guide. The whistle otherwise typical for chain drives which occurs when the link comes into contact with the disk wheels, can be more or less avoided by an appropriately designed chain.

In order to keep the centrifugal load on a U-band as low as possible, Gates Rubber, USA, developed a rubber belt called power-trac which is reinforced by kevlar fibers. In order to increase the transversal loading capacity, the rubber band is provided with an integrated metallic structure. An oil lubrication, as required in steel-steel friction pairings, is not necessary. This allows for an open transmission design associated with a lower weight.

Kumm Industries, USA, proposes the fourth transmission system shown in Fig. 3-72. In contrast to former U transmissions in which the force transmission element is clamped between two conical disk pairs, the Kumm CVT includes a rubber band which is also reinforced by kevlar (Gates-development), and runs at each disk end on studs. The studs can be shifted in spirally-shaped grooves. No lubrication is required here as well.

b) Pitch transmission

Since the first development of automobiles, pitch transmissions have been designed, manufactured, and tested over and over again, however, with no lasting success. Fig. 3-73 puts together some basic forms of pitch body force transmission.

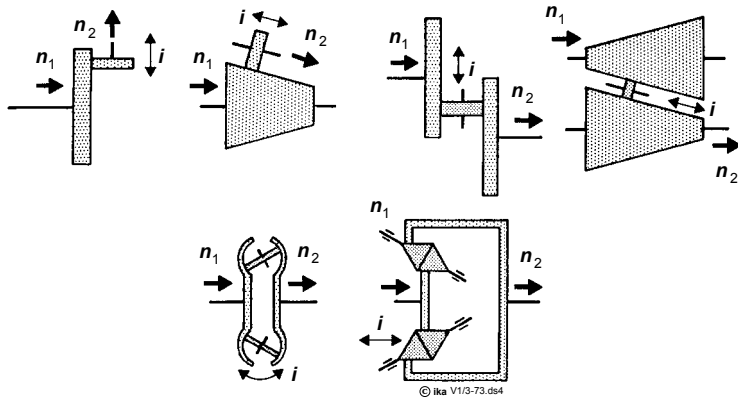


Fig. 3-73: Force transmission concepts in pitch transmissions

The continuously variable change of the transmission ratio takes place under load by varying the point of action of the force, meaning the effective pitch body radius. Apart from the friction coefficient, the pressure force of the pitch bodies determines the transferable force.

A renewed interest in pitch transmissions can be attributed to the development of new high-friction lubricants which enable nearly twice as large force transmission between the pitch bodies when compared to conventional lubricants.

For the application in motor vehicles, only those concepts which enable a firm position between input and output shaft can be used. This can be achieved by an intermediate link in the form of a rotationally symmetrical body.

3.4.3 Hydraulic Continuously Variable Transmissions

Transmissions in which power is transferred by an incompressible fluid can be classified, according to the type of function, into:

- Hydrodynamic and
- Hydrostatic systems.

By connecting in parallel a mechanical transmission unit, besides there are hi-tech variations.

a) Hydrodynamic transmissions

In hydrodynamic transmissions, torque transmission takes place according to the *Föttinger* principle using two rotating turbine blade wheels, the pump and the turbine wheel. In contrast to hydrodynamic clutches (Chapter 3.3), in addition the hydrodynamic transmission includes a stator as moment support that props up at the housing.

As a result:

$$M_T = M_P + M_S \quad \text{Eq. 3-51}$$

where: M_T = turbine wheel moment

M_P = pump wheel moment

M_S = stator moment

Fig. 3-74 shows the constructional details of a hydrodynamic transmission (Trilok converter) which is almost exclusively used today, with accompanying blade principle and flow conditions for $n_A/n_E = 0.7$.

The working fluid, which is generally oil, is accelerated by the pump wheel linked to the engine, transferred to the turbine wheel where it is slowed down. While doing so, it gives away its energy to the transmission output. An additional redirection more or less without delay leads to torque reinforcement. This reinforcement is the higher, the larger the speed difference is between pump and turbine. With $v = 0$, implying a firmly braked turbine, the torque conversion reaches its maximum value. With an increasing turbine speed, torque conversion drops almost linearly up to the torque ratio 1:1 (coupling point).

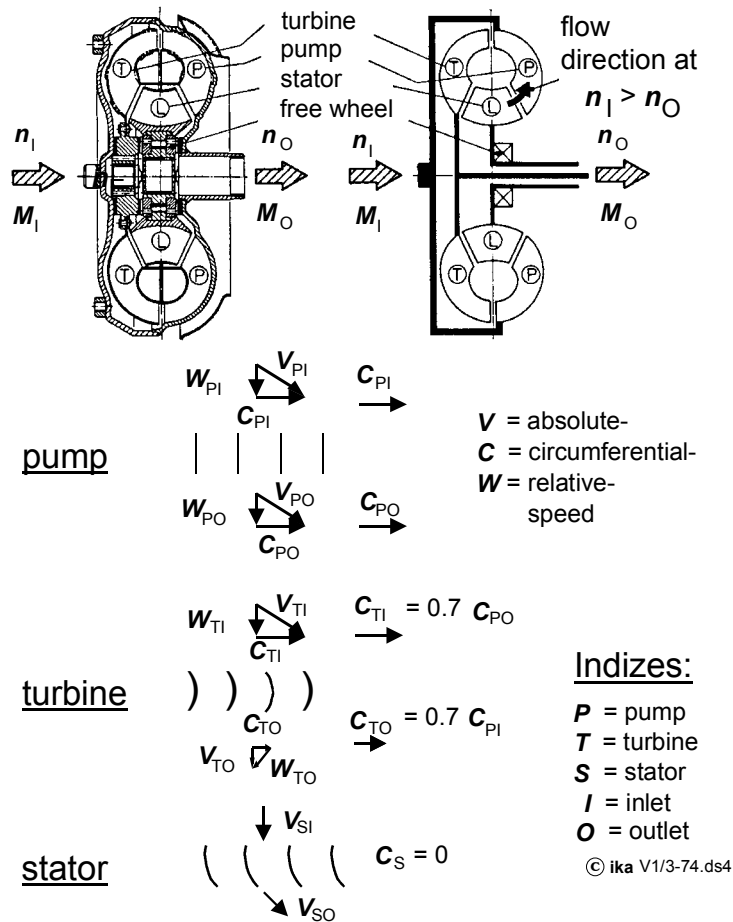


Fig. 3-74: Trilok converter with accompanying blade principle

In the example, at the indicated speed ratio, an optimal point of operation appears at approx. $n_A/n_E = 0.7$ without impact loss. If the speed ratio increases further, the stator is increasingly streamed from behind until at a speed ratio at approx. $n_A/n_E = 0.9$ (coupling point), the stator does not produce a deflection anymore, meaning that it does not absorb a reaction moment. In order to avoid this torque deterioration at a further increased speed ratio, the stator is connected with the transmission housing by a one-way clutch so that it can run along without moment transmission at speed ratios above the coupling point. In this area, the trilok converter works as a clutch.

Fig. 3-75 shows in an idealized way the torque- and efficiency course over the speed ratio.

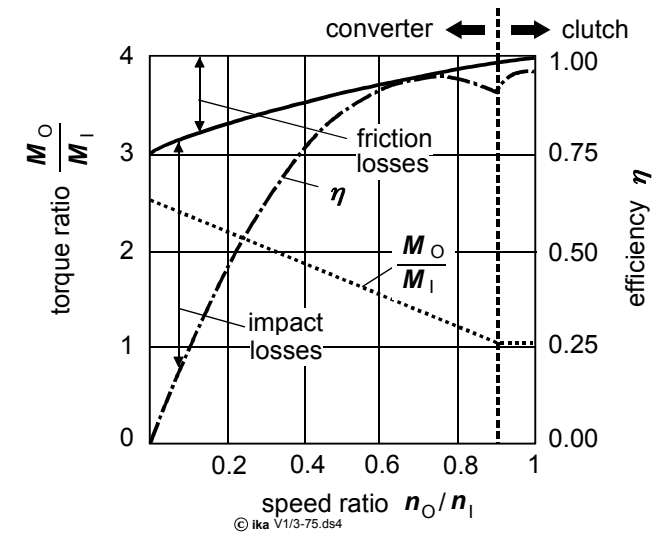


Fig. 3-75: Characteristic of a trilok converter

Moreover the picture shows the rough division of efficiency loss in impact (no correspondence between flow and blade direction) and friction losses (friction between fluid and walls).

The interaction between trilok converter and the combustion engine proceeds as follows:

Up to the coupling point, the flow ratios at the pump wheel input are independent of the ones at the transmission output because of the stator, which is at a standstill. The important k -factor (Equation 3-18) and thus the pump characteristic curve are constant. In this case, the pump characteristic curve, if possible, should be located within the area of optimum engine efficiency. Above the coupling point, when stator and turbine wheel rotate as a "common" turbine wheel, the same principles as in the hydrodynamic clutch are applicable. There, the pump characteristic curve moves dependent on the speed ratio n_A/n_E . We can see in Fig. 3-76 that only a part of the engine map can be used in this way. through the interaction of the engine torque and trilok converter characteristic, the output torque which is indicated in Fig. 3-76 finally results.

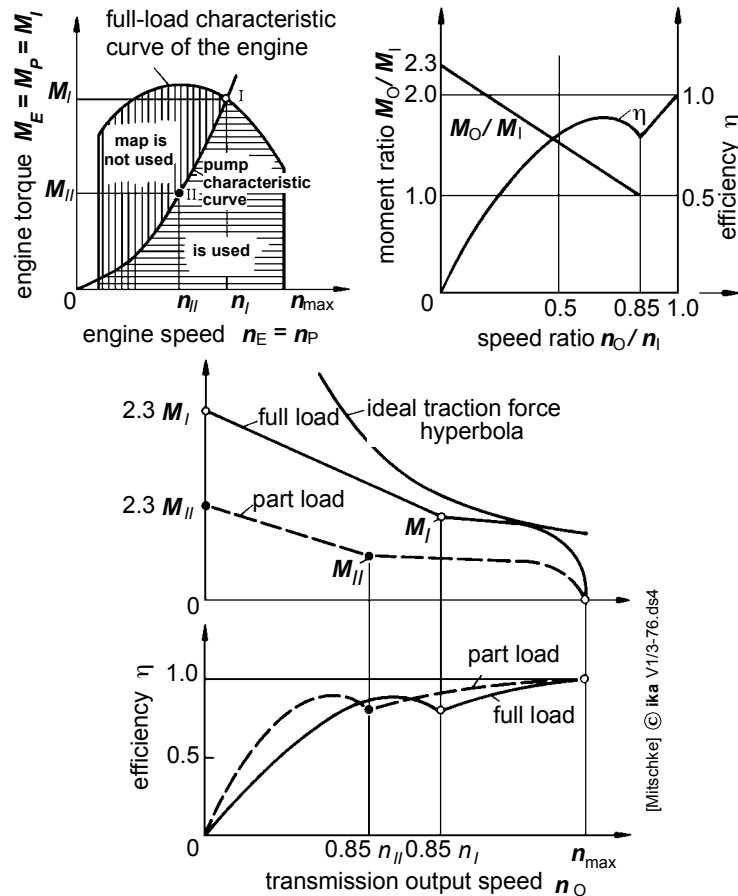


Fig. 3-76: Interaction between combustion engine and trilok converter (Source: Mitschke; „Dynamik der Kraftfahrzeuge“)

Compared with the ideal torque curve, there are still relatively large differences between the torque demand and torque supply. This means that the attainable conversion area (starting conversion approx. 2.0-2.5) and the poor efficiency at high torque ratios are not sufficient for the exclusive employment as a torque converter in the motor vehicle. Trilok converters are thus combined with stepped transmissions to automatic transmissions. Apart from the favourable approach towards the demand characteristic curve achievable here, the advantages of the trilok converter become obvious only in this combination. The advantages include compactness, good heat

dissipation over the hydraulic fluid, the fact that it is largely free of wear, and also in its function as a torsional vibration damper.

b) Hydrostatic transmission

Hydrostatic transmissions with two displacement machines working as a pump or engine enable a variation of the combustion engine speed independent of load. By shifting the hydraulic machines, e.g. axial piston pump or engine, the negative velocity of flow can be set in any direction between zero and the maximum value. Thus, in hydrostatic transmissions, neither starting clutches nor gear clusters are required as back up. Transmission that have these characteristics, are called IVTs (Infinitely Variable Transmissions). Fig. 3-77 shows the conversion characteristic of such a transmission. It essentially corresponds to the principle of the aforementioned mechanical CVT.

The disadvantage here is that each of the two hydraulic machines have to transfer the entire driving power and thus they have to be dimensioned correspondingly large. This has a particular negative effect on the efficiency of the transmission. Further disadvantages of hydrostatic transmissions include the unfavorable specific power output, high production costs, and the noise development. Therefore we do not deal with this transmission any further. Such transmissions are often used in construction and agricultural machinery, partly as components of hi-tech transmissions in which a mechanical branch is responsible for the improvement of efficiency.

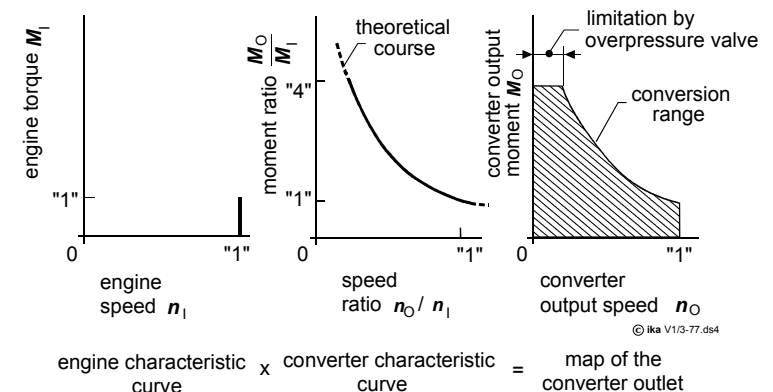


Fig. 3-77: Characteristic conversion of an IVT

3.4.4 Automatic transmissions (AT)

There are different possibilities to realize automatic transmissions. In doing so, the following concepts are mostly used:

- U Transmission with Hydrodynamic Converter,
- Automated Manual Transmission,
- Mechanical CVT.

a) U Transmission with Hydrodynamic Converter

The by far most widespread combination is the one consisting of a trilok converter and power-shifted U-transmission. It has already been explained that the trilok converter alone does not provide a sufficient delivery map (Fig. 3-76). Fig. 3-78 shows the torque characteristic which can be achieved using a rear-mounted stepped transmission.

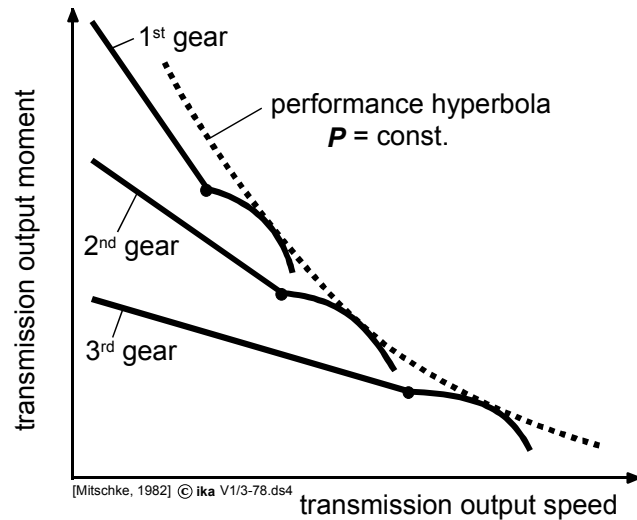


Fig. 3-78: Delivery map of an automatic transmission (Source: Mitschke; „Dynamik der Kraftfahrzeuge“)

Fig. 3-79 shows the example of a passenger car three-speed automatic transmission.

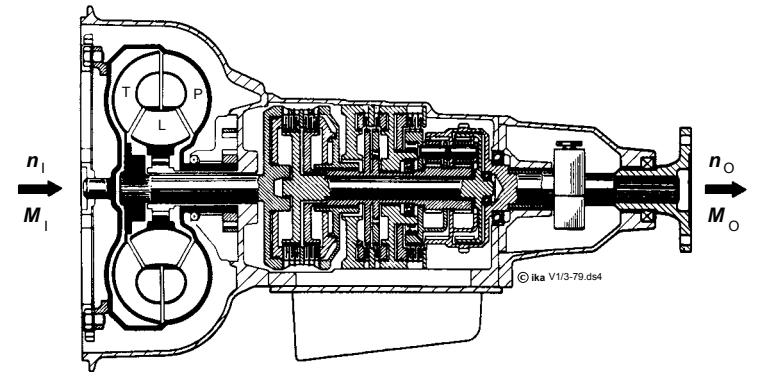


Fig. 3-79: Passenger car three-speed automatic transmission (ZF)

Fig. 3-80 shows the power flow of this automatic transmission for the three forward gears and the reverse gear. The gearshift from 3 to 5 in passenger cars and 4 to 6 in commercial vehicles, takes place hydraulically over frictional gearshift agents (clutches, brakes, and one-way clutch). The supply to the control hydraulics takes place over an oil pump attached to the transmission input. If such a transmission is to be tow started, i.e. if the engine is dead, an additional pump has to be arranged at the transmission output.

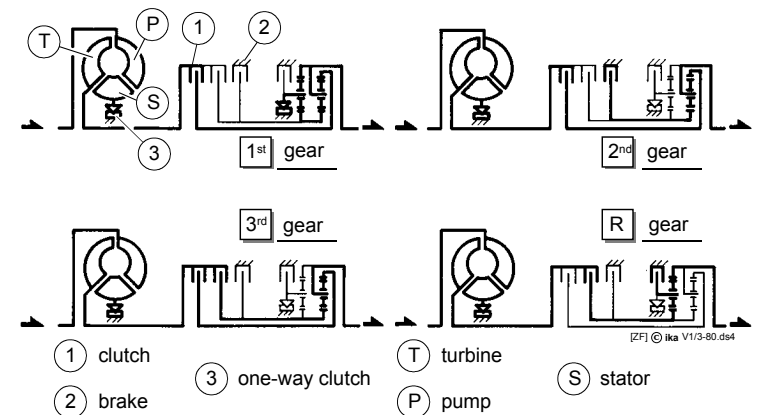


Fig. 3-80: Power flow diagram of a passenger car three-speed automatic transmission (ZF)

The classical automatic transmission offers a number of advantages:

- Favorable symmetrical design,
- Simple arrangement of shifting elements,
- Low construction volume,
- High transferable power output,
- Power shifting capability,
- Excessive increase of torque at start,
- Excessive torsional vibration damping.

Apart from higher production costs, an essential disadvantage compared to usual manual transmissions, is the loss of efficiency as a result of the hydrodynamic converter operating under slip. When a lock-up clutch is installed, these losses can be minimized. In the higher gears and at a minimum speed, a mechanical lockup is hydraulically actuated. The advantage of excellent torsional vibration damping is however lost in this case. This is the reason why not all manufacturers have accepted this solution. The number of gears was also increased in order to achieve better driving performances and low consumption. Fig. 3-81 shows a modern 5-speed automatic transmission for passenger cars.



Fig. 3-81: 5-speed automatic transmission (Mercedes-Benz)

The quality of an automatic transmission is not exclusively dependent on the mechanical layout, rather more decisively on the control strategy. Fig. 3-82 shows the basic gearshift strategy for a four-speed AT. In this AT, the driver can choose between a driving style oriented towards economy or power and directly select individual gears using a selector.

Depending on different engine and vehicle parameters, it should be possible to shift into an optimum gear for the particular driving situation. This can only be achieved by expensive electronic control. Fig. 3-83 shows the functional structure for a modern adaptive transmission control that automatically adjusts to the driving behavior. It is adjusted in such a way that the gearshift limits move, as it is shown in Fig. 3-82.

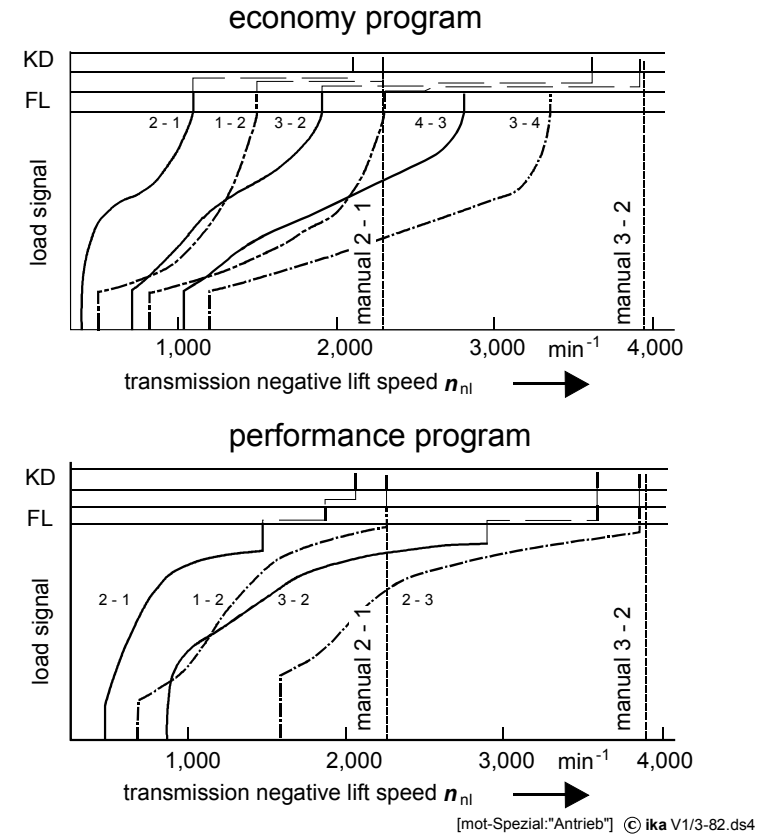


Fig. 3-82: Shifting strategy for a 4-speed automatic transmission (Source: mot Spezial, Antrieb) © Ika V1/3-82.ds4

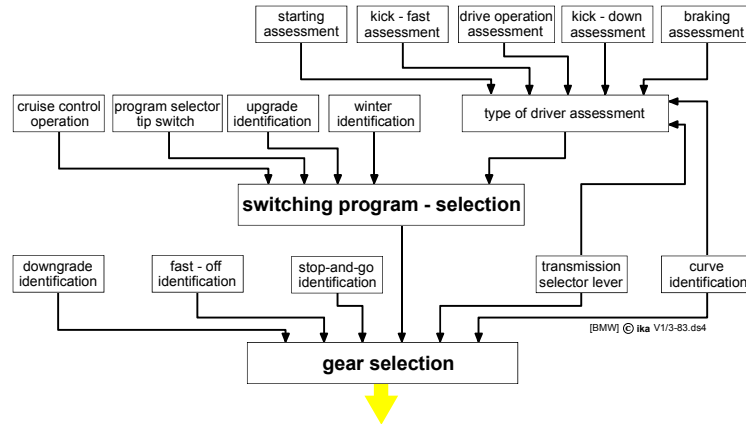


Fig. 3-83: Functional structure of an adaptive transmission control (BMW)

b) Automated Manual Transmission

Due to the already-mentioned disadvantages of the classical AT, which include high production costs and low efficiency, there are repeated attempts at automating the conventional manual transmission. It is not difficult to automate the transmission. In small transmissions it can be supported by the gear selector drum used in motor vehicles. It is more complicated to automate clutch operation (Chapter 3.3.1). Particularly in trucks, the semi-automated manual transmission is often used. Here, the choice of the gear and the clutch operation are left to the driver. The pre-chosen gear is shifted automatically.

The twin-clutch transmission is a special form in which shifting and coupling cannot be carried out manually. The driver decides whether the transmission is to be operated as a fully automatic transmission or a semi-automatic. Here, the driver chooses the gears himself, but does not engage the clutch (Chapter 3.4.1).

c) Mechanical Continuously Variable Transmissions

CVTs, like twin-clutch transmissions, can only be realized as automatic transmissions. As an example of a U transmission, Fig. 3-84 shows Ford's CTX transmission.

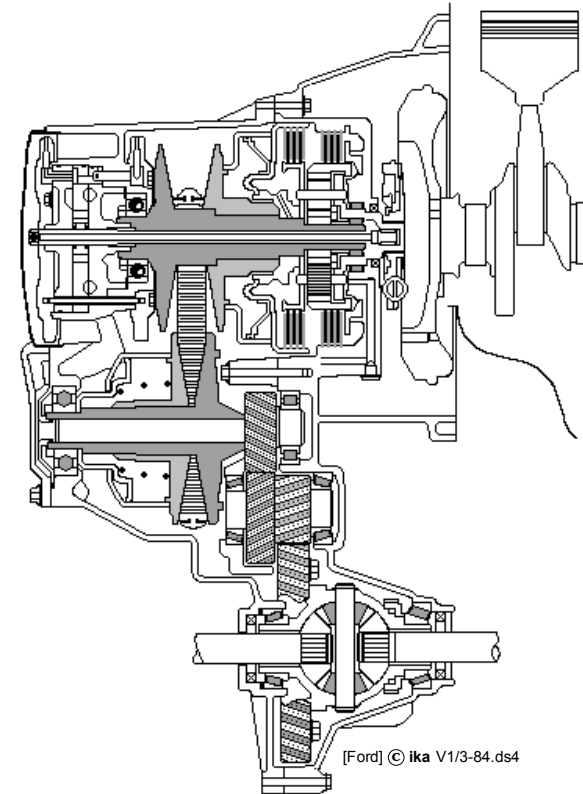


Fig.3-84: U transmission (Ford)

A planetary transmission with a pair of planetary trains and two hydraulically controlled clutches is front-mounted to the U transmission. This enables the starting and backing up functions. The continuously variable adjustment of the U transmission takes place by displacing the axially adjustable cone disks within the variator. This leads to a variation of the running radius of the thrust link band. Subsequently the driving torque is transferred over gear wheels onto the differential. For bigger vehicles with correspondingly higher transmission input torques, U transmissions in combination with a hydrodynamic clutch have already been realized (Fig. 3-85).

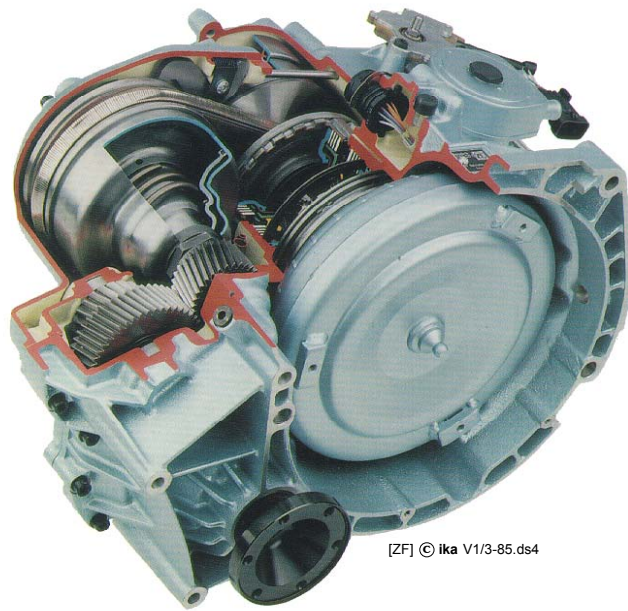


Fig. 3-85: U transmission with hydrodynamic clutch (ZF)

3.4.5 Comparison of Transmissions

Every conventional vehicle drivetrain today, is equipped with a torque converter since either the propulsion characteristic is not suitable or the required control area cannot be realized without conversion or if at all, only in association with large losses. The selection criteria for torque converters, similar to engines, can be classified into the three areas

- Operating behavior,
- Economy, and
- Influence on the environment.

The following transmissions will be compared:

- Manual Transmission in Countershaft Design (MT)
- Automatic Transmission with Hydrodynamic Converter (AT)
- Twin-Clutch Transmission (TCT)
- U Transmission (UT)

- High performance Hydrostatic Transmission (HT)

For the types of transmissions already considered, the following Tables 3-10 up to 3-12 lists criteria according to which transmissions have been compared with each other. The evaluations here also have to be considered as generalizations. Isolated cases cannot be characterized appropriately.

a) Operating behavior

In order to have a high excess torque in each operating area, the transmission should convert the delivery map of the reciprocating engine in such a way that the maximum torque curve approaches the ideal power curve. CVTs fulfill this requirement. In CVTs an improved approach to the ideal power curve can be achieved by raising the number of gears.

The shifting time of manual transmissions with 0.5-1.0 s corresponds to that of automatic transmissions. CVTs continuously adjust to the particular driving condition. The shifting time here, depends on the magnitude of the desired change in ratios. Since extreme adjustments are required only rarely (kick down), CVTs have advantages when it comes to shifting time.

Manual transmissions that can only be shifted by a torque interruption, produce high acceleration and deceleration forces when shifting gears. These forces generally depend on how the clutch is operated by the driver. In automatic transmissions the dependence on the driver does not exist. In the ideal transmission, shifting from one gear to another would take place continuously variable and hence without jerks. When operating CVTs in practice, an essential problem consists in the driver expecting an increasing number of revolutions with a rising vehicle speed when accelerating the vehicle. The CVT with a subjectively complete separation of engine and vehicle behavior is contradictory to the usual impression and is hence not really considered as pleasant.

The shifting and information expense of the driver is high only in manual transmissions. The remaining systems are run automatically. With regards to the convenience of operation however, the degree of automation and the possibilities for intervention result only in minor differences.

Transmission	MT	AT	CTC	UT	HT
Approach to power curve	5	4	5	10	10
Shifting / Switching time	5	6	6	8	8
Shifting comfort	2	8	6	8	8
Control comfort	1	8	6	8	8

Table 3-10: Assessment of the operating behavior of different transmission designs

b) Economy

Fig. 3-86 shows a comparison of the efficiencies of different transmission concepts. The transmission efficiency of the manual 5-speed transmission, particularly in the lower part-load range, is superior to the remaining systems since no **setting hydraulic** has to be operated. In the central part-load range, the U transmission has a slight advantage compared to ATs. The high performance hydrostatic transmission has significantly lower values due to **running blind outputs** when starting and the losses in the hydrostatic control circuit.

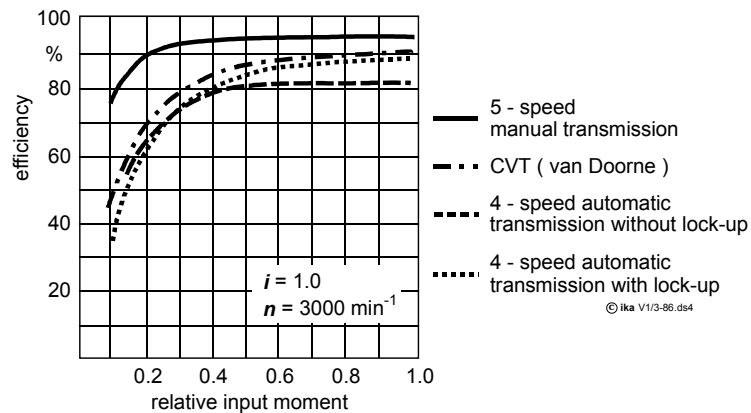


Fig. 3-86: Comparison of transmission efficiency

The total efficiency of engine and transmission, apart from the transmission efficiency, is essentially determined by the possibility to operate the engine in a low fuel consumption range. This is possible by a nearly continuously variable conversion and a large spread. Compared to stepped transmissions, the continuously variable transmission concept receives a better evaluation on the basis of range of transmission ratios which can be realised:

- U transmission: i_{ges} up to 7
- Hydrostatic Transmission: i_{ges} up to 25
- Passenger car Manual and Automatic Transmission: i_{ges} up to 5
- Truck Group Transmission: i_{ges} up to 15

Compared to the manual transmission, the electronic control of the double clutch transmission provides low fuel consumption values. In the automatic and U transmission, the energy consumption of the control hydraulics has a negative effect.

The production costs of the manual transmission are low. The high construction expense and the low stage of development make both the high-performance hydrostatic transmission as well as the U transmission cost-intensive concepts, the latter only to a limited extent.

The weight of the U and Double Clutch Transmission is just slightly higher than the manual transmission. Classical ATs generally weigh more. The high-performance hydrostatic transmission which is much heavier and requires much more space, has a different position.

Large differences between individual transmission designs cannot be expected when considering the criteria life and maintenance. An advantage of the AT is that the trilok converter largely compensates rotational vibrations and thus takes care of the entire drive train at an open bridging clutch.

Transmission	MT	AT	CTC	UT	HT
Transmission efficiency					
Overall efficiency					
Manufacturing costs					
Mass/volume					
Maintenance/service life					

Table 3-11: Assessment of economy of different transmission designs

c) Influence on the Environment

Similar to the characteristic curve of low consumption, every reciprocating engine also shows characteristic curves of low exhaust emission. In order to follow these characteristics, the transmission has to also operate as far as possible along a continuously variable characteristic with a large range of ratios.

The different transmission types differ only slightly in their noise behavior. Only the hydrostatic transmission tuned for high performance is generally very loud.

The possibility to lower the engine's noise emissions by following a noise-optimized control line, represents an advantage for CVTs. The hydrostatic transmission is an exception here since its noise emission is too high.

Transmission	MT	AT	CTC	UT	HT
Influence on emissions	6	5	7	8	10
Acoustic behavior	6	6	6	8	0

Table 3-12: Assessment of the noise behavior of different transmission designs

d) Summary

In summary, the following can be stated based on the assessment:

- Manual transmissions are still preferred because of their good economy in commercial vehicles and also in inexpensive passenger cars.
- With respect to several criteria, classical automatic transmissions come off with a poor evaluation when compared to U transmissions. Due to its high comfort and sophistication it is still quite suitable for high-quality passenger cars.
- CVTs will have good chance in the market in the future, particularly due to their good operating behavior, especially in small passenger cars, as an alternative to classical ATs which are not as suitable. When considering economy, CVTs are still not as good as manual transmissions.
- The twin-clutch transmission is a special form which in the sum total of all its characteristics, has no essential advantages when compared to conventional manual or automatic transmissions.
- Due to their size, weight, and noise behavior, high-performance hydrostatic transmissions only seem to be suitable for special vehicles in the future.

3.5 Differential (Transfer Gearbox)

In every dual-track vehicle with consisting of one driving unit, the driving torque has to be divided on both wheels of the driven axle. Additionally, in multi-axle drives, it is required to distribute the torque on the driven axles. When the vehicle is in motion, differences in road layout and speeds between the wheels of an axle or of the driven axles are encountered. As a result of this, the possibility of speed equalization should be available when distributing the torque. Both these tasks, i.e., torque distribution as

well as speed equalization are taken over by the differential. The differentials used in automobiles can be classified into:

- Bevel-gear differentials,
- Spur-gear planetary differentials.

3.5.1 Bevel-Gear Differential

The bevel-gear differential divides the driving torque symmetrically into two output torques of equal magnitude. It is a U transmission whose "hollow gear" and "sun gear" have the same radius or the same corresponding number of teeth. This is realized by using a bevel-gear teeth. Drive is transferred generally over two bevel-gear planetary wheels. The input torque is distributes to both output shafts in the ratio 1:1.

The drive axle differential is the version which is the most frequently used in nearly all two-track vehicles. Here, additionally, a torque converter with a firm gear ratio (drive axle ratio $i_{h(v)}$) is integrated. Fig. 3-87 shows a rear-axle differential for a standard drive. Driving shaft and rear axle are perpendicular to each other. The bridging of this axle angle and the torque conversion are taken over by a spiral or hypoid-toothed bevel-gear pair which consists of the drive-shaft pinion and the differential ring gear connected to the planetary support.

In the figure on the right side, the distribution of torque is indicated by representing the reaction forces. The input force F_E acting on the planetary pins is divided according to the lever arm a in the ratio 1:1. The external forces represented by the frictional forces $\mu \cdot F_{Z,W}$ between tires and road support the partial forces $\frac{1}{2} F_E$.

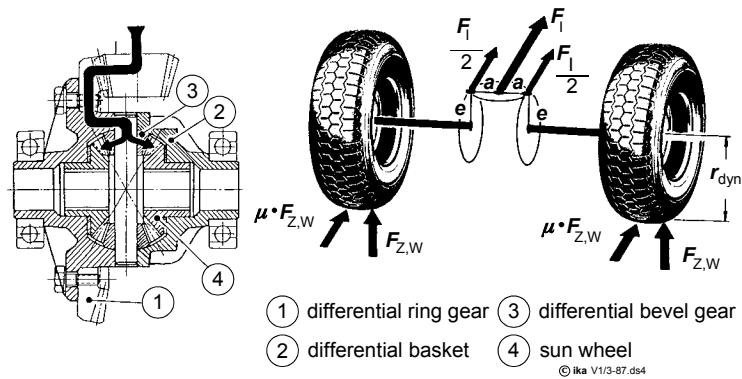


Fig. 3-87: Bevel-gear differential

The bevel-gear differential can further also be applied in the form of a center differential or transfer case in all-wheel drive vehicles. The driving torque in this case is symmetrically distributed between the front and rear axles. In the example presented in Fig. 3-88, the differential directly follows the transmission. The design of the transmission output shaft as a hollow shaft enables a compact design of the transmission unit.

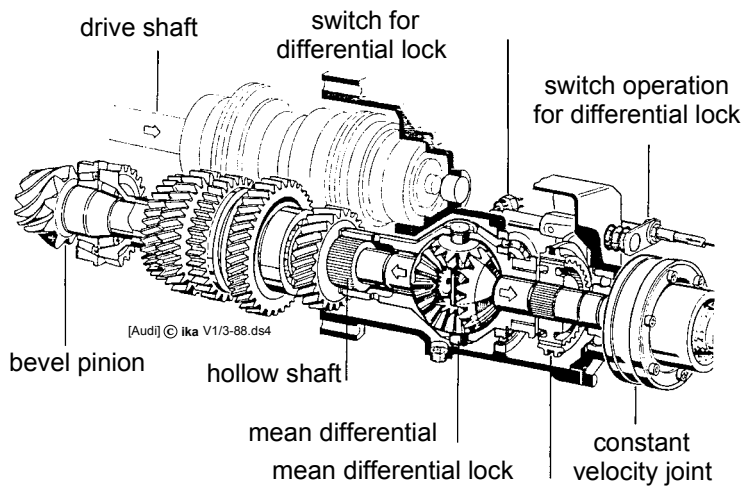


Fig. 3-88: Bevel-gear center differential (Audi Quattro)

3.5.2 Spur-Gear Planetary Differential

The spur-gear planetary differential produces an unequal torque distribution on the drive shafts. A spur-gear toothed planetary set is used along with hollow and sun wheels of different size. As illustrated in Fig. 3-89, the input is transferred over the planetary support. Hollow and sun wheel, each form an output. The ratio of the output torques is determined by the ratio i_0 of the pitch circle radii of hollow and sun wheel.

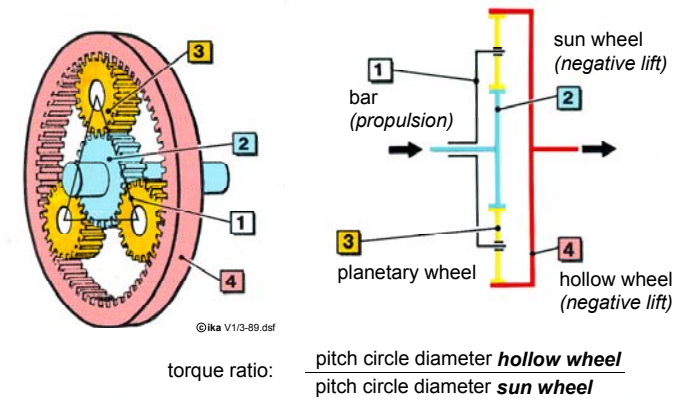


Fig. 3-89: Spur-Gear Planetary Differential

The spur-gear planetary differential is used as a center differential in multi-axle drives with asymmetrical torque distribution.

Fig. 3-90 shows the transfer case of an all-wheel driven vehicle.

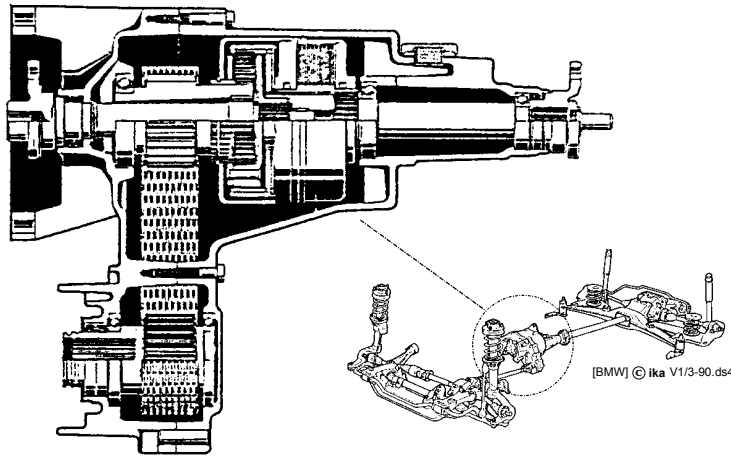


Fig. 3-90: Transfer gear (BMW 325 iX)

3.5.3 Differential Locks

Since the so-called “free” differentials only allow a certain torque ratio between both driving shafts, the tractive driving force of a vehicle is limited by the road-tire pair with the lowest usable friction. E.g. for a differing road surface contact between the left and right lanes, the wheel with the higher coefficient of friction can only transfer the maximum traction torque of the wheel from the smooth road. If the driving torque exceeds the maximum transferable torque limited by friction (Chapter 4.4), the wheel with the lower **friction potential** spins. The same happens in multi-axle drives between the axles. The torque can only be transferred in the determined ratio. In real driving conditions, differing road grip (μ -split) and differing road loads, e.g. in cornering, are reasons for the differing friction potentials.

If the wheels of an axle were rigidly connected with each other, in case of unequal friction between the right and the left lane, the larger adhesion potential could be utilised (when doing so, a moment around the vertical axis of the vehicle which can have a negative effect on driving stability arises). In order to achieve the transmission of a maximum driving torque, also when differentials are employed, differential locks were developed, which according to the design, enable the effect of a differing locking intensity between both output shafts. The achieved locking effect is characterized by the locking value S . It is defined as the ratio between the maximum possible torque difference at both differential outputs and the input moment M_E :

$$S = \frac{M_{Amax} - M_{Amin}}{M_E} \cdot 100\% \quad \text{Eq. 3-52}$$

When the losses through friction in the differential are ignored, we get:

$$M_E = M_{Amax} + M_{Amin} \quad \text{Eq. 3-53}$$

According to their efficiency, differential locks can be classified into:

- Positively-engaged differential locks,
- Frictional differential locks.

a) Positively-engaged Differential locks

The most simple and obvious design of a differential lock involves a positively-engaged combination that connects the output shaft with the planetary support and in turn with the other output shaft. With an engaged clutch (lock), the behaviour of the output corresponds to the one of an undivided shaft. The locking value amounts to 100%.

The clutch is manually engaged by the driver over a gearshift linkage or gearshift hydraulics. Fig. 3-91 illustrates a positively-engaged bevel-gear differential lock and a corresponding spur-gear differential.

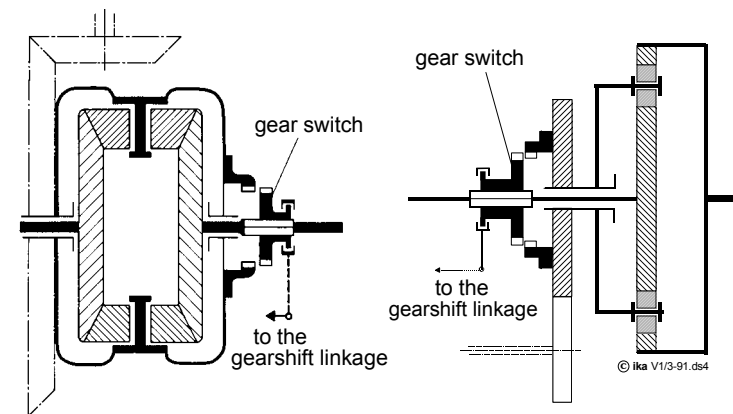


Fig. 3-91: Positive-engaged differential locks

The manual operation of positively-engaged differential locks requires higher attention on the part of the driver since a locked differential during normal driving leads to higher tire wear, strong additional loading of the drive train, ABS malfunction, and impeded maneuverability of the vehicle due to the absent speed compensation.

b) Adherent Differential locks

The locking effect of designs used today is based on adherent friction that hinders the compensation. It is produced by friction elements between the drive shafts over which a torque can be transferred, according to the design or driving condition it has a differing size.

One can distinguish between differential locks with self-regulating effect, without a possibility for external engagement, and the ones with regulating effect.

- Differential Locks with self-regulating effect

The simplest design of a self-regulating differential lock is the fixed value lock. Fig. 3-92 shows the design in a diagram.

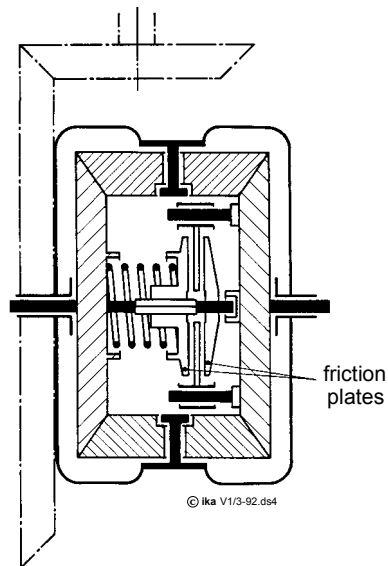


Fig. 3-92: Diagram of a fixed value lock

The fixed value lock complicates the compensation movement of both output shafts exclusively by a firmly defined friction moment, which is determined by the layout of the spring and friction surface. Thus the maximum torque difference between the outputs is constant. The locking value S defined in Eq. 3-52 depends on the specific loading condition. With an increasing load it diminishes.

In the case of a spin-resistant-differential (Borg-Warner) shown in Fig. 3-93, the frictional moment is transferred over two cone brakes, each linked with an output shaft-on the common differential housing and thus between both outputs. The braking force is produced on the one hand by a constant spring load and on the other hand, depending on the input torque, by the **axial tooth meshing forces** between planetary wheels and axle cone wheels. Thus the locking torque consists of a fixed and a load-dependent share.

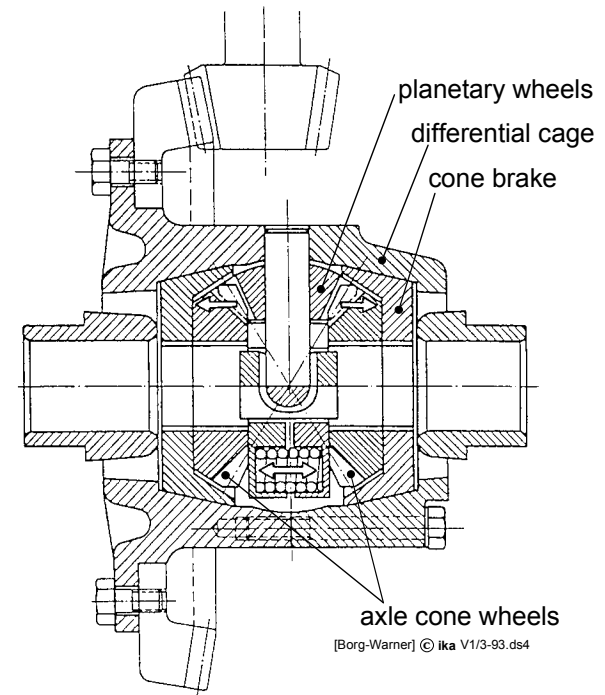


Fig. 3-93: Spin-resistant-differential lock (Borg-Warner)

The locking effect of locking worm differentials is based on an increased tooth friction during the rotational compensating motion between the output shafts. This can be achieved using worm gears with a leading angle that is adjusted for high tooth face friction and by an increased number of gear wheel pairs. Using this principle, locking values up to 90% are achieved in the load-independent friction moment.

A special design of the locking worm differential, the Torsen-differential (TORque-SENSing) shown in Fig. 3-94, enables a frictional moment dependent on the input torque.

The worm wheel pairs presented in Fig. 3-95 indicate a varying slope over their length and can be axially displaced against a spring. At low loads, the slope associated with low losses comes into play such that as a result of the low number of gear wheel pairs meshed with the spur gears, the compensation is hardly affected. If the input torque increases, the worm wheels are displaced in the direction of the friction-intensive lead angles as a result of increasing circumferential forces. Locking values of about 90% are achieved.

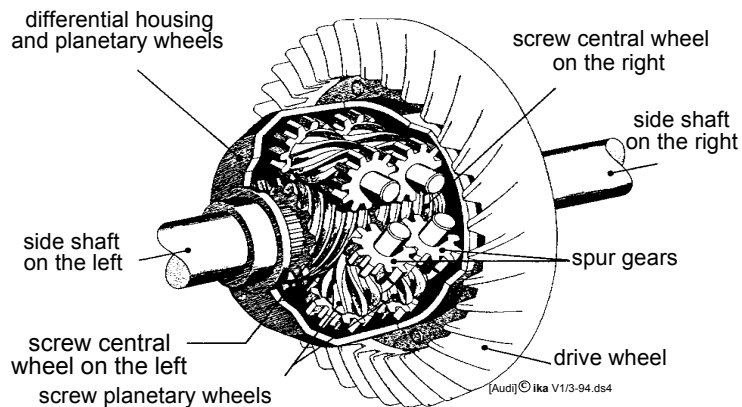


Fig. 3-94: Torsen-differential (Audi)

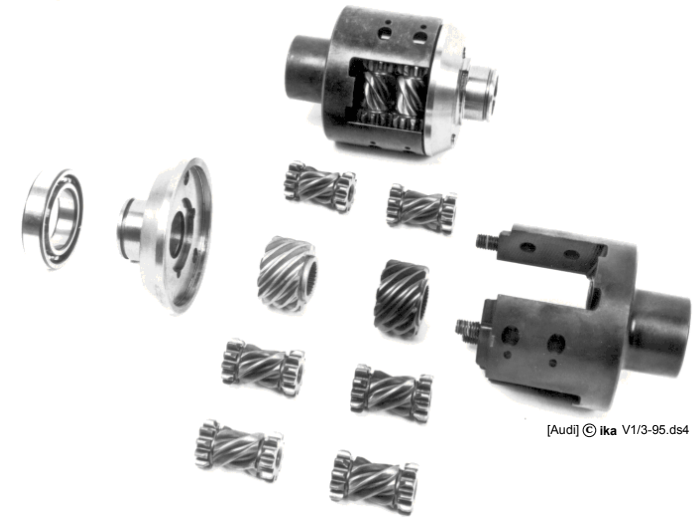


Fig. 3-95: Components of the Torsen-differential (Audi)

The differential lock represented in Fig. 3-96 operates according to the principle of the visco-hydraulic clutch.

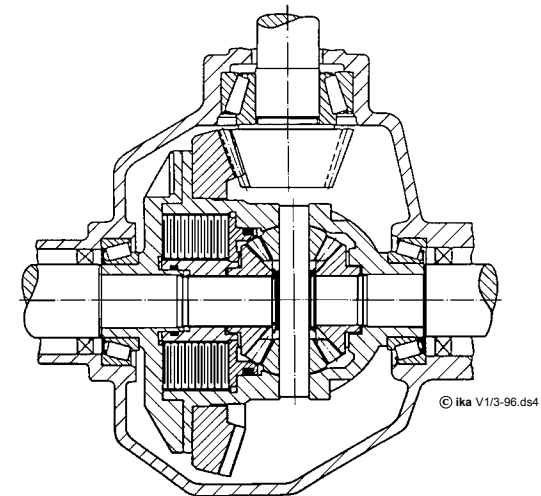


Fig. 3-96: Differential lock with visco-hydraulic clutch

In the design shown, the visco-hydraulic clutch transfers a frictional torque between the output shaft and the differential housing. This results in the relative motion between the wheels being impaired. The magnitude of the frictional moment depends exclusively on the speed of the lamellae against each other. Through a suitable layout (temperature and pressure development in silicone oil), an increased locking effect can be achieved by the so-called "Hump-effect".

The characteristic of the frictional torque over the differential speeds of the output shafts of the described self-regulating differential lock is shown in Fig. 3-97. A variation of the input torque is not considered in this diagram.

The transitional range of the higher static frictional torque to the constant locking torque in the range of sliding friction can be noticed in differential locks which operate based on mechanical friction. The frictional torque of the differential lock with visco-hydraulic however, degressively increases over the differential speed and thus provides a soft transition. In the transitional range, however, the visco-lock allows higher differential speeds when compared to mechanical locks. On a smooth snowed road, this can lead to the so-called "polishing" of the road, such that even with an increasing locking effect driving is no more possible. Here, differentials which already generate high locking effects at small differential speeds, are significantly better. This has led to a higher developmental effort in the direction of differential locks with regulating effect.

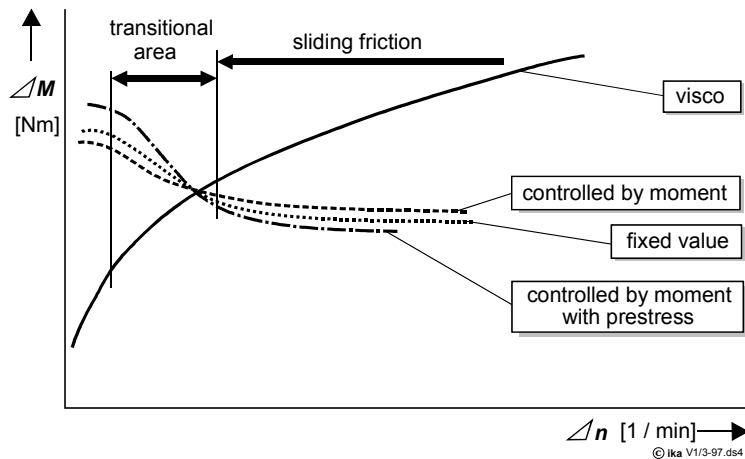


Fig. 3-97: Characteristic curves of different self-regulating differential locks

- Differential Locks with regulating effect

Controllable differential locks enable an adjustment of the frictional torque. This can take place by a stepped or continuously variable approach. The multi-disk clutch, which can be controlled using the vertical pressure, can be used as a friction element.

In the layout of such differential locks, it has however to be considered that the locking torque always remains smaller than the difference in transferrable traction forces. At the most, a wheel can spin (the corresponding is applicable to driven axles when a differential lock is used on a transfer case).

If the instantaneous frictional torque of axle differential lock is larger than the difference of the friction potentials at the left and right wheel of an axle, both wheels could spin at a correspondingly high input torque: the vehicle then would be difficult to control.

A continuously variable control of the frictional torque, based on the utilisable friction potential or driving condition, can take place using electronics. The possible input data include wheel speeds, wheel loads, steering angle, and transverse acceleration. The hence resulting locking torque, dependent on the driving situation, leads to optimum traction with simultaneously high driving stability. Fig. 3-98 shows an example for a hydraulically controlled automatic differential lock.

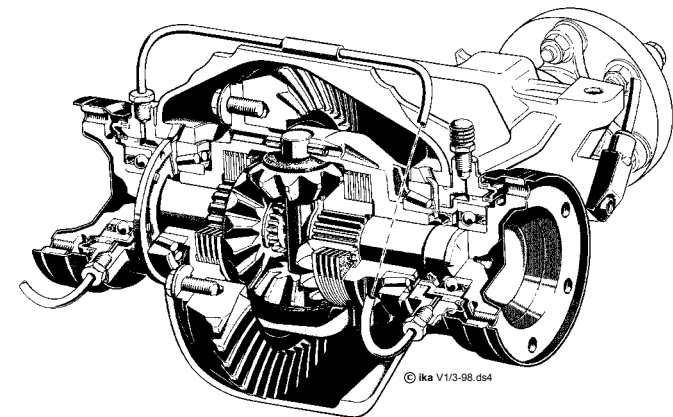


Fig. 3-98: Automatic differential lock

c) Comparison of the Systems

In extreme driving conditions, the manually operated, positively-engaged differential lock offers good improvement in traction compared to conventional differentials at a simultaneously low additional construction expense. This differential lock with its locking effect of 100% has to be released when used in general driving conditions for reasons of driving safety. It is employed in inexpensive vehicles and in the off-road-segment.

In differing driving situations, self-regulating differential locks do not require any control effort on the part of the driver or by control electronics. The locking torque is generally variable, dependent on the differential speed or load torque. The frictional torque characteristic is however fixed. Hence, for a range of frictional conditions, no favourable results can be achieved with respect to traction and driving stability.

The use of automatic lock release (see Chapter 4.4.4) turns out to be problematic since their control electronics require a decoupling of the wheels. In addition there are construction-specific disadvantages such as increased wear in the locking worm differential or the response of the visco-lock, which only occurs at a considerable wheel slip.

The electronically controlled differential lock delivers optimum traction in stable driving conditions by adjusting its map according to the specific driving condition. An antiskid device (ASD) can be taken into consideration in the program layout (withdrawal of the locking torque when braking). The high additional expense as a result of measuring detectors, microprocessors, and additional hydraulics stand in the way of broader application.

3.6 Brakes

Brakes are primarily used to decelerate a vehicle beyond its road resistances and the braking effect of the engine. Kinetic energy of the vehicle is then transformed into heat. Besides this principal form of employment, the task of brakes also consists in keeping the vehicle at standstill or (in downward gradients) at a constant speed.

Generally, the braking systems in a vehicle can be classified as follows:

- Service brake system: general decrease of speed in normal driving
- Emergency brake system: takes over the function of the service brake when it fails

- Parking brake system: prevents unwanted rolling of a parked vehicle
- Continuous service brake system: for longer uninterrupted braking and frequent stops (public busses)

The service, emergency and parking brake systems directly work on the wheel brakes. Brake elements of the continuous service brake system generally exert their decelerating moment on the drive train.

A common vehicle brake system consists of:

- The control device : pedals
hand-brake lever
 - The transmission unit: components between control device and brake
 - hydraulic
 - pneumatic
 - mechanical
- } brake-power transmission
- The wheel brakes

Fig. 3-99 shows the components of a 4-wheel disk-brake system with a diagonal brake circuit layout (Chapter 3.6.3) for passenger cars.

The formation of the wheel braking force in the tire tread area as a result of the driver exerting a foot force along with the accompanying internal forces is shown in Fig. 3-100.

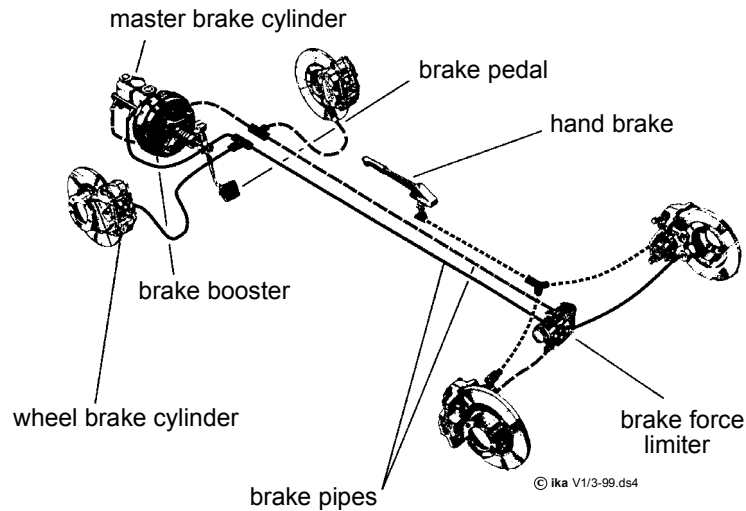


Fig. 3-99: Passenger car 4-wheel disk-brake system

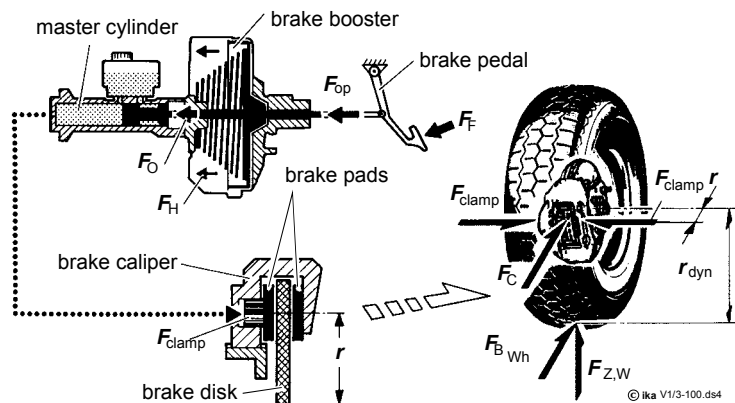


Fig. 3-100: Forces in a braking system

The foot force F_F applied by the driver is initially multiplied by the brake pedal leverage into the input force for the control device, the control force F_{con} . The transmission device, which in this case is hydraulic, converts the control force which is possibly supported by a brake booster which amplifies it further, into the auxiliary force F_{aux} which is in turn transferred as hydraulic pressure to the master brake cylinder. The brake force reaches the wheel brake cylinders via the brake lines and

forms the clamping force F_t required for operating the wheel brakes. The clamping forces cause a circumferential force F_{per} over the friction linings at the effective braking radius r and thus a wheel brake force F_{Bwheel} in the tire tread area.

The resulting wheel brake force results from the circumferential force to:

$$F_{Bwheel} = F_{per} \cdot \frac{r}{r_{dyn}} \quad \text{Eq. 3-54}$$

by the brake characteristic C^* , also called inner multiplication or multiplication of the wheel brake:

$$C^* = \frac{F_{per}}{F_t} \quad \text{Eq. 3-55}$$

and the outer multiplication i_o (multiplication of the transmission device):

$$i_o = \frac{F_t}{F_{con}} \quad \text{Eq. 3-56}$$

we get:

$$F_{Bwheel} = \underbrace{F_{con} \cdot i_o}_{F_t} \cdot C^* \cdot \frac{r}{r_{dyn}} \quad \text{Eq. 3-57}$$

Before dealing with the components of the braking system in more detail, the fundamental regulations associated with brakes should be briefly mentioned here.

3.6.1 Fundamental Regulations

Apart from the German national provisions specified by the *Straßenverkehrs-Zulassungs-Ordnung (StVZO)* under § 41 "Brakes and **Wheel arresters**", since 1980 the provision of the European Community ECE-R 13/05 "Brake Systems" is also applicable to the Federal Republic of Germany. It is valid for all road vehicles with a design related maximum speed of more than 25 km/h (15 mph). It subdivides these vehicles into different classes according to Table 3-13. For the sake of completeness, the classes of trailers are also mentioned here.

The vehicles of the classes M and N have to fulfill certain requirements according to the service, emergency, and parking brake system, which will be described more precisely in the following Table 3-13.

Class M	Class N	Class O
for passenger transportation determined motor vehicles with not less than 4 wheels or with 3 wheels and an overall weight > 1 ton	for goods transportation determined motor vehicles with not less than 4 wheels or with 3 wheels and an overall weight > 1 ton	trailer (including semitrailer)
M1: apart from driver's seat up to max. 8 seats	N1: maximum permissible weight < 3.5 ton	O1: maximum permissible weight < 0.75 ton
M2: apart from driver's seat more than 8 seats and maximum permissible weight 5 ton	N2: maximum permissible weight > 3.5 ton and < 12 ton	O2: maximum permissible weight > 0.75 ton and < 3.5 ton
M3: apart from driver's seat more than 8 seats and maximum permissible weight > 5t	N3: maximum permissible weight > 12 ton	O3: maximum permissible weight > 3.5 ton and < 10 ton
		O4: maximum permissible weight > 10 ton

4. Table 3-13: Classification into vehicles categories (ECE-R13/05)

a) Service Brake System

It must be possible for the driver to operate the service brake system in a graduated way without taking away his hands from the steering. Generally at least two independent braking circuits are required (dual-circuit brake system) since a minimum prescribed deceleration should be achieved in case of partial failure.

The prescribed brake values for the service brake system can be seen in Table 3-14.

Normal service:

Class	M ₁	M ₂	M ₃	N ₁	N ₂	N ₃
Test type	0-I	0-I	0-I-II	0-I	0-I	0-I-II
V1 [km/h]	80	60	60	80	60	60
aB [m/s ²]	5.8	5.0	5.0	5.0	5.0	5.0
sBmax [m]	50.7	36.7	36.7	61.2	36.7	36.7
FFmax [N]	500	700	700	700	700	700

In case of failure:

v1 [km/h]	80	60	60	70	50	40
Minimum aB [m/s ²]						
unloaded:	1.5	1.3	1.5	1.1	1.1	1.3
loaded:	1.7	1.5	1.5	1.3	1.3	1.3
FFmax [N]	700	700	700	700	700	700

where: v_1 =initial speed s_B =braking distance
 a_B =average full deceleration F_F =driver foot force

brake tests:

track calculation s_{Bmax} :

type 0: cold brake

type A: $0.1 \cdot v_1 + v_1^2/150$

type I: "fading" test

type B: $0.15 \cdot v_1 + v_1^2/130$

type II: braking on downgrades

(type A for class M₁, otherwise type B)

5. Table 3-14: Requirements for service brake system (ECE-R 13/05)

b) Emergency Brake System

The emergency brake system has to be operated in a graduated way as well. However, the vehicle driver can take one hand off the steering in order to do so. The deceleration achieved using the emergency brake system has to be at least half as big as the one prescribed for the service brake system. In this case, the hand and foot forces for the driver stated in the table below cannot be exceeded.

Vehicle class	max. hand force	max. foot force
M ₁	400 N	500 N
remaining classes	600 N	700 N

c) Parking brake system

6. The parking brake system must be able to hold the loaded vehicle at standstill on a gradient of 20%, in case of trailer trucks, the towing vehicle coupled to a trailer or semitrailer should be held at standstill in a gradient of 12%. The maximum permissible hand or foot forces are shown in the following table.

Vehicle class	max. hand force	max. foot force
M ₁	400 N	500 N
remaining classes	700 N	700 N

In the parking brake system, power transmission should only take place mechanically. In most cases, the parking brake system at the same time takes over the job of the emergency brake system.

d) Continuous Service Brake System

The provisions of the EC stipulate the use of continuous service brakes for vehicles of the classes M₂, M₃, and N₃, if the service brake does not pass the prescribed tests (type I and II). The function of the continuous service brake system has to be guaranteed independent of the condition and efficiency of the remaining brake systems.

The German *Straßenverkehrszulassungsordnung* (road traffic registration regulation) goes beyond this. It stipulates continuous service brakes for all commercial vehicles and trailers with a permissible total load $G_{\max \text{ perm}} > 9 \text{ t}$ and all busses with $G_{\max \text{ perm}} > 5.5 \text{ t}$.

Here, the continuous service brake when independently operated should ensure an uninterrupted speed of $v = 30 \text{ km/h}$ (19 mph) over a distance of 6 km (3.7 miles) on a 7% downward gradient.

The ECE-R 13/05 represents the existing law within the European Community. In other countries often different provisions have to be observed, such as the existing Federal Motor Vehicle Safety Standards (FMVSS) in the USA.

3.6.2 Wheel Brakes

Service, emergency, and parking brake systems exert their decelerating or clamping forces on the wheel brakes. Drum and disk brakes are used as wheel brakes.

a) Drum Brake

Drum brakes can generally be realised in the form of band or shoe brakes. The frictional braking force acts on the rotating brake drum in one of two ways, internally outwards or externally inwards. The conventional design consists of the internal shoe brake which generally includes two shoes in one drum.

According to the form of transfer of the clamping force and the shoe support, drum brakes are classified into different categories:

- Simplex-brake
- Duplex-brake
- Servo-brake
- Duo-Duplex-brake
- Duo-Servo-brake

As far as the shoe support is concerned, one can distinguish between leading and trailing shoes. As a result of the support of the leading shoe, a self-energization of the brake effect results from the frictional force which acts around the center of rotation of the shoe in the running direction. Correspondingly, in the trailing shoe the friction force leads to a self-attenuation since it does not act around the center of rotation of the shoe but against the clamping force. Table 3-15 compares the different categories of drum brakes.

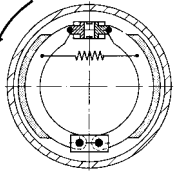
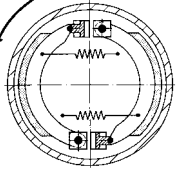
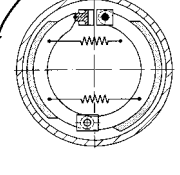
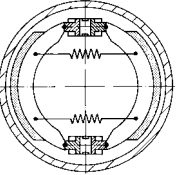
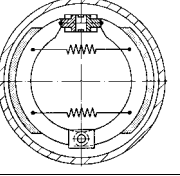
Configuration	Actuation	Support	Reinforcement
Simplex 	by double brake cylinder	axle fixed rotation point always one leading shoe and one trailing shoe	low self-energization, independent of rotation direction
Duplex 	by single brake cylinder (one each shoe)	axle fixed rotation point for each shoe both leading shoes in main running direction	maximum self-energization in primary rotation direction opposite direction high self-weakening
Servo 	only by single brake cylinder	supporting force of primary shoe is application force to secondary shoe, secondary shoe is supported against axle fixed rotation point moveable rotation point between shoes	maximum self-energization in primary rotation direction opposite direction high self-weakening
Duo-Duplex 	by two double brake cylinder with integrated piston stop	self-energizing arrangement of leading and trailing shoe, same for both running directions	high self-energization in both rotation directions
Duo-Servo 	by one double brake cylinder with integrated piston stop	moveable rotation point between shoes	maximum self-energization in both rotation directions

Table 3-15: Drum brake designs

b) Disk Brake

In the disk brake, friction linings are axially pressed against the brake disk which rotates with the wheel. This brake design, in contrast to the radially acting drum brake, is also called the Axial Brake.

Disk brakes used in today's motor vehicles are realized without exception in the form of partial disk brakes. This means that the friction lining only covers a sector of the disk friction lining. One can distinguish here between fixed and floating caliper disk brakes. Table 3-16 compares these systems.

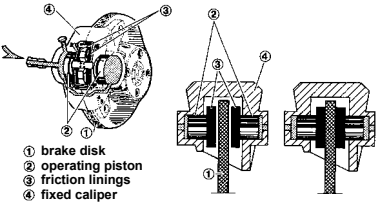
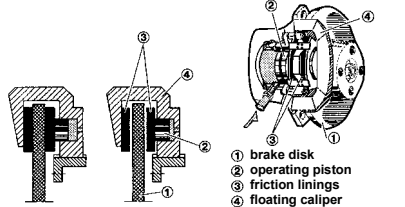
Fixed-caliper disk brake	Floating-caliper disk brake
 ① brake disk ② operating piston ③ friction linings ④ fixed caliper	 ① brake disk ② operating piston ③ friction linings ④ floating caliper
The brake disk is surrounded by the caliper, which is bolted to the wheel carrier On both disk sides an axial sliding, hydraulically or mechanically actuated, piston presses the friction lining against the disk Generally even pressure force of both brake pads Hydraulic line in caliper above hot brake disk may produce vapor lock Costly design	Axial moveable caliper with only one operating piston Floating caliper and operating piston pressing friction linings against disk Higher clamping forces are required because of friction in caliper guide Better cooling of brake disk as a result of lower volume of work smaller space requirements on the side facing to the rim less costly in manufacturing

Table 3-16: Comparison of disk brake designs

c) Brake characteristic

The brake characteristic or inner multiplication C^* characterizes the transmission of the induced clamping force into a braking circumferential force within the wheel brake.

$$C^* = \frac{F_U}{F_{Sp}}$$

Eq. 3-58

Table 3-17 summarizes the accompanying equations for disk- and drum brake.

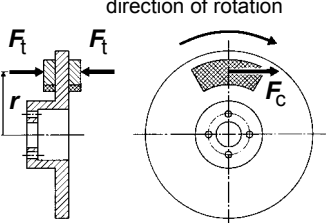
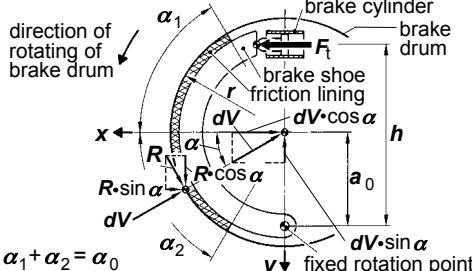
disk brake	drum brake
	
Clamping force corresponds to vertical force on disk: $F_{Sp} = N$ Circumferential force on both sides of disk: $F_{U1} = F_{U2} = m \cdot F_{Sp}$ brake coefficient: $C^* = \frac{F_U}{F_{Sp}} = \frac{2 \cdot \mu \cdot F_{Sp}}{F_{Sp}}$ $C^* = 2 \cdot \mu$	Clamping force has no constant vertical force distribution over the friction lining circumference. Approximation: cos-formulation of pressure distribution Coefficient of leading shoe: $C_1 = \frac{\frac{h}{r}}{f(\alpha_0) \cdot \frac{a_0}{r} - \mu} \cdot \mu$ with: $f(\alpha_0) = \frac{\alpha_0 + \sin \alpha_0}{4 \cdot \sin(0,5 \cdot \alpha_0)}$ Coefficient of trailing shoe: $C_2 = \frac{\frac{h}{r}}{f(\alpha_0) \cdot \frac{a_0}{r} + \mu} \cdot \mu$ Overall coefficient (e.g. for simplex brake) $C^* = C_1 + C_2$ $C^* = \frac{f(\alpha_0) \cdot \frac{a_0}{r}}{\left(f(\alpha_0) \cdot \frac{a_0}{r}\right)^2 - \mu^2} \cdot \frac{h}{r} \cdot 2 \cdot \mu$ self-locking condition: $\mu = f(\alpha_0) \cdot \frac{a_0}{r}$

Table 3-17: Calculation of disk- and drum brake

Fig. 3-101 compares the brake characteristics of different brake designs over the friction coefficient.

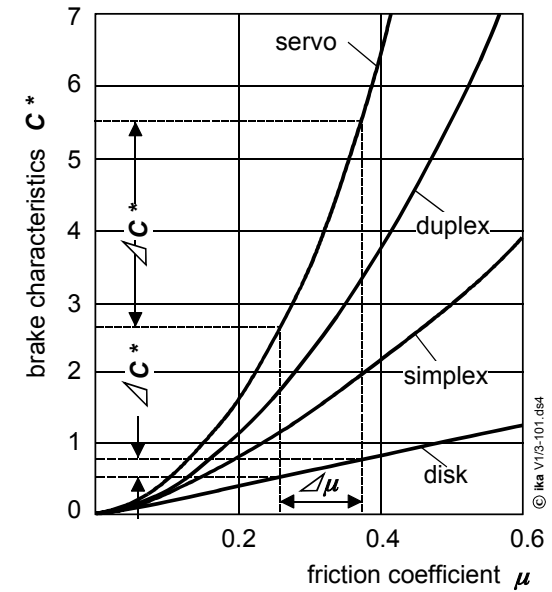


Fig. 3-101: Brake characteristic for different brake types

It becomes clear that in the case of brakes with high self-energization the variations in friction coefficient $\Delta\mu$ have a stronger effect on the circumferential force than in the case of brakes with low self-energization. A measure on how much the brake characteristic depends on changes of the friction coefficient is the so-called sensitivity E_0 :

$$E_0 = \frac{dC^*}{d\mu} \quad \text{Eq. 3-59}$$

A constant value results for the disk brake:

$$E_{0\text{disk}} = 2 \quad \text{Eq. 3-60}$$

The sensitivity of the Simplex-brake rises progressively already at low friction coefficients:

$$E_{0\text{simplex}} = 2 \cdot f(\alpha_0) \cdot \frac{a_0}{r} \cdot \frac{h}{r} \cdot \frac{(f(\alpha_0) \cdot \frac{a_0}{r})^2 + \mu^2}{((f(\alpha_0) \cdot \frac{a_0}{r})^2 - \mu^2)^2} \quad \text{Eq. 3-61}$$

Fig. 3-102 shows the sensitivity for disk and simplex-brake. For the simplex-brake, additionally the shares of both shoes are drawn in.

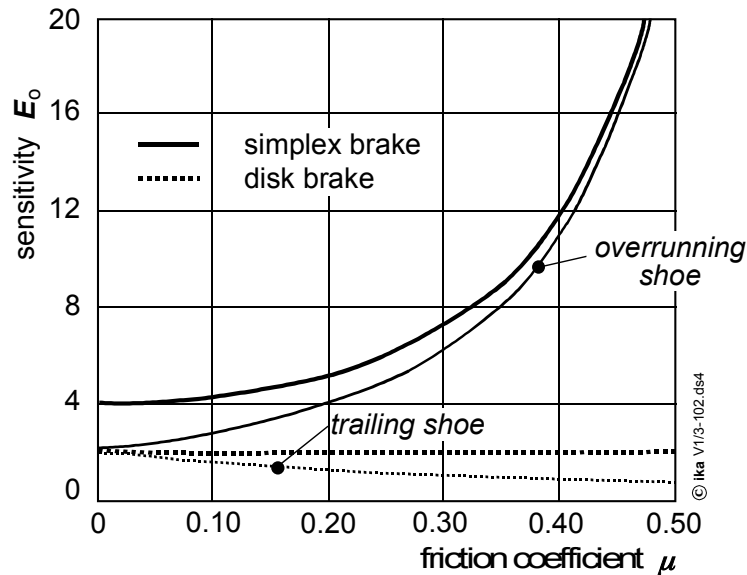


Fig. 3-102: Sensitivity E_0 dependent of the friction coefficient for disk-brake and simplex-brake

3.6.3 Braking Circuit Layout

In order to improve the safety of brake systems, the braking lines are generally arranged in the form of a double-circuit. In this way, a failure in one of the braking circuits, guarantees a braking effect of limited extent from the intact brake circuit.

In practice, five different brake-circuit layouts have been realized that are compared in Table 3-18:

Braking Circuit	Distribution	At failure
Standard 	one brake circuit per front- and rear axle	at brake-circuit failure vehicle remains stable if front circuit fails the deceleration will be lower (axle-load shift)
Diagonal 	each brake circuit contains one front brake and the diagonal rear brake	at circuit failure remains half of total brake power but: yawing moment to vertical vehicle axis because of different brake forces on front- and rear wheel
Extend. standard 	front brakes are in both circuits integrated	in relation to standard splitting better braking effect at failure of front circuit
Extend. diagonal 	each circuit on both front wheels and one rear wheel	at circuit failure always complete front brakes and one rear brake still working but: yawing moment to vertical axis
Double four-wheel 	two independent brake circuits on all four wheels	at brake-circuit failure maximum braking effect is still accessible

Table 3-18: Comparison of different braking-force divisions

3.6.4 Hydraulic Brakes

In service brake systems of passenger cars and lightweight commercial vehicles, the braking forces are generally transmitted hydraulically. The advantages of this form of force transfer lie in the incompressibility of the transferred medium, the low transmission losses, and last but not least, the higher flexibility in transfer lines when compared to mechanical elements.

A hydraulic brake system generally consists of the following design elements:

- Pedals (actuating device)
- Brake booster (optional)
- Master cylinder (transmission system)
- Wheel brake cylinder (transmission system)

a) Pedals

According to legal regulations, service brake systems are to be operated by the driver's foot over pedals. The pedals have to mechanically transfer the foot force while simultaneously amplifying it. The principal structure of a brake pedal is shown in Fig. 3-103.

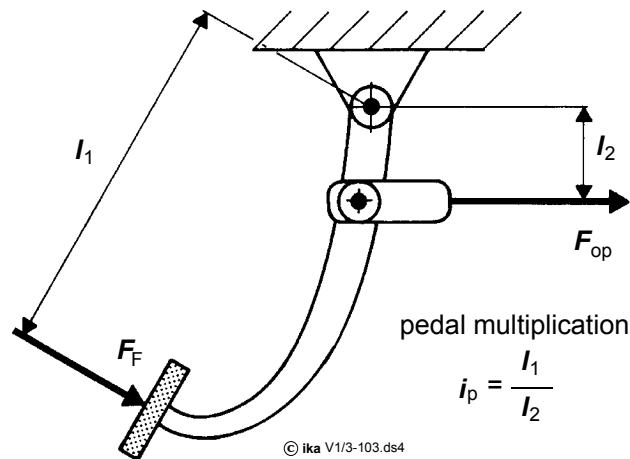


Fig. 3-103: Foot force and resulting operating force at the pedal

The foot force F_f is transformed over the pedal multiplication i_p into the actual control force F_{op} acting on the transmission system:

$$F_{op} = i_p \cdot F_f \quad \text{Eq. 3-62}$$

b) Brake Booster

If the legally limited control force is not sufficient both for the production of the braking force necessary for the vehicle's traffic safety as well as in terms of comfort, a brake

booster is used to generate an auxiliary force F_H , which superimposes this control force. Thus for the output force of the booster F_O we get:

$$F_O = F_{op} + F_H \quad \text{Eq. 3-63}$$

As characteristic value of the brake booster, we introduce the boost B^* :

$$B^* = \frac{F_O}{F_{op}} \quad \text{Eq. 3-64}$$

However, the boost has to be limited using constructive measures since the foot force required for reaching the prescribed braking retardation is not allowed to exceed a maximum value even in case of a failure of the brake booster which is a part of the transmission system. In order to achieve this, a so-called modulation point (F_{H0} , F_{op0}) is determined beyond which the auxiliary force is not increased any further by limiting the pressure (Fig. 3-104).

For $F_{op} < F_{op0}$ the boost factor is:

$$B^* = \frac{F_{op} + F_H}{F_{op}} \quad \text{Eq. 3-65}$$

with an auxiliary force proportional to the control force.

Over the proportionality factor that is valid up to the modulation point (and in it):

$$f_B = \frac{F_H}{F_{op}} \quad \text{Eq. 3-66}$$

as a result:

$$B^* = 1 + f_B = \text{const.} \quad \text{Eq. 3-67}$$

For $F_{op} > F_{op0}$ the boost factor is:

$$B^* = \frac{F_{op} + F_{H0}}{F_{op}} \quad \text{Eq. 3-68}$$

or:

$$B^* = 1 + \frac{F_{H0}}{F_{op}} = 1 + f_B \cdot \frac{F_{op0}}{F_{op}} \quad \text{Eq. 3-69}$$

In the design of braking systems, the modulation point should be fixed in such a way that the loaded vehicle can be braked at the blocking limit using the operating force F_{op0} so that the driver does not enter into the range of diminishing boost. Fig. 3-104 shows a usual booster diagram.

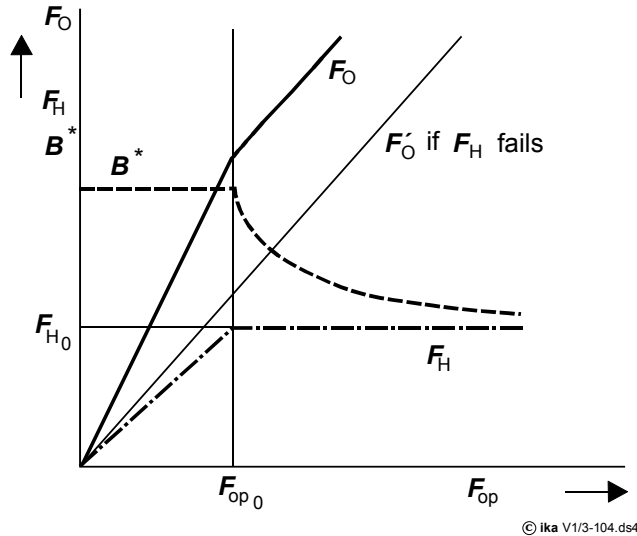


Fig. 3-104: Booster diagram

Meanwhile electronically regulated brake boosters are available today. They can fulfill the following tasks:

- In emergency situations, drivers often do not generate the necessary braking force for drastic braking. A sensor identifies this condition and ensures maximum possible deceleration (brake assistant).
- For traction and driving dynamics control, among others, certain targeted brake interventions are necessary. Since the driver does not operate the brake pedal while doing so, the necessary brake force has to be generated by the brake booster.

- The latest generations of cruise control also require brake interventions in the absence of a foot force created by the driver, such as on steep downward gradients or in automatic convoys.
- From the point of view of improved comfort and safety, the creation of a constant foot force at a constant deceleration, independent of load would be favourable. This also requires an individual adjustment of the brake boost.

It has to be stressed, however, that an electronically controlled brake booster does not regulate the brake force at the wheel, but is rather involved in the creation and regulation of the output force F_O .

The production of an auxiliary force in hydraulic brake boosters can in general be achieved pneumatically via the intake vacuum of the engine, or hydraulically using a separate pump. A vacuum brake booster is shown in Fig. 3-105. The intake manifold vacuum acts on both sides of an extensive diaphragm plate. A soft conical spring moves the diaphragm plate to one side of the vacuum booster. As soon as the driver operates the brake pedal, air at atmospheric pressure flows onto the opposite side of the diaphragm plate. The resulting pressure difference on both sides of the diaphragm plate produces the supporting auxiliary force. A control valve provides a constant boost factor up to the modulation point.

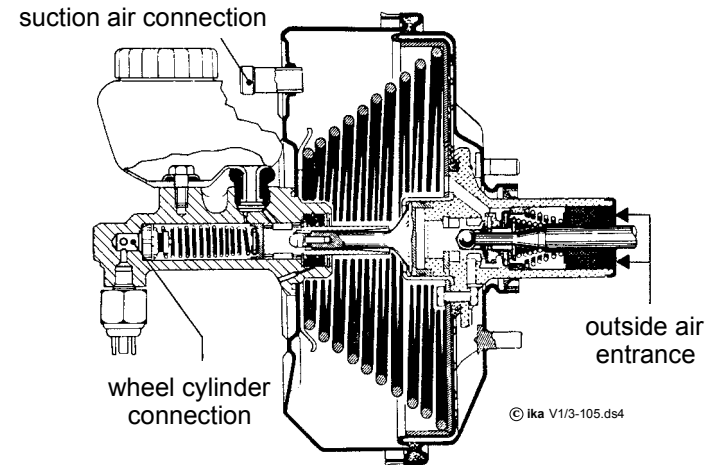


Fig. 3-105: Vacuum brake booster

If the engine conceptually does not produce sufficient vacuum (diesel engine) and if an hydraulic pump is already available, hydraulic brake boosters can be used (Fig. 3-106).

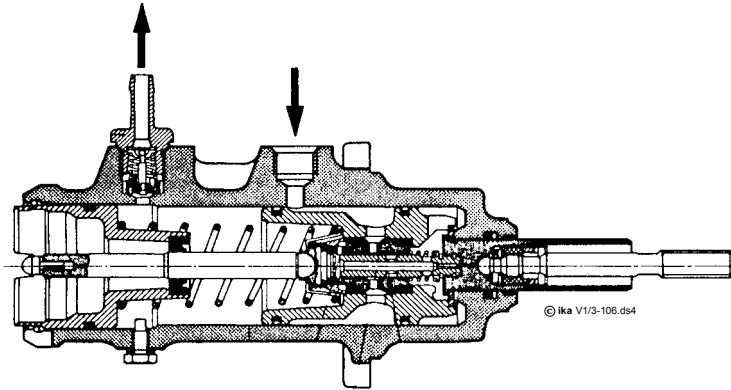


Fig. 3-106: Hydraulic brake booster

The advantages of a hydraulic brake booster are:

- short response time,
- full brake boosting even when the engine is cold,
- small design volume when hydraulic components are already available.

c) Master Cylinder

The master cylinder forms the central constructional element of the hydraulic braking circuit. It converts a mechanical operating force which is directly applied by the pedal or supported by a booster, into a hydraulic service pressure and simultaneously distributes it into two braking circuits.

Two basic concepts are used in vehicle engineering:

- Tandem Master Cylinder
- Stepped Master Cylinder

Table 3-19 shows the principal design and comparison of both master cylinder concepts.

Tandem Master Cylinder	Stepped Master Cylinder
Two single pistons, kept apart by compression springs, same effective area on both sides In case of circuit failure: Pedal-travel extension, same pressure with same operation force, but only in one brake circuit To produce an equal brake force the operating force has to be increased according to the braking balance Main advantage: relatively low production costs Pressure: $p_{mcT} = \frac{F_{op} \cdot B^* \cdot \eta_{mc}}{A_{mc}}$ B^*: coefficient brake booster η_{mc}: master cylinder efficiency A_{mc}: piston cross-sectional area master cylinder	One piston with two effective areas, one for each brake line In case of circuit failure: Same operation force in intact brake circuit produces increased working pressure, depending on effective area Relation between brake force and operating force is nearly constant pressure: $p_{mcS1} = F_{op} \cdot B^* \cdot \eta_{mc} \cdot \frac{A_{mc1}}{A_{mc1}^2 + A_{mc2}^2}$ $p_{mcS2} = F_{op} \cdot B^* \cdot \eta_{mc} \cdot \frac{A_{mc2}}{A_{mc1}^2 + A_{mc2}^2}$
Pressure in brake cylinder: $p_B = p_{mc} \cdot \eta_{bl}$ Clamping force in brake cylinder: $F_t = p_B \cdot A_{wc} \cdot \eta_{wc}$ Outer transmission: $i_o = \frac{F_t}{F_{op}}$	η_{bl}: brake-line efficiency A_{wc}: piston surface wheel cylinder η_{wc}: efficiency wheel cylinder
$i_{oT} = B^* \cdot \frac{A_{wc}}{A_{mc}} \cdot \eta^*$	$i_{oS1} = B^* \cdot \frac{A_{wc} \cdot A_{mc1}}{(A_{mc1}^2 + A_{mc2}^2)} \cdot \eta^*$ $i_{oS2} = B^* \cdot \frac{A_{wc} \cdot A_{mc2}}{(A_{mc1}^2 + A_{mc2}^2)} \cdot \eta^*$
with: $\eta^* = \eta_{mc} \cdot \eta_{bl} \cdot \eta_{wc}$	

Table 3-19: Comparison of Master Brake Cylinders

d) Wheel Brake Cylinder

The last link in the braking circuit of a service brake system is the wheel brake cylinder which converts the hydraulic brake pressure into a clamping force. At a prescribed service pressure, the cross-sectional area of this simple hydraulic component decides the magnitude of the clamping force and thus the brake force. The cross-sectional area of the brake piston is limited by the packaging space within the brake drum or on the wheel.

3.6.5 Air brake system

Average and heavy commercial vehicles are generally equipped with pneumatic braking systems. The possibility of the convenient coupling of additional braking circuits particularly when speaking of trailers serves as an advantage. The disadvantage of higher loads and space requirements is accepted in this class of vehicles. Fig. 3-107 shows the design and components of a double-circuit air brake system.

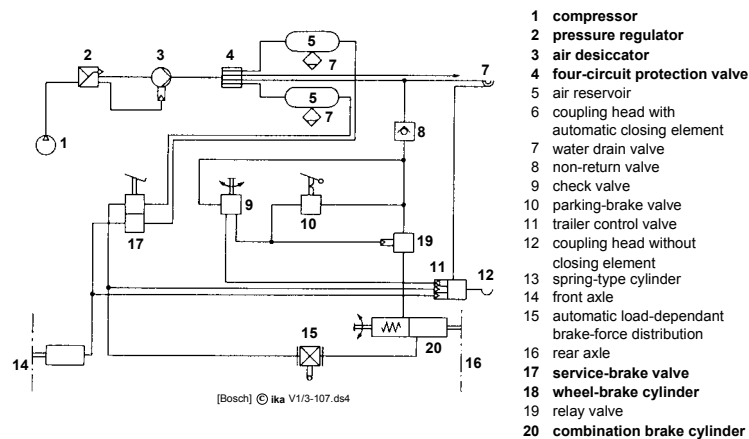


Fig. 3-107: Double-circuit air brake system (Bosch)

While the wheel brakes of the air brake system largely correspond to the ones in the hydraulic brake system, the constructional components of the control and transmission devices differ considerably.

In the following, the most important components of an air brake system are described.

Compressor (1)

The compressor is responsible for the production of compressed air and generally consists of a mechanical reciprocating compressor coupled to the combustion engine.

Pressure Regulator (2)

A pressure regulator follows the compressor. The regulator is nothing but an automatic switching valve which controls the operating pressure in the air-brake circuit. If the operating pressure set by a spring is achieved (approx. 8-10 bar), the air is blown off through a separate connection. The steadily-running compressor then only works against atmospheric pressure.

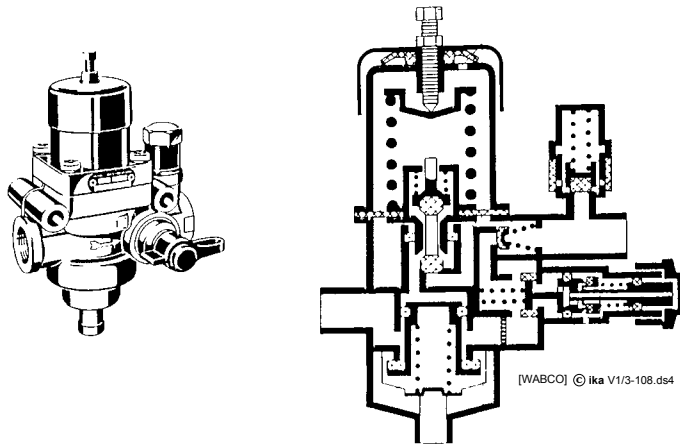


Fig. 3-108: Pressure regulator (WABCO)

Air Desiccator (3)

The air desiccator extracts moisture from the compressed atmospheric air. This is undertaken in order to prevent the condensation of water in the pneumatic system, which would otherwise lead to corrosion or blockages due to freezing at low temperatures.

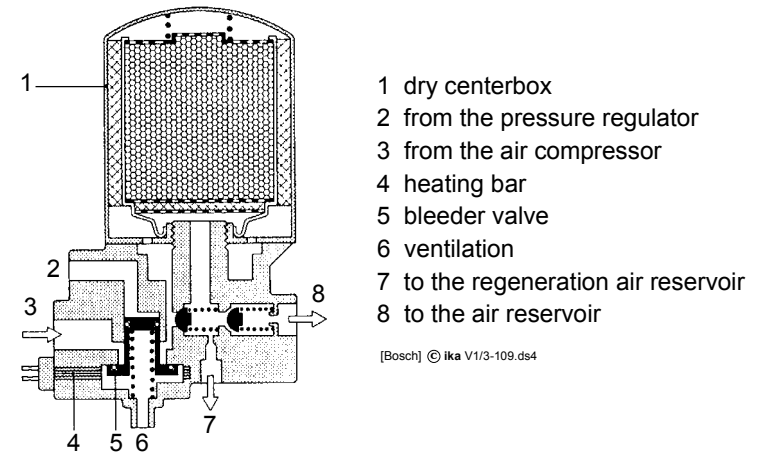


Fig. 3-109: Air desiccator (Bosch)

Multi-circuit Safety Valve (4)

The multi-circuit safety valve is responsible for the distribution of the compressed air into the brake circuits independent of each other. In case of a failure, one or more circuits protect the functional circuits against drops in pressure. The four-circuit safety valve is a commonly used version (Fig. 3-110). The four brake circuits are divided into two service brake circuits: one circuit is responsible for the compressed-air supply of the emergency and parking brake system as well as of the trailer connection and one circuit for further accessories such as the control of the exhaust throttle brake (engine brake). In case of a failure of one brake circuit, the pressure drop leads to the closure of a non-return valve on the damaged circuit, such that a counterpressure of a magnitude which prevents the opening of the intact brake circuits can be maintained.

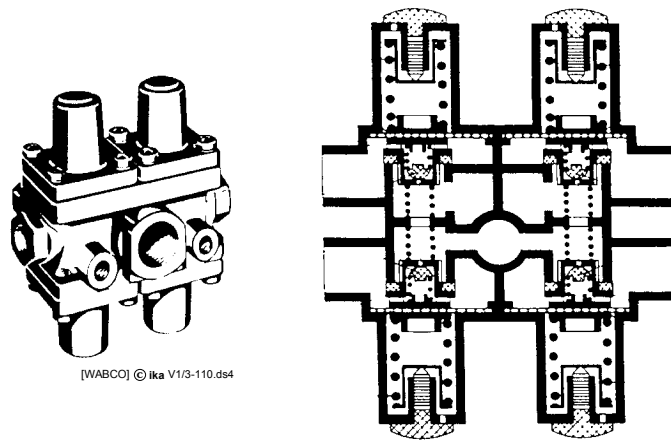


Fig. 3-110: Four-circuit safety valve (WABCO)

Service-Brake Valve (17)

The service-brake valve controls the level of the air pressure that acts on the wheel brakes according to force exerted by the driver on the pedal or the pedal position. The foot force, however, is used exclusively for brake force control and not for brake force generation.

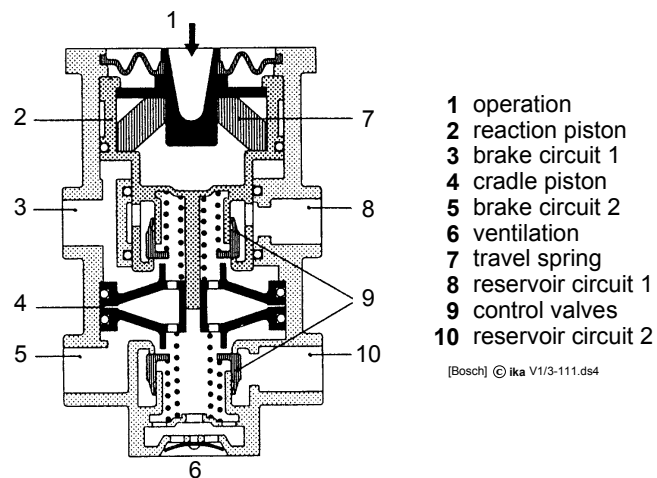


Fig. 3-111: Service-brake valve (Bosch)

Wheel-Brake Cylinder (18, 20)

Wheel-brake cylinders can be distinguished into simple diaphragm cylinders that convert air pressure into a clamping force at the brakes and combination brake cylinders with integrated springs (Fig. 3-112).

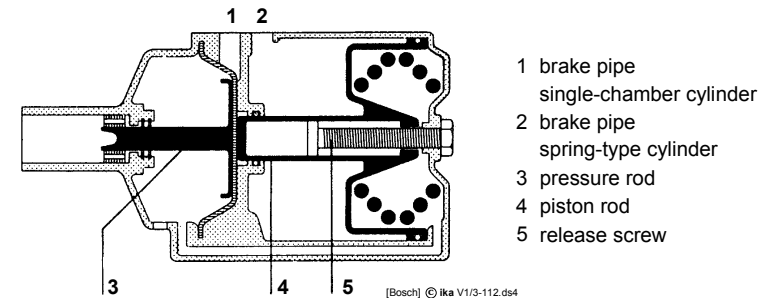


Fig. 3-112: Combination Brake Cylinder (Bosch)

The combination brake cylinder consists of a single-chamber diaphragm cylinder for the service brake and a spring-type piston cylinder for the parking brake. They are arranged one behind the other and act on a common pressure rod. They can also be operated independent of each other. When operated simultaneously, their forces add up. The central release screw allows the tightening of the spring-type cylinder so that in case of an emergency the brake can also be released in the absence of compressed air.

When compressed air flows in front of the spring-loaded piston (release of the parking brake), the piston moves against the spring force, compresses the spring and releases the brake (Fig. 3-112). When the service brake is actuated, compressed air flows behind the diaphragm and displaces the piston disk and pressure rod against the brake linkage.

3.6.6 Hybrid Braking Systems

A braking system made up of a combination of pneumatic and hydraulic systems represents a further possibility. It is most frequently used in lightweight commercial vehicles of a maximum permissible weight of approx. 6 t to 12 t.

Fig. 3-113 shows the functional diagram of an externally loaded, dual-circuit, dual-line brake system with hydraulic transmission unit.

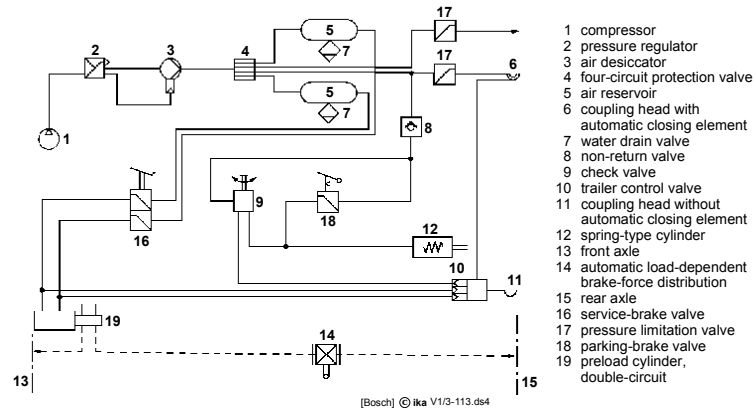


Fig. 3-113: Combined pneumatic-hydraulic system

When compared to a purely pneumatic system, it becomes obvious that a hybrid braking system generally includes the identical components.

The wheel brakes in this case, are however actuated hydraulically. This requires the use of so-called preloaded cylinders. These cylinders convert the pneumatic pressure into hydraulic pressure which in turn creates the clamping forces at the wheel brakes.

Advantages of the pneumatic braking system with hydraulic transmission unit are:

- The packaging volume available at the rim for the wheel brakes is sharply limited in lightweight commercial vehicles as a result of the small rim diameters,
- Due to the fact that heavy vehicles, particularly when driven with a trailer, require brake pressures higher than that which can be achieved using vacuum-boosted brake system in association with a realistic layout (diaphragm diameter, foot force, legal regulations), a power brake system is used,
- Lightweight trucks can be coupled to trailers with a maximum load of more than 3.5 t. Trailers weighing higher than 3.5 t cannot be equipped with an overrun brake*.

* independent mechanical braking concept used predominantly on passenger car trailers, where the generated braking force is a function of the force with which the trailer impinges against the leading vehicle.

Disadvantages of an air brake system with hydraulic transmission system are:

- The second braking system with its additional components means additional weight and thus a reduction of the payload,
- The additional components require additional maintenance expense,
- New developments allow the use of compressed-air disk brakes at all axles, also in lightweight trucks. In this way, the additional expense for hydraulic components of the braking system need not be applied.

3.6.7 Electric Brake Systems

Since the time when antilock systems and traction assistants were introduced, the electronic control of the brake force on individual wheels has become quite normal (Chapter 4.4.7).

Since these control systems generally require electrical actuators in order to transform electronic control signals, it would make sense designing the entire brake system electrically. The brake pedal then only serves as a signaling device.

Advantages of a purely electrical brake system are:

- Absence of all hydraulic or pneumatic components,
- Absence of brake force transmission since the brake force is only created on the wheel.

Disadvantages of a purely electrical brake system:

- No emergency braking function when electric control fails,
- High additional load of the electrical system of the vehicle in order to create a braking force,
- Considerably higher space required for the wheel brakes,
- Considerably higher weight of the wheel brakes.

Since electric components have approx. sixty-times the power to weight and approx. thirty-times power to volume compared to hydraulic components, the generation of the necessary braking force remains a problem. Fig. 3-114 shows the prototype of an electric disk brake.

In order to solve the problem related to brake force creation, drum brakes can be used as an alternative due to their high self-energization. Drum brakes on the other hand have essential disadvantages associated with their operating behavior.

In the medium term future, the use of purely electrical braking systems hence remains doubtful. Semi-electronic braking systems on the other hand have a higher probability of seeing application, since here the wheel brakes are actuated hydraulically while the foot pedal serves only as a signal generating device. The generation and control of hydraulic pressure takes place electrically or electronically.

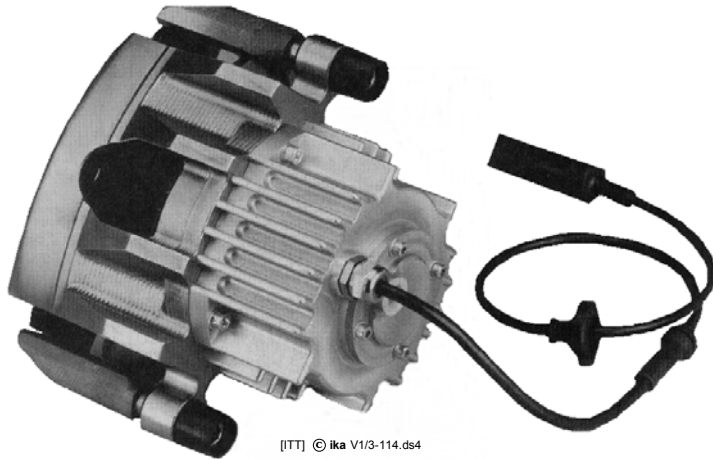


Fig. 3-114: Electric Disk Brakes (ITT)

3.6.8 Continuous Service Brakes

The effectiveness of wheel brakes is prescribed by law. When friction brakes are applied continuously in association with high braking loads, it can lead to signs of overheating which in turn results in a sharp decrease in the brake effect (fading). For heavy commercial vehicles therefore continuous largely friction-free service brakes independent of the wheel brakes are prescribed by law.

Generally three different continuous service brake types are employed:

- Engine brake,
- Hydrodynamic retarder,

- Electric retarder.

a) Engine brake

When the fuel supply is cut off, the engine provides a braking moment which is created by pumping work. The negative area of brake power in the p_i - V -diagram of the engine can be extended by two different measures:

- Exhaust Brake

In this case, the free cross-section of the exhaust manifold is partly or totally blocked by a throttle. As a result, the exhaust pressure increases, thus causing the surface area of the gas exchange loop to increase (Fig. 3-115).

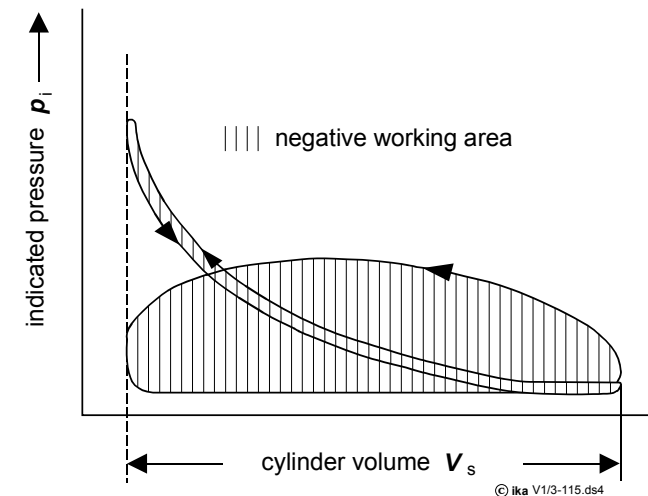


Fig. 3-115: p_i - V -diagram for exhaust brake

- Change of Valve Timing

By turning the camshaft, the exhaust valve is opened in the area of the upper dead center causing the compressed air to flow directly into the exhaust manifold. No work is done on the engine during the expansion of the compressed air. It evaporates in the exhaust pipe. The braking power can be controlled by varying the exhaust timing (Fig. 3-116).

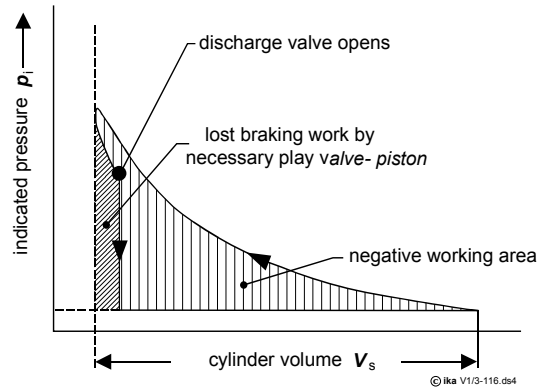


Fig. 3-116: p_i - V -diagram for motor brake by control time change

The engine brake is a cost-efficient solution and in addition can be varied over the transmission ratio (however, limited to the admissible engine speed range!).

b) Hydrodynamic Retarder

The hydrodynamic retarder (Fig. 3-117) works according to the functional principle of the *Föttinger* clutch.

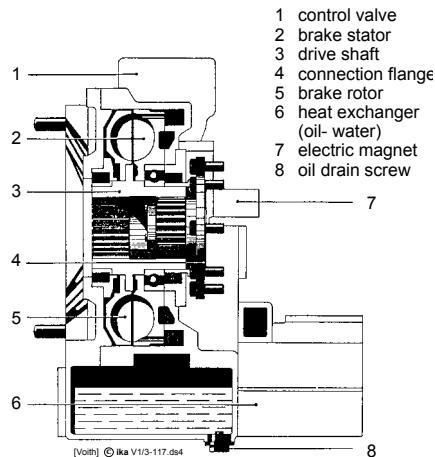


Fig. 3-117: Hydrodynamic retarder (Voith)

The mechanical energy of the drive shaft is converted over the pump wheel into kinetic energy of the fluid. This energy is converted into heat in the turbine wheel which is rigidly fixed to the vehicle body. A cooling of the working fluid hence becomes necessary. The brake moment is controlled by the volumetric efficiency of the retarder. The brake effect of retarders can be very large, in heavy trucks greater than 3,000 Nm or 400 kW.

7. The retarder is usually positioned behind the transmission in the cardan-shaft loop. New developments integrate it directly into the transmission. In automatic commercial vehicle transmissions, the existing converter can be used as a retarder. This way, the retarder characteristic can to a large extent be varied by torque conversion in the transmission.

A relatively high constructional expense opposes low weight. The magnitude of the braking torque is independent of the service temperature. The braking heat can be dissipated over an oil-water heat exchanger to the engine cooling.

c) Electrical Retarder

The Electrical Retarder, also known as the Eddy-Current Brake, shown in Fig. 3-118, functions on the electromagnetic principle. Excitation coils are mounted on a disk which forms the stator. Two braking rotors are arranged, one in front and the other behind the stator, through which the drive shaft passes. Cooling fins are present on the braking rotors for better heat dissipation. To create a braking torque, the exciting coils are fed with electric current (from the battery or alternator) in order to produce a magnetic field. If the brake rotors rotate in this field, eddy currents are induced which generate a braking torque whose magnitude depends on the exciting current of the stator coils and the speed of the drive shaft.

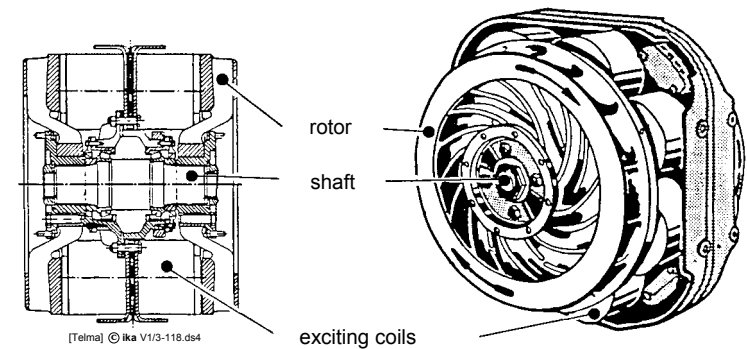


Fig. 3-118: Eddy current brake (Telma)

When it comes to the design of this retarder, a comparatively low construction expense is confronted against high weight. A trouble-free operation requires a sufficiently dimensioned battery and alternator. The achievable braking effect drops with increasing temperatures. Hence when the braking rotors are in a red-hot state for example, only half of the nominal torque is available.

d) Comparison of Continuous Service Brakes

Fig. 3-119 shows in a comparative diagram the torque characteristic of discussed continuous service brakes.

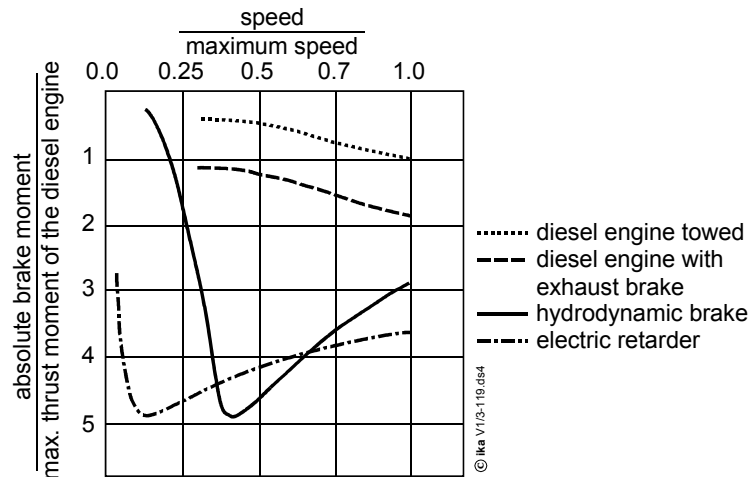


Fig. 3-119: Continuous service brake characteristics

The braking torque of a diesel engine with exhaust brake is far below that which is achieved by non-engine related continuous service brakes. It can however be converted to a higher level using the transmission. In this case however, the basic engine speeds (maximum permissible engine speed) have to be taken into account. If for example, a maximum speed of 60 km/h (37 mph) is achieved at a transmission ratio of 1:2 and at a maximum engine speed, at the transmission output twice the maximum engine braking moment is available. The retarders act as so-called secondary retarders directly on the cardan shaft. For both retarder designs, there are nearly the same maximum braking moments, however at different cardan shaft speeds. Eddy current brakes generate their maximum braking torque already at low speeds which only slightly decreases in the range of higher speeds. The maximum

braking effect of the hydraulic continuous service brake, on the other hand, is achieved in the average speed range and falls steeply particularly at low speeds.

The characteristics of retarders and eddy current brakes show the variation of the braking torque at the maximum utilisable volumetric efficiency of oil or at the maximum current input. In general, new developments allow for the constant adjustment of the braking torque to that desired by the driver. This is achieved by varying the volumetric efficiency of oil and the supply current in broad ranges, independent of the speed, using electronics.

3.7 Vibrations

The entire drive train of a conventional motor vehicle consisting of engine, clutch, transmission, cardan shaft, axle and tires, is a system which is susceptible to torsional vibrations and can be excited to critical vibrations. Fig. 3-120 shows the torsional vibration model of a drive train consisting of rotational masses and spring-damper elements.

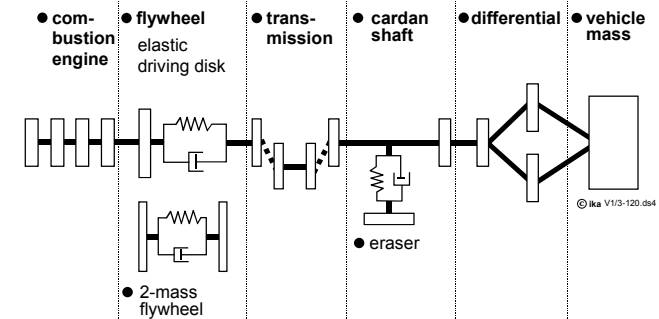


Fig. 3-120: Torsional vibration model of a drive train

Vibrations are undesirable for different reasons:

- All vehicle structures and aggregates would be loaded to a higher level mechanically and would thus have to be dimensioned more stable otherwise their life-span would be reduced,

- Vibrations in the drive train can impair driving comfort as well as driving safety (e.g. Bonanza effect*),
- vibration produces noise which imposes a strain on vehicle passengers and the environment.

The wind noise in passenger cars today has been sharply decreased by aerodynamic optimization such that noise from the engine and drivetrain in the maximum speed range is no longer masked. Meanwhile, the sensitivity to vibrations and noises in less comfort-orientated vehicles, e.g. trucks, is also growing.

The task of effectively compensating vibrations is complicated by:

- Fuel consumption and emission optimized engines which partly generate unfavorable vibrations, e.g. direct-injection diesel engines,
- New, drive train components optimized for minimum losses (e.g. light-running transmissions) which do not effectively insulate vibrations,
- Engines and drive train components are flexibly assembled based on the modular principle for economic reasons. As a consequence, specialised damping cannot be realized.

For the design and evaluation of the effects of vibration-damping, the following measures are used:

- Vibration calculations,
- Objective measurements, and
- Subjective measurements

3.7.1 Sources of Vibration

The main source of torsional vibrations in the entire drive train is the non-uniform torque output of the combustion engine. In addition, the transmission and drive shafts also act as sources of vibration.

* oscillatory transfer of driving and braking forces on the driven wheels as a result of drivetrain vibrations. Usually experienced as jerks when the vehicle is subjected to heavy load changes, i.e. sudden acceleration or release of accelerator pedal.

- Engine

The tangential forces, periodically acting on the crankshaft journal consisting of gas and mass forces of the individual cylinders accelerate or decelerate the rotational motion of the crankshaft. These variations in the angular accelerations periodically create fluctuating relative angular movements (torsional vibrations) between the individual torsional masses. Further, the mass moment of inertia of the reciprocating engine is not constant. The linearly running piston and the swinging motion of the connecting rod gives rise to a periodic vibration of the mass moment of inertia. When excited by these mass forces, the actual forces that result depend on the amplitudes of vibration.

All those speeds at which the angular frequency of the forces of excitation generated by the engine matches the natural angular frequency of the system are considered critical (resonance).

However, not all points of resonance are significant. The critical nature of vibrations can only be decided based on the deflections resulting from the respective vibrations. Mass and spring forces maintain the equilibrium in the resonance zones. The magnitude of the amplitude of vibration only depends on the effective force of excitation and the available damping.

- Transmission

The sum-total of the vibration behavior of the drive train is also continuously changed by the masses, stiffnesses, variations in tooth rigidity, and damping of the engaged gear. The excitation is essentially generated by the constantly fluctuating spring constants of the teeth during tooth engagement and by defective teeth.

- Drive Shafts

The kinematics of universal joints used in drive shafts also give rise to vibrations. In the description of the vibrational behavior of systems consisting of universal joints (e.g. cardan shaft), the transfer characteristics of universal joints play an important role. When deflected through an angle (buckling angle α), the universal joint does not transfer rotational motion uniformly. At a constant input speed ω_1 the output speed ω_2 fluctuates periodically (Fig. 3-121). The parameter-excited vibrations produced as a consequence lead to resonances.

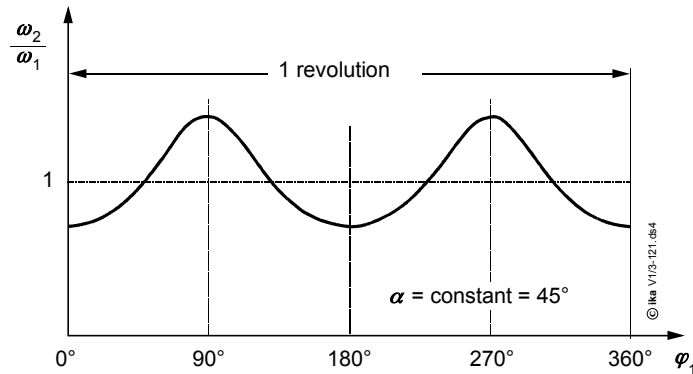


Fig. 3-121: Transmission behavior of universal joints

- Axle Suspension

Not only unevenness in road surfaces, but also imbalances in rotating parts (tires, rim, brake disk) or high concentric running tolerances of the tires excite the drive train to uncontrolled vibrations during the compression of the axle suspension.

3.7.2 Possibilities of Damping

All vehicles today should preferably use systems which ensure a decoupling at the point of separation of the engine and transmission. Possibilities to remedy the emergence of resonances on the one hand consists in raising the system damping and on the other hand in using so-called vibration eliminators. As mentioned before, the hydrodynamic clutch or converter conceptually indicates good damping characteristics. The conventional mechanical clutch however requires additional constructive measures.

Fig. 3-122 on the left shows an elastic add-on disk which acts as a torsional-spring-damper-element. In its simple conventional design, the add-on disk usually does not produce satisfying damper characteristics since the differing amplitudes of vibration cannot be absorbed evenly. In order to improve the characteristics of the elastic add-on disk, elements with multi-level friction characteristics or rubber elements are realized.

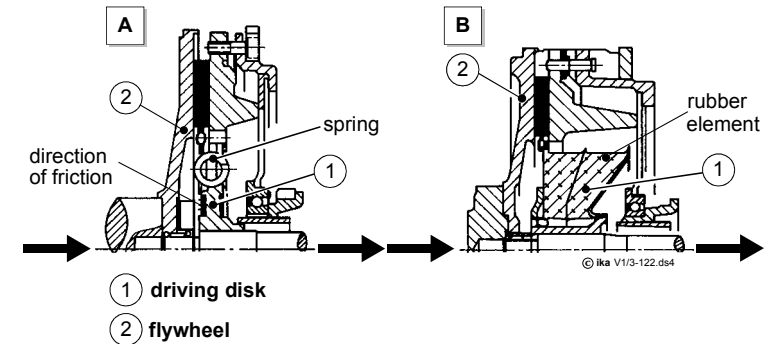


Fig. 3-122: Spring-damper-elements in a clutch disk

In critical vibrational systems or in vehicles where high demands on comfort are made, the dual-mass flywheel is employed (Fig. 3-123).

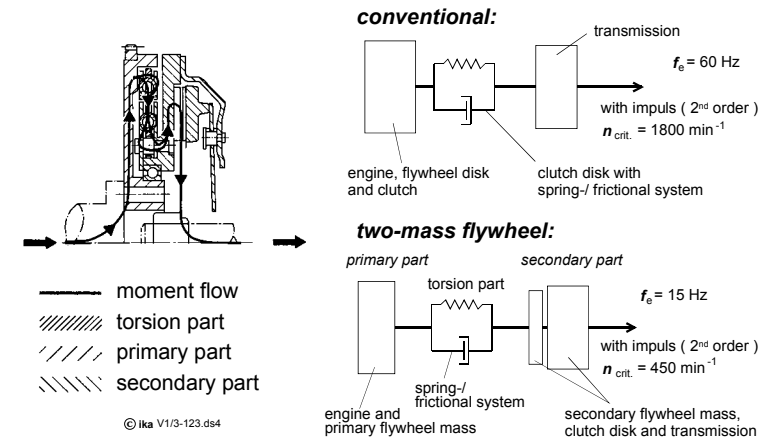


Fig. 3-123: Dual-mass flywheel

While the spring stiffness of a torsion damper in a disk clutch is varied resulting in frictional damping, in the dual-mass flywheel and in vibration eliminators, an extensive elimination of the emergence of resonance is achieved by the elastic coupling of an additional mass. Fig. 3-124 shows the example of a dual-mass flywheel.

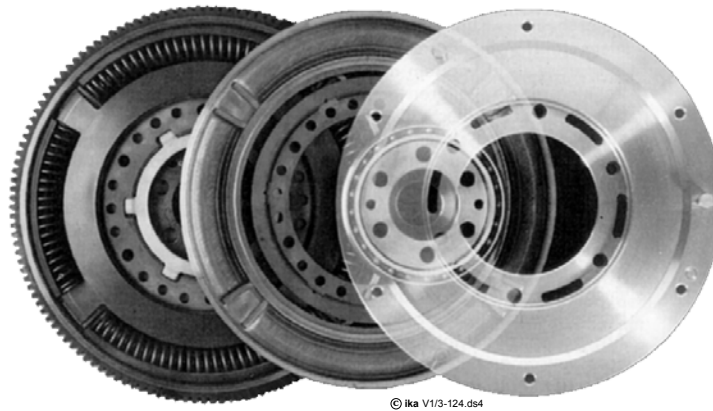


Fig. 3-124: Dual-mass flywheel

The main disadvantage of these solutions compared to an additional torsion damper in the clutch disk is most of all the higher expense. In addition, the dual-mass flywheel can only be partly used in front-wheel driven vehicles due to the limited available space (constructional width) or the additional constructive expense that has to be made which can be prove to be negative from the point of view of maintenance (clutch overhaul) and costs.

The slipping friction clutch appears as a further possibility for damping rotational vibrations. This physical principle of acceleration damping is based on the fact that a slipping clutch can only transfer only one definite torque. The torque peaks that accompany these rotational vibrations exceed the maximum transferable clutch torque and are hence not transferred by the clutch.

The so-called “dry” friction clutch which is standard fitment in series vehicles is actuated by an adjusting mechanism which is controlled via a microprocessor (Fig. 3-125).

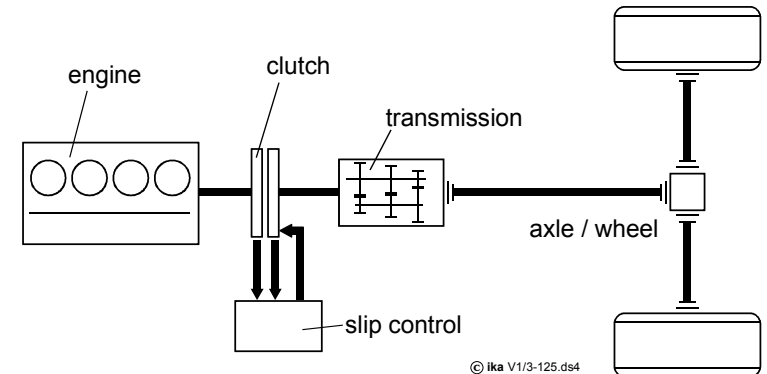


Fig. 3-125: Model of a clutch slip control

Inductive sensors detect engine output speeds and transmission input speeds. The desired variable is determined from these signals using a microprocessor, which is then used by the setting unit to realise a definite slip value by varying the clutch pressure.

When passing through load ranges in which critical rotational vibrations occur, a map-controlled damping of rotational vibrations with slip values varying between 1-5% can be realized.

In Fig. 3-126, at a critical engine speed of $1,850 \text{ min}^{-1}$, the engine output speed, transmission input speed as well as slip value are indicated as a function of time. A reduction of the high-frequency components of speed superimposed on the transmission input shaft (behind the slipping clutch) can be clearly seen.

Negative effects on the clutch life can be compensated by new linings of sintered metal, ceramics, or fiber-reinforced plastics. An increase in fuel consumption has not been determined.

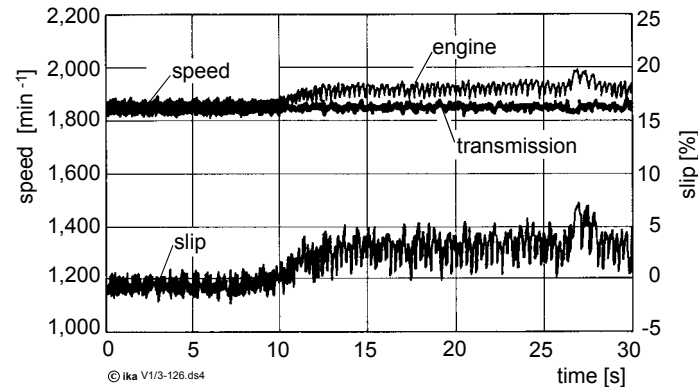


Fig. 3-126: Speed- and slip course in a clutch slip control

In order to stabilize transmission noises, vibration eliminators specially designed for the respective drive train, are employed (Fig. 3-127). They eliminate the critical vibration amplitudes.

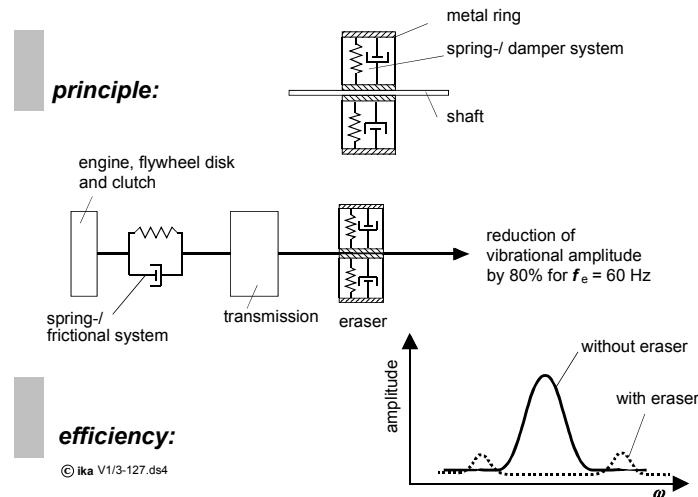


Fig. 3-127: Vibration eliminator

4 Vehicle Dynamics

After the consideration of the road resistances (and the resulting power and energy demand on the vehicle) dealt with in Chapter 2 and the drive components (and the resulting power and energy supply) considered in Chapter 3, this chapter examines the interaction of supply and demand on the entire vehicle.

Apart from driving power and fuel consumption, The different drive layouts as well as the friction-defined driving limits are also considered since they have considerable ramifications.

4.1 Driving Performance

A vehicle's driving power is characterized by the acceleration and climbing ability as well as by maximum speed. It can be determined by equating the available engine power and the power demand necessary for a particular driving state. The tire tread area of the driven wheels hence forms the interface. The comparison usually takes place in the traction force or driving power diagram.

4.1.1 Power Balance

The traction force F_X is defined as the driving force acting at the tire treads of the driven wheels. A relation for the traction force can be determined from the driving torque available at the wheel hubs, also called the hub torque M_H , as follows:

$$F_X = \frac{M_H}{r_{dyn}} \quad \text{Eq. 4-1}$$

Considering the torque loss M_{loss} arising in the drive train from the engine to the wheel hub as well as due to the transmission and differential ratios, the hub torque is determined from the engine torque M_{eng} as follows:

$$M_H = \underbrace{(M_{eng} - M_{loss})}_{M_n} \cdot i_G \cdot i_r(t) \quad \text{Eq. 4-2}$$

The difference $M_{eng} - M_{loss}$ is called net torque M_n . The net torque is only relevant from the point of view of calculations and converts the hub torque to the engine speed.

Speed, torque, transmission ratio and oil temperature have an essential influence on the torque loss. As an initial point, the torque loss can be assumed to be constant for a specific gear ratio over speed (see Fig. 4-1).

In order to estimate the magnitude of the constant torque loss, a drive train nominal efficiency η_0 is defined. This nominal efficiency describes the torque loss as a proportion of the engine torque at nominal power.

$$M_{\text{loss}} \approx M_{\text{eng}, P_0} \cdot (1 - \eta_0) = \text{constant for one gear level} \quad \text{Eq. 4-3}$$

$$M_{\text{eng}, P_0} = \frac{P_0}{2 \cdot \pi \cdot n_0} \quad (\text{engine torque at nominal power}) \quad \text{Eq. 4-4}$$

Fig. 4-1 shows the transmission efficiencies of a passenger car manual transmission measured in the third gear for different input torques. They can be approximated by straight lines.

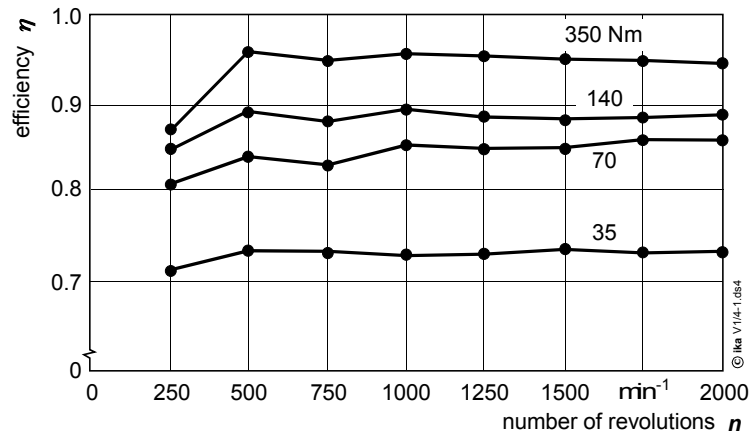


Fig. 4.1: Transmission efficiencies for different input torques

In the presented example, at an efficiency of approx. 95%, a constant torque loss of $(1-0.95) \cdot 350 \text{ Nm} = 17.5 \text{ Nm}$ for a nominal torque of 350 Nm results.

In this way, the characteristic of the net torque can approximately be determined (Fig. 4-2) from the engine's full load torque curve by a simple subtraction of the constant torque loss.

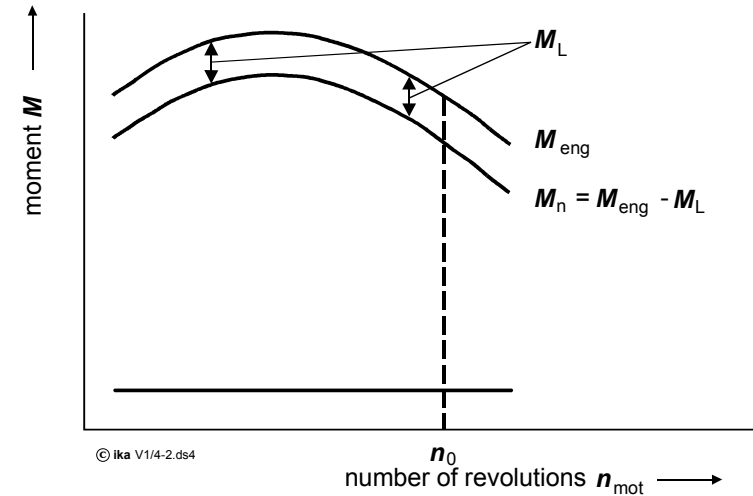


Fig. 4-2: Driving torques dependent on engine speed

From the definition of the net torque M_n , a relation for the traction force dependent on the engine speed can be defined as follows:

$$F_X = \frac{M_n \cdot i_G \cdot i_{r(f)}}{r_{\text{dyn}}} \quad \text{Eq.4-5}$$

From the speed-velocity-diagram (Fig. 3-46) already indicated in Chapter 3.4, the velocity dependent traction force available at full engine load for different transmission levels can be determined.

When the road resistances are included, e.g. at constant velocity in the plane, a so-called traction force diagram (Fig. 4-3) which compares the demand and supply, can be drawn up. Possible vehicle operating points are indicated by the intersection of the road resistance and traction force curves. In addition, the curve of maximum power (power hyperbola) is also included as a function of driving speed.

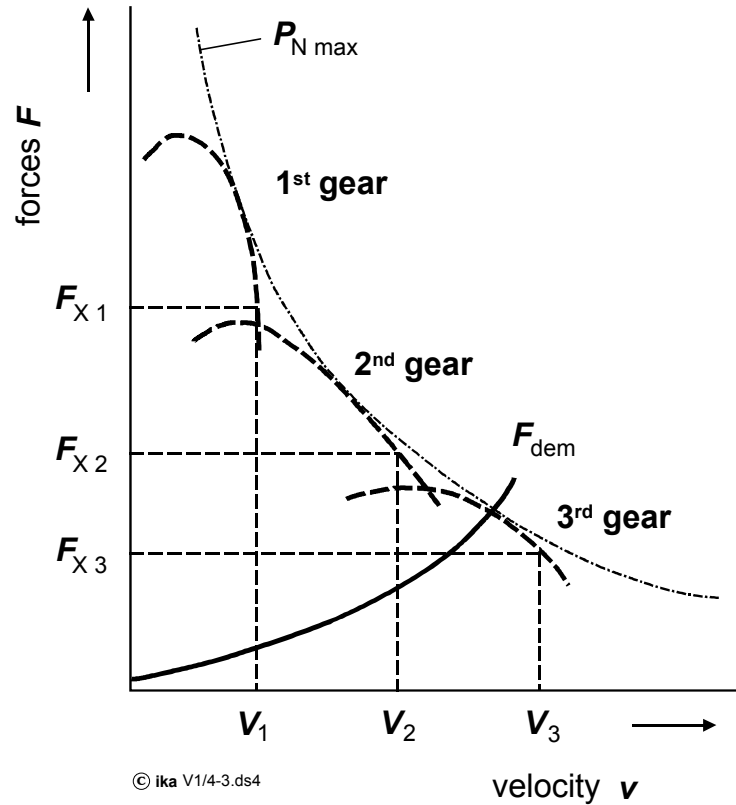


Fig. 4-3: Traction force diagram

After having multiplied the respective traction force supply and demand with the particular driving speed, the hub power P_H available at the wheels and the demand power P_{dem} necessary for driving at a constant velocity in the plane can be compared:

$$P_H = F_X \cdot v \quad \text{Eq. 4-6}$$

$$P_{dem} = F_{dem} \cdot v \quad \text{Eq. 4-7}$$

In this way, the driving power diagram arises from the traction force diagram. In the former, the line of constant maximum power is also drawn in (Fig. 4-4). The full load power curves that can be driven in different gears are also indicated.

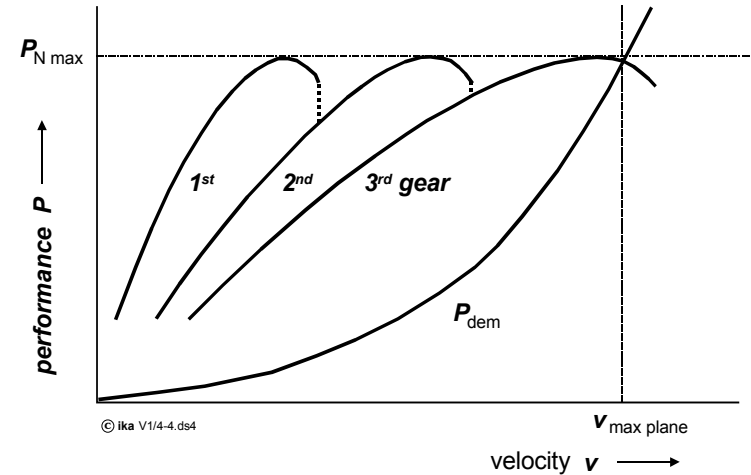


Fig. 4-4: Driving performance diagram

From the driving power as well as traction force diagrams for driving in the plane, the maximum speed v_{max} and the excess traction force $F_{X,ex}$ available for acceleration or climbing, can be determined.

4.1.2 Power-defined Climbing Ability

Using the traction force diagram, it is possible to determine in a simple way, the excess traction force $F_{X,ex}$ as a difference of the demand force F_{dem} and the available traction force F_X . The resulting characteristic of excess traction force over the driving speed is shown in Fig. 4-5.

In order to drive on an upward gradient, the climbing resistance F_{gr} has to be overcome (see Chapter 2.3). The following is applicable for the climbing resistance:

$$F_{gr} = (m_v + m_{cap}) \cdot g \cdot \sin \alpha_{gr} \quad \text{Eq. 4-8}$$

where: α_{gr} = angle of gradient

Since driving at a constant velocity is a prerequisite for the determination of the power-defined climbing ability, the entire excess traction force is available by comparing supply and demand at a constant velocity drive in the plane in order to overcome the climbing resistance.

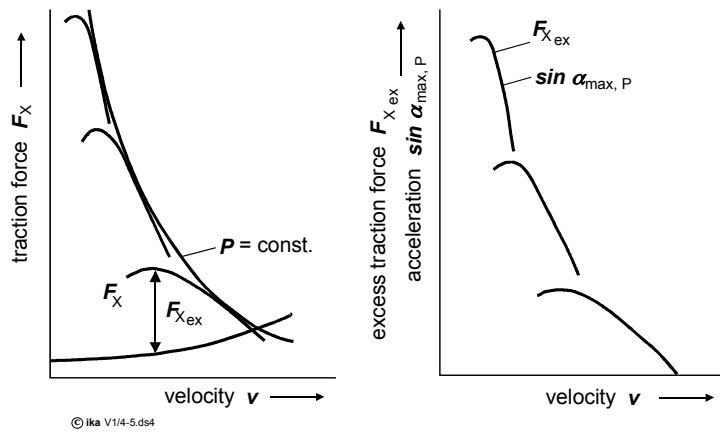


Fig. 4-5: Traction force diagram, excess traction force and climbing ability

At a prescribed speed and gear level, the maximum climbing ability defined by power results from the following relation:

$$\sin \alpha_{gr_{max,P}} = \frac{F_{X,exc}}{(m_v + m_{cap}) \cdot g} \quad \text{Eq. 4-9}$$

Thus the power-defined climbing ability is proportional to the excess traction force (Fig. 4-5).

4.1.3 Power-defined Accelerating Ability

The accelerating resistance at the maximum possible acceleration $a_{max,P}$ determined by power is overcome by the instantaneous excess traction force $F_{X,ex}$. When the translational and rotational accelerated masses are considered, the following results:

$$F_{X,exc} = (e_i \cdot m_v + m_{cap}) \cdot a_{max,P} \quad \text{Eq. 4-10}$$

$$a_{max,P} = \frac{F_{X,exc}}{e_i \cdot m_v + m_{cap}} \quad \text{Eq. 4-11}$$

where: e_i = mass factor in the gear level i

The characteristic of the power-defined accelerating ability corresponds to the characteristic of the excess traction force when the rotational accelerating resistance ($e_i = 1$) is neglected.

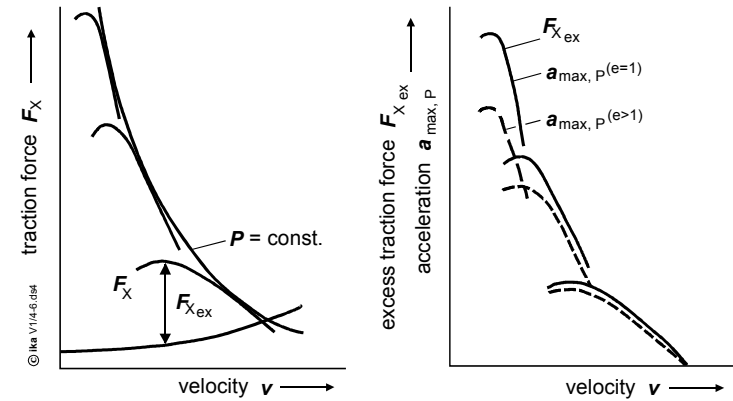


Fig. 4-6: Traction force diagram, excess traction force, and accelerating ability

As far as the rotational masses ($e_i > 1$) are concerned, due to the increasing mass factors at a falling gear number, the characteristic shown in Fig. 4-6 indicated by dotted lines, results. In order to achieve a high climbing ability even at maximum load in commercial vehicles, a very high ratio has to be realized in the smallest gear. In this case the value of the mass factor can be so high that the accelerating ability in the first gear is smaller than in the second. Fig. 4-7 shows the characteristic of accelerating ability in such an extreme case.

In order to achieve maximum acceleration, gears have to be first shifted at those speeds in which the next higher gear allows for a larger acceleration than the previously engaged gear or when the maximum engine speed is reached (Fig. 4-7).

Due to the linear relationship, the maximum accelerating ability in the individual transmission levels coincides with the maximum excess traction force.

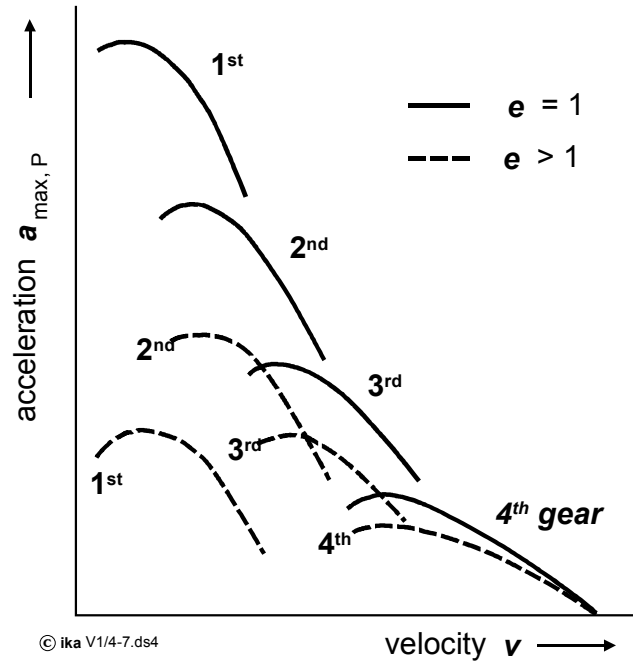


Fig. 4-7: Influence of the mass factor on the power-defined accelerating ability in a highly reduced commercial vehicle

In comparison, the maxima of the excess power are shifted towards higher speeds since:

$$P_{\text{exc}} = F_{X,\text{exc}} \cdot v = (e_i \cdot m_v + m_{\text{cap}}) \cdot a_{\text{max},P} \cdot v \quad \text{Eq. 4-12}$$

This means that for a constant acceleration, an excess power proportional to the velocity is required.

Fig. 4-8 shows the characteristics of accelerating ability and excess power.

The relationship between the power-defined accelerating ability and the maximum climbing ability can be determined by the relations:

$$F_{X,\text{exc}} = (e_i \cdot m_v + m_{\text{cap}}) \cdot a_{\text{max},P} \quad \text{Eq. 4-13}$$

$$F_{X,\text{exc}} = (m_v + m_{\text{cap}}) \cdot g \cdot \sin \alpha_{\text{gr,max},P} \quad \text{Eq. 4-14}$$

By equating the excess traction forces:

$$a_{\text{max},P} = \frac{m_v + m_{\text{cap}}}{e_i \cdot m_v + m_{\text{cap}}} \cdot g \cdot \sin \alpha_{\text{gr,max},P} \quad \text{Eq. 4-15}$$

A proportional dependence between accelerating and climbing ability which depends on the gear due to the influence of rotational masses (in Equation 4-15 considered by e_i) results.

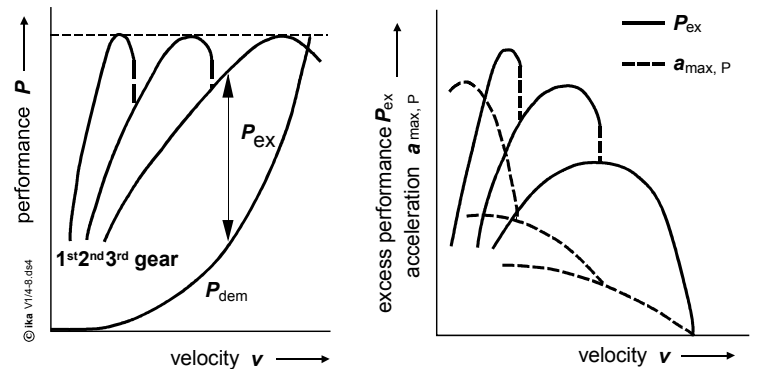


Fig. 4-8: Driving power diagram, excess performance, and accelerating ability ($e = 1$)

4.1.4 Influence of Transmission Layout

Apart from the road resistance parameters and the engine characteristics of the vehicle which are characterized by the form and position of the full-load curve in the engine map, the layout or the control strategy of the transmission has an enormous influence on driving performance.

The required maximum speed v_{max} generally determines the smallest ratio of the entire drive train ($i_{\text{ent}} = i_G \cdot i_{r(f)}$). If a stepped transmission with a direct final gear ($i_G = 1$) should be used, the axle ratio has to be adjusted in such a way that the required maximum speed can be reached over the entire ratio. This on the other hand, has an influence on the ratios of the remaining transmission levels (see also Chapter 3.4). If,

however, the drive axle ratio ($i_{r(f)}$) is prescribed, the required maximum speed determines the smallest transmission ratio.

The maximum speed is achieved, if the power supply and the power demand are at equilibrium. Three possible layouts are then available, as presented in Fig. 4-9:

- Layout on $v_{\max, \text{theor.}}$,
- Overspeeding layout,
- Underspeeding layout.

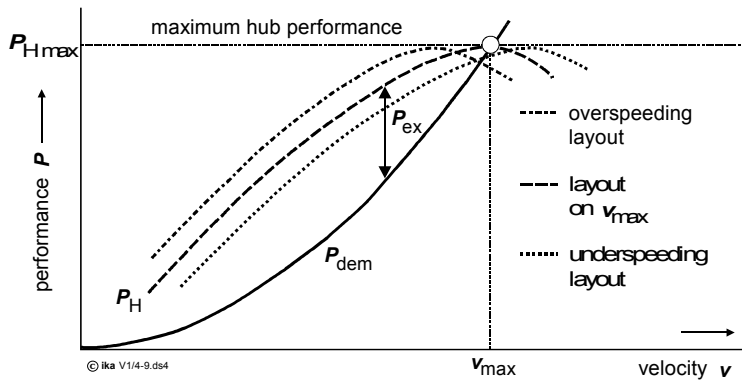


Fig. 4-9: Transmission layouts

- Layout on $v_{\max, \text{theor.}}$

The equation between power demand and supply is reached at the nominal speed n_0 and hence at the maximum hub power. With this layout, the theoretically largest possible speed is realized. In the range below the maximum speed however, only a relatively small excess power is available for acceleration, climbing, or headwind.

- Overspeeding Layout

The nominal speed is exceeded before the point of equilibrium. The engine speed at the maximum velocity is larger than the nominal speed. In this way, a large amount of excess power is available in the maximum speed range. The theoretical maximum speed is however not achieved. The high engine speed at $v_{\max}$ causes a higher driving noise and greater wear. The engine operating point which is driven in this case is generally in the range of high fuel consumption.

The fourth gear of common 4-gear transmissions, 5-gear transmissions with the fifth gear in overdrive and the largest gear of sporty 5-gear transmissions in passenger cars are mostly designed based on the overspeeding layout.

- Underspeeding Layout

The maximum speed in this case is achieved before the nominal speed, meaning at a lower engine speed. The theoretical maximum speed is not achieved. It is true that noise level, fuel consumption, and wear are lower, but since the excess power is smaller than the other two layouts, the maximum speed can only be reached slowly and drops steeply when upward gradients or headwinds are encountered.

A combination of the advantages of overspeeding and underspeeding layouts can be realized in 5-gear transmissions by designing the fourth gear based on overspeeding and the fifth gear as E-gear on the underspeeding layout. In this case, the maximum speed can be the same in both gears. Generally the maximum speed is lower in the fifth gear than in the fourth gear. Fig. 4-10 shows the characteristics of hub power in both variants.

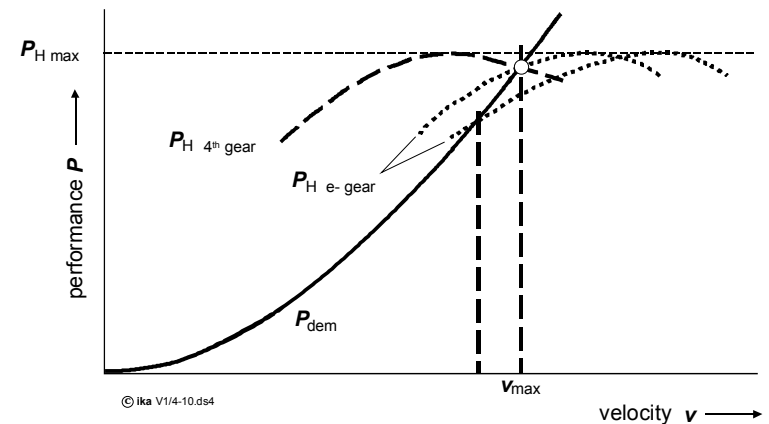


Fig. 4-10: Layout of the 5th gear as E-gear

The maximum speed can also be determined by a speed limiter, as is shown in Fig. 4-11 using the example of a diesel engine.

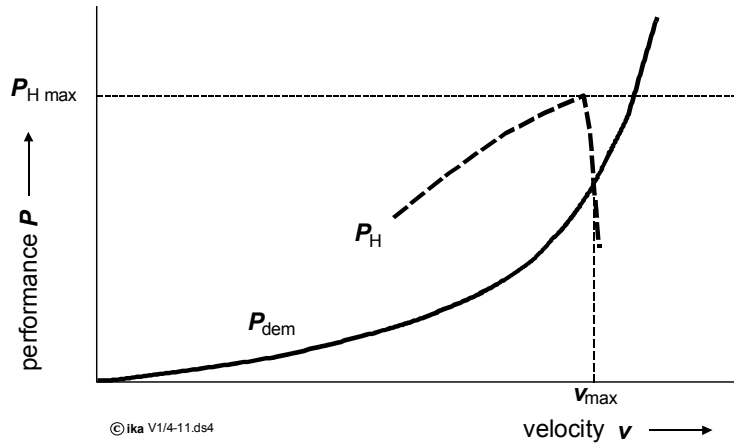


Fig. 4-11: Effect of a speed limiter (diesel engine)

In commercial vehicles the curves of road resistance are dispersed over a wide range (Fig. 4-12) due to the large variations in laden and unladen weights and the large influence of the vehicle weight on the total road resistance. Hence a basic layout of the maximum speed corresponding to the above-mentioned cases is not possible. The rear-axle layout is thus chosen in such a way that at the most frequently driven speed, as good a total efficiency as possible can be realized. In long-distance trucks for example, the most frequent speed is the maximum speed of 80 km/h (50 mph) prescribed by law. Fig. 4-12 shows a typical layout for a truck.

The unladen truck can accelerate in the plane up to a speed limit of approx. 120 km/h (75 mph). When completely loaded, a speed of approx. 105 km/h (65 mph) can be achieved at slight underspeeding. The direct gear cannot cope with the gradient represented by the road resistance curve on the left. Using a smaller gear however, the full engine power can be used and the theoretical maximum speed of approx. 85 km/h (53 mph) be reached.

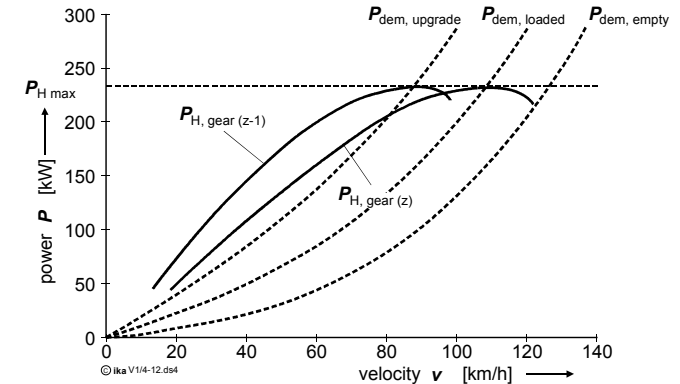


Fig. 4-12: Transmission layout in a truck

The total transmission layout has to be examined from the point of view of the accelerating and climbing ability in the individual gears. The excess traction forces and powers resulting for the respective gear ratios have to sufficiently meet the driving demands.

The achievable driving powers can be checked by drawing in the demand curves for climbing or acceleration in the traction force or power diagram and by analyzing the excess traction forces and powers available in the gears. In this way, large gaps in power can be avoided, which would require frequent shifting from the driver.

Fig. 4-13 shows the exemplary demand curves for different gradients.

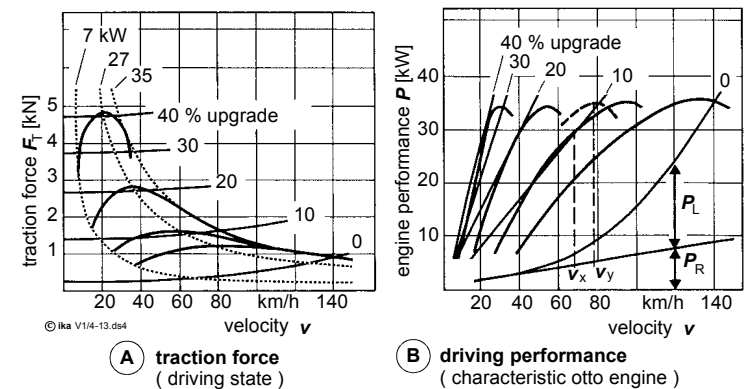


Fig. 4-13: Driving power and traction force diagram with demand curves for different gradients

The combustion engine can realize the required power at different operating points (part load). Thus the choice of the shifting point in stepped transmission and the control strategy in CVTs has a large influence on the driving performance.

In order to achieve high driving performance, a maximum excess traction force is necessary. As far as this is concerned, the points of intersection of the characteristics of the excess traction force in the respective gears represent the correct shifting point between neighboring transmission levels in regard to the optimal accelerating ability in stepped transmissions. These points of intersection are shifted towards lower speeds as a result of the mass factor (Fig. 4-14).

The optimum shifting points in reference to the maximum climbing ability in a constant velocity drive, lie at the points of intersection of the characteristics of excess traction force. Thus they lie within the points of intersection of the power curves independent of the mass factor.

In contrast to stepped transmissions, CVTs offer the possibility to freely choose the engine operating point for a prescribed power demand. With a corresponding control strategy, the engine can be operated constantly at the point of maximum power output. The vehicle has an optimum climbing and accelerating ability.

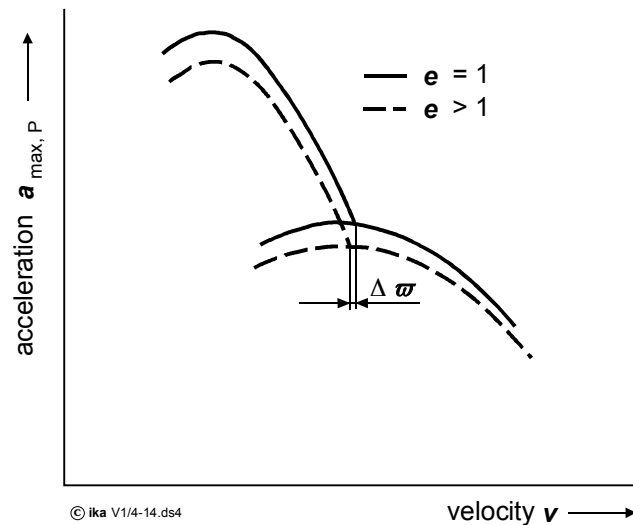


Fig. 4-14: Influence of the mass factor on the optimum shifting points when accelerating

4.1.5 Total Braking Force and Braking Potential

When a vehicle is subjected to retardation, the instantaneous acceleration is negative. In order to represent the braking intensity quantitatively, the deceleration z which refers to the ratio of the deceleration a_B to the acceleration due to gravity g is introduced:

$$z = \frac{|a_B|}{g} \quad \text{Eq. 4-16}$$

The total braking force, which decelerates the vehicle, considering the road resistances and the sum of all the forces acting on the vehicle in case of a coupled engine, is given by:

$$F_{B_{\text{ret}}} = -F_a = F_R + F_{\text{drag}} + F_{\text{gr}} + F_{\text{eng}_{\text{thrust}}} + F_{\text{perm brake}} + F_{B_{\text{wheel}}} \quad \text{Eq. 4-17}$$

The required wheel brake force in order to achieve a deceleration a_B is then given by:

$$F_{B_{\text{wheel}}} = (e_i \cdot m_v + m_{\text{cap}}) \cdot a_B - (F_R + F_{\text{drag}} + F_{\text{gr}} + F_{\text{eng}_{\text{thrust}}} + F_{\text{perm brake}}) \quad \text{Eq. 4-18}$$

- Stability in downward Gradients

From the qualitative description in Fig. 4-15, it can be noticed that in the downward gradient, a smaller traction force demand when compared to driving in the plane results due to the force F_{gr} acting in the direction of the negative slope, i.e., in the driving direction. According to the downward gradient, a brake force has to be created in the lower and average speed range, in order to keep a constant driving speed. In case of heavy commercial vehicles, continuous service brakes are used for braking in downward gradients. Their characteristics, in this case an engine brake and electric retarder, are also included in the diagram. If the magnitude of the brake force of the continuous service brake is larger than the resulting force demand $F_R + F_{\text{drag}} + F_{\text{gr}}$, a stable driving condition is said to be achieved.

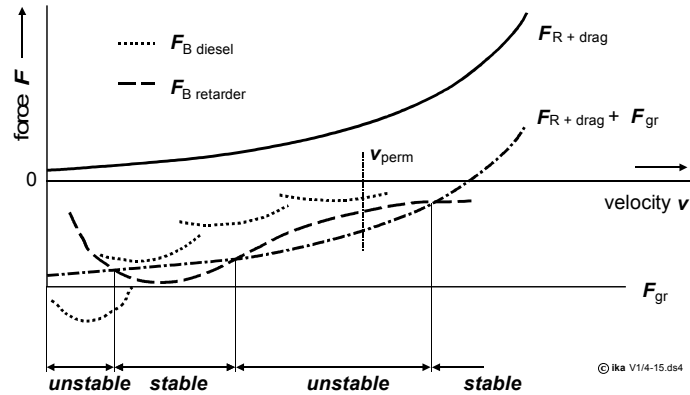


Fig. 4-15: Stable driving conditions in downgrade drive with engine brake and electric retarder

The stability ranges in Fig. 4-15 refer to a driving condition in which the electric retarder is exclusively used. In the range of the maximum speed v_{perm} prescribed by law, a higher constant speed can be achieved with the electric retarder than with the engine brake. In the lower speed range however, a stable condition can be achieved only using the engine brake.

4.2 Fuel Consumption

In order to realize the described driving performance, the propulsion unit has to be fed with energy. Liquid fuel forms the energy source in the combustion engine used in nearly all vehicles today. The fuel consumption of a vehicle is determined by:

- Track fuel consumption in l/100 km
- Temporal fuel consumption in l/h

4.2.1 Determination of Fuel Consumption

- a) Theoretical determination of Fuel Consumption

On the basis of initial drive data it is possible to make relatively precise estimations of the fuel consumption mathematically, in the early development stages of a vehicle. In addition, parametric variations can be undertaken without additional expense.

In order to undertake this however, an engine map which contains the lines of a constant specific fuel consumption b_e (Fig. 4-16) is a prerequisite.

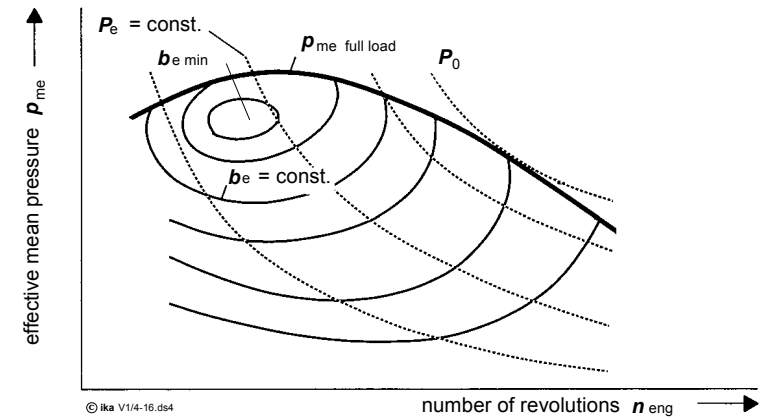


Fig. 4-16: Schematic engine map

From the force demand at the speed which is to be analyzed (constant velocity in the plane):

$$F_{dem} = (m_v + m_{cap}) \cdot g \cdot f_R + c_w \cdot A \cdot \frac{\rho_a}{2} \cdot (v \pm v_w)^2 \quad \text{Eq. 4-19}$$

considering the torque loss M_{loss} , the engine torque demand $M_{eng_{dem}}$ can be determined according to the relation:

$$M_{eng_{dem}} = \frac{F_{dem} \cdot r_{dyn}}{i_G \cdot i_{r(f)}} + M_{loss} \quad \text{Eq. 4-20}$$

The engine mean effective pressure p_{ae} is then given by:

$$p_{ea} = \frac{2 \cdot \pi \cdot M_{eng_{dem}}}{V_S \cdot i} \quad \text{Eq. 4-21}$$

$$p_{ea} = \frac{2 \cdot \pi}{V_S \cdot i} \cdot \left(\frac{F_{dem} \cdot r_{dyn}}{i_G \cdot i_{r(f)}} + M_{loss} \right) \quad \text{Eq. 4-22}$$

The engine speed can be calculated from the driving speed as follows:

$$n_{eng} = \frac{v \cdot i_G \cdot i_{r(f)}}{2 \cdot \pi \cdot r_{dyn}} \quad \text{Eq. 4-23}$$

The corresponding operating point in the engine map can now be clearly determined. For this operating point, the specific fuel consumption can be read from the map. Using the power demand and the fuel density the temporal fuel consumption can be determined:

$$B_{temp} = \frac{b_e \cdot P_e}{\rho_{fuel}} = \frac{b_e \cdot p_{ea} \cdot V_S \cdot n_{eng} \cdot i}{\rho_{fuel}} \cdot \text{conversion factor} \quad \text{Eq. 4-24}$$

The track fuel consumption results from the relation:

$$B_{tr} = \frac{B_{temp}}{v} = \frac{b_e \cdot p_{ea} \cdot V_S \cdot n_{eng} \cdot i}{\rho_{fuel} \cdot v} \cdot \text{conversion factor} \quad \text{Eq. 4-25}$$

where:	B_{temp} :	temporal fuel consumption	[l/h]
	B_{tr} :	track fuel consumption	[l/km]
	b_e :	specific fuel consumption	[g/kwh]
	P_e :	effective power	[kW]
	ρ_{fuel} :	fuel density	[g/l]
	p_{ea} :	mean effective pressure	[bar]
	V_S :	displacement	[cm ³]
	n_{eng} :	engine speed	[min ⁻¹]
	v :	vehicle speed	[km/h]
	i :	power strokes per crankshaft revolution	[-]
		($i=0.5$ for 4-stroke, $i=1$ for 2-stroke)	
	conversion factor:	1/6 10 ⁻⁵ (to adjust the units)	

If the fuel consumption for a driving cycle is to be calculated, apart from the steady-state phases, the transient driving conditions have also to be taken into account. Similar to the determination of the energy demand, the speed characteristic is divided into small time intervals. In these intervals, the acceleration can be assumed as constant and the actual driving speed can be averaged. The traction force demand in the respective intervals for a drive in the plane is then given by:

$$F_{dem_i} = (m_v + m_{cap}) \cdot g \cdot f_R + c_w \cdot A \cdot \frac{\rho_a}{2} \cdot (v_i \pm v_W)^2 + (e_i \cdot m_v + m_{cap}) \cdot a_i \quad \text{Eq. 4-26}$$

When considering the losses in the drive train, we obtain the effective fuel consumption from the engine map. The amount of fuel consumed in each interval can be determined by the time interval Δt_i :

$$m_{fuel_i} = b_{e_i} \cdot p_{ea_i} \cdot V_S \cdot n_{eng} \cdot i \cdot \Delta t_i \quad \text{Eq. 4-27}$$

The track fuel consumption in order to drive through such a cycle results from the sum of the individual fuel consumption values:

$$B_{tr} = \frac{1}{\rho_{fuel} \cdot s} \cdot \sum_{i=1}^n m_{fuel_i} \quad \text{Eq. 4-28}$$

Since the time intervals selected have to be as small as possible, a fuel consumption calculation of this sort becomes meaningful only when supported by a computer. Sufficiently precise results can then be achieved. One of the limitations encountered here results from the fact that only engine maps for stationary operating points are included, hence leading to the results for transient driving conditions being only partly (although with a good approximation) applicable.

b) Determination of Fuel Consumption by Measurement

The fuel consumption measurement is the basis of all fuel consumption standards and limits. In order to undertake measurements, a vehicle prototype in which parametric variations are associated with relatively high costs, is necessary.

Fuel consumption measurements can be carried out on a roller test bench, driving tests on test tracks or on public roads. The roller test bench has advantages from the point of view of better reproducibility of the test conditions and shorter implementation times. In addition, the required measurement devices do not necessarily have to be in the vehicle.

The measurement methods can be distinguished into continuous flow measurement and the measurement of fuel consumption over a definite distance covered or in a definite time.

In the latter, the track fuel consumption results from the gravimetric measurement of the consumed fuel mass as follows:

$$B_{tr} = \frac{m_{fuel}}{\rho_{fuel} \cdot s} \quad \text{Eq. 4-29}$$

In a volumetric measurement the thermal expansion of the fuel has to be considered:

$$B_{tr} = \frac{V_{fuel} \cdot (1 + \alpha_{fuel} \cdot (T_0 - T_{fuel}))}{s} \quad \text{Eq. 4-30}$$

where: α_{fuel} = coefficient of thermal expansion (gasoline, diesel = 0.001 K^{-1})

T_0 = reference temperature ($T_0 = 293 \text{ K}$ after DIN 70030=

T_{fuel} = fuel temperature

The gravimetric or volumetric measurement method, though being very accurate, cannot provide instantaneous values, which are significant from the point of view of a closer examination of the consumption behavior, e.g. in the steady-state driving conditions.

If the instantaneous consumption values are required, flow measurement is used. Due to low fuel flows being common in vehicles, high technical demands are made on the turbine, positive-displacement pump, or heated thermo element measuring systems.

A third method determines the converted fuel mass from the compounds in exhaust emission over a carbon balance. It is mainly used in connection with emission studies in which exhaust is collected and analyzed. This method is very accurate but is also associated with high expense.

4.2.2 Fuel Consumption Norms

Fuel consumption norms determine the procedures for the (metrological) determination of the fuel consumption. For the vehicle buyer they provide a basis for comparing different vehicle models.

Based on the driving style, road and traffic conditions, environment and vehicle condition and fitment, in practice, fuel consumption values that vary from the values determined by the standards will arise.

Until 1996 the usual manufacturer's data of the fuel consumption of passenger cars corresponded to the DIN standard 70030 part 1. The inspection contained the following points:

Related to vehicle:

- In series
- Engine run-in (min. 3,000 km (1,864 miles))
- Service temperature

Related to Inspection:

- Standard drive with driving cycle according to ECE-R 15 ("ECE-cycle")
- Constant drive at 90 km/h (56 mph)
- Constant drive at 120 km/h (75 mph) (if $v_{max} > 130 \text{ km/h}$ (81 mph))

Additional conditions:

- Fuel according to DIN 51600 (Otto, leaded)
- DIN 51607 (Otto, unleaded)
- DIN 51601 (diesel)
- Environmental conditions $P_0 = 1 \text{ bar}$; $T_0 = 293 \text{ K}$
(air density may deviate by +/- 7.5% from this condition)
- Wind speed $v_W < 3 \text{ m/s}$

The fuel consumption values are then presented as follows:

City cycle :	11.5l/100 km
90 km/h :	7.0l/100km
120 km/h :	9.2l/100 km

or according to the standards:

Consumption DIN 70030-1 - $11.5 \times 7.0 \times 9.2$.

Fig. 4-17 presents the driving cycle which is applicable to city traffic.

The driving cycle consists of 22 phases (idling, acceleration, constant drive, deceleration). Driving through this cycle four times results in a cycle length of 4.052 km (2.519 miles), a test duration of 13 minutes, and an average speed of 18.7 km/h (11.6 mph).

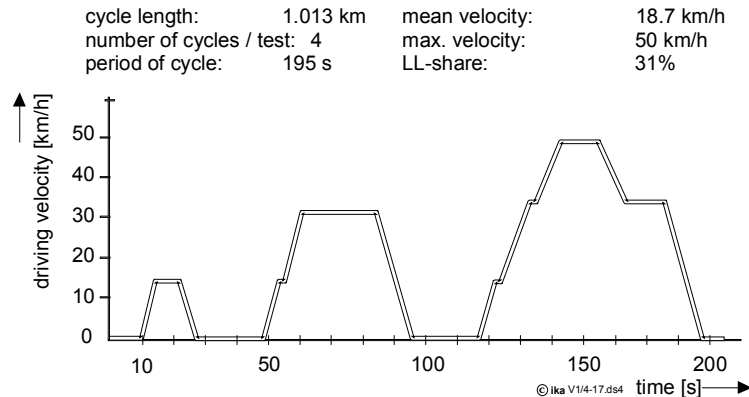


Fig. 4-17: ECE-city driving cycle for manual transmission vehicles

The determination of the standard fuel consumption according to the above mentioned method has been increasingly criticized for the strong emphasis given to constant drives which was considered unrealistic. Furthermore, the determined values were below those realized in usual drives. Fig. 4-18 compares consumption values in a standardized 1/3-mix and consumptions of the same motor vehicles ascertained in tests by experts. On average they are about 20% higher.

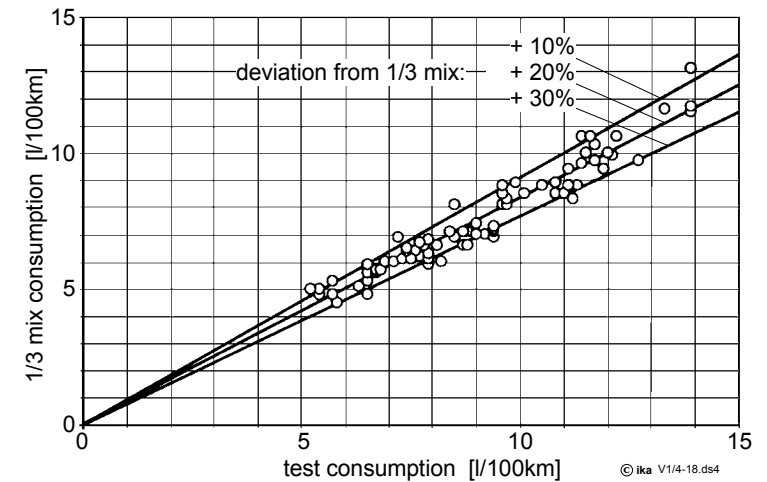


Fig. 4-18: Comparison 1/3-mix consumption with test values (ATZ)

According to the EC standard ECE-R 83/02, the driving cycle is also includes an upper speed range. The new EC driving cycle (NEFZ) therefore consists of two parts (Fig. 4-19). In this case the constant drives are dropped.

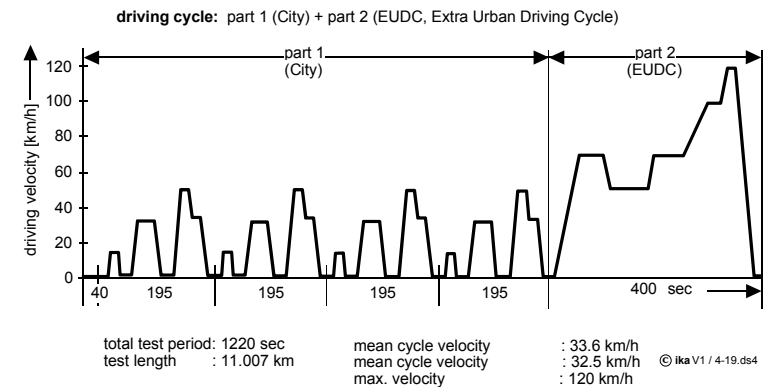


Fig. 4-19: NEFZ-driving cycle

As before, the first part consists of 4 ECE-city-driving cycles. It is started with a cold engine. Only 40 s are available for the warm-up phase. Part 2 (outer city driving cycle) consists of 13 phases and results in a cycle length of 6.955 km (4.32 miles) and a test duration of 400 seconds with a resulting average speed of 56.9 km/h

(35.36 mph). The cycle has a total length of 11.007 km (6.8 miles) with a duration of 1220 s, at an average speed of 32.5 km/h (20.2 mph).

The values determined according to the new EC standard are generally about 10% above the old DIN-values.

The fuel consumption for commercial vehicles are determined according to the DIN standard 70030. The indicated value refers to the fuel consumption of the vehicle half-loaded vehicle at a constant speed dependent on the vehicle. The speed is indicated as well.

4.2.3 Fuel-Consumption Limits

In the USA, within the scope of measures implemented for the energy budget, a limitation of the fuel consumption is prescribed by law starting with the model year 1978. The regulations limit the average consumption of a particular model year from a specific manufacturer. Fig. 4-20 shows the threshold values for years 1980 to 1990. An tightening of the threshold values for fuel consumption is not planned for future legislation. Within the scope of a “credit program”, automobile manufacturers are also allowed to exceed the existing consumption limits, provided it is compensated by the introduction of “more economical” and “cleaner” vehicles, driven with alternative fuels.

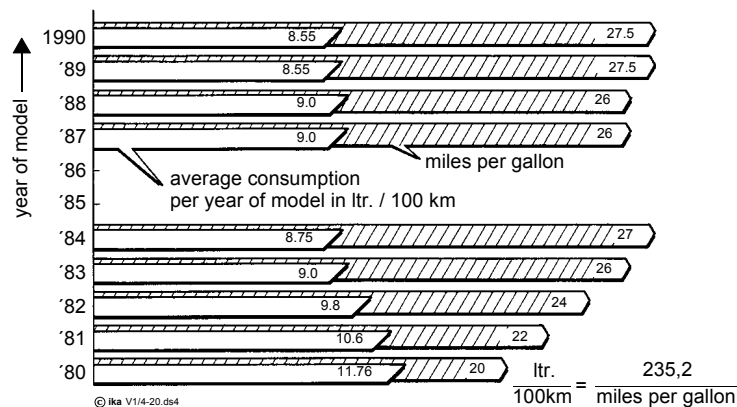


Fig. 4-20: Targeted consumption values according to the US Energy Act

The test cycle FTP-75 is used in order to determine the fuel consumption values. One can distinguish between a city-cycle and a highway-cycle that are each driven on the test bench (Fig. 4-21, Fig. 4-22).

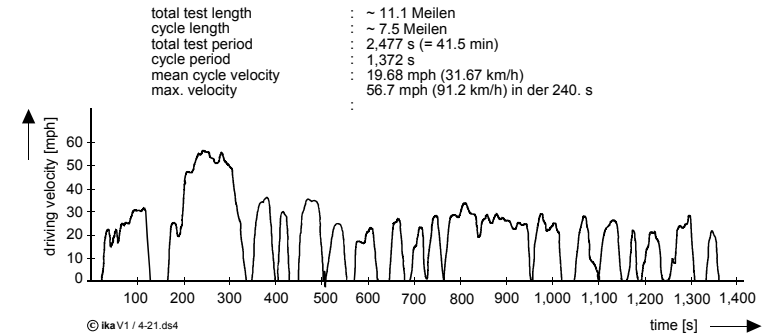


Fig. 4-21: Test cycle FTP 75, city-cycle

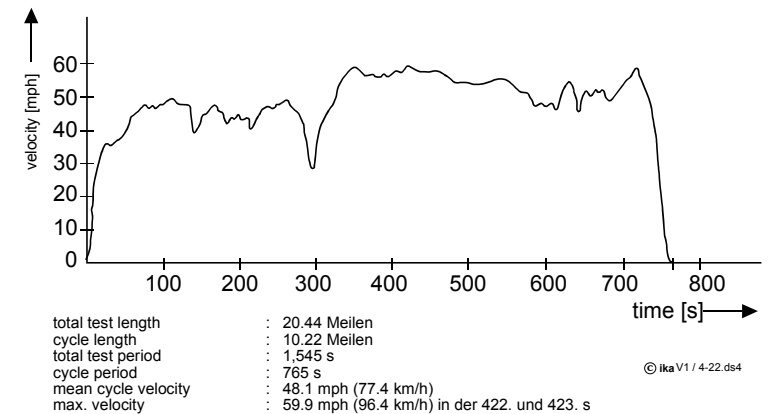


Fig. 4-22: Test cycle FTP 75, highway-cycle

In the fuel-economy of a vehicle, it is distinguished between the consumption values:

MpG_h (miles per Gallon) = highway fuel economy
 MpG_c = city fuel economy
 MpG_{c/h} = combined fuel economy

where:
$$\text{MpG}_{c/h} = \frac{1}{\frac{0.55}{\text{MpG}_c} + \frac{0.45}{\text{MpG}_h}}$$
 Eq. 4-31

The fuel measurement takes place indirectly by undertaking a carbon balance of the exhaust compounds (HC, CO, CO₂). The following is applicable here:

$$\text{MpG} = \frac{x}{0.866 \cdot \text{HC} + 0.429 \cdot \text{CO} + 0.273 \cdot \text{CO}_2} \quad \text{Eq. 4-32}$$

where: x = 2,424 for gasoline
 x = 2,778 for diesel
 HC, CO, CO₂ = measured exhaust in gram per mile

From the fuel economy of each type of car assessed according to the number of units sold, the average annual consumption of a particular model from a manufacturer is determined as illustrated in Fig. 4-23.

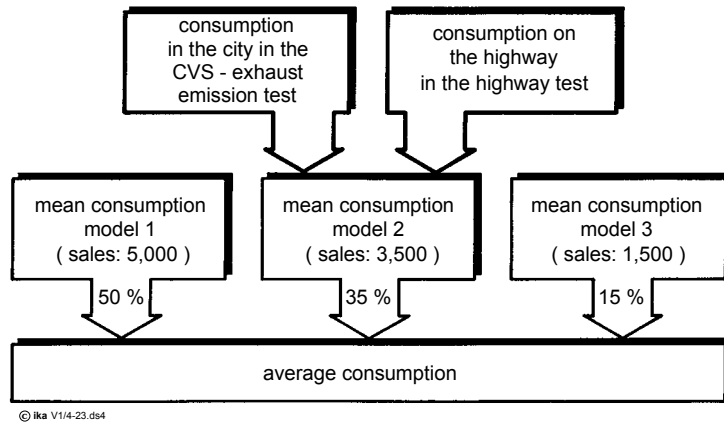


Fig. 4-23: Determination of the fleet consumption of a manufacturer

A statutory limitation of the fuel consumption is also being discussed in Europe. However, a uniform test concept does not exist yet.

4.2.4 Reduction in Fuel Consumption

The fuel consumption of a vehicle can be described by known relations. It is:

$$\begin{aligned} B_{tr} &= \frac{b_e}{\rho_{fuel}} \cdot \left(F_{dem} + \frac{M_{loss} \cdot i_G \cdot i_{r(f)}}{r_{dyn}} \right) \\ &= \frac{b_e}{\rho_{fuel}} \cdot \frac{1}{\eta_{train}} \cdot F_{dem} \end{aligned} \quad \text{Eq. 4-33}$$

where: η_{train} = drive train efficiency

When the expressions for force demand deduced in Chapter 2 are included, the fuel consumption for a general driving situation is given by:

$$B_{tr} = \frac{b_e}{\rho_{fuel}} \cdot \frac{1}{\eta_{train}} \cdot [(m_v + m_{cap}) \cdot (f_R \cdot \cos \alpha_{gr} + \sin \alpha_{gr}) + c_w \cdot A \cdot \frac{\rho_a}{2} \cdot v^2 + (e_i \cdot m_v + m_{cap}) \cdot a] \quad \text{Eq. 4-34}$$

With help of this equation, the different influencing parameters can be seen and thus the possibilities for a reduction in fuel consumption can be deduced:

- Traffic-related measures such as traffic guiding systems, traffic light circuitry or driver training, basically influence the characteristics of vehicle acceleration and speed.
- Vehicle-specific road resistance parameters such as vehicle mass, road resistance, coefficient of air resistance and main bulkhead cross-section have to be minimized considering the requirement criteria safety, ergonomics, economy, or comfort.
- The mechanical drive energy should ideally be entirely available at the wheels. This implies a high drive train efficiency which also takes into account losses due to aggregates (air conditioning, steering boost, power brakes).

The specific fuel consumption, is the amount of fuel utilised relative to the mechanical power produced and is a measure for the quality of the energy conversion in the combustion engine. In the engine map, lines of constant specific fuel consumption form so-called "shell curves".

As the lines of constant power indicate, the combustion engine can achieve a certain power output at different operating points. Thus the layout of the transmission and axle ratio as well as the gear shifting point in stepped transmissions or the control strategy in the case of infinitely variable torque converters have a large influence on the fuel consumption.

If the road resistance curves dependent on speed are introduced into the engine map, while taking into consideration the transmission ratio and the losses in the drive train, the indicated demand curves result for constant velocity drives in plane.

Fig. 4-24 shows an engine map along with the demand curves for a stepped transmission and the control characteristic curve for a CVT tuned for minimum fuel consumption.

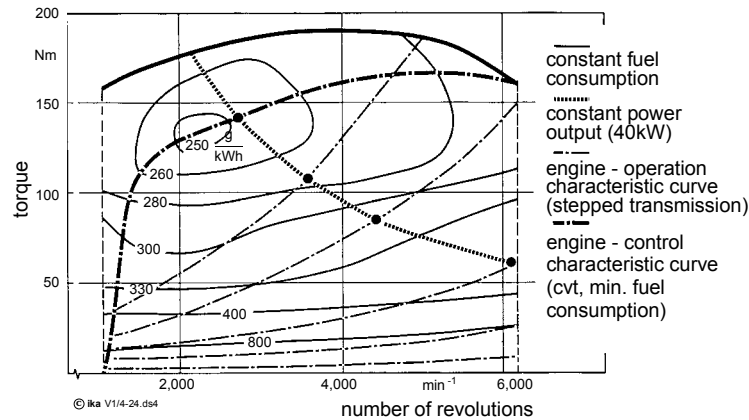


Fig. 4-24: Engine map with demand curves

In stepped transmissions, the layout of the largest gear is decisive since the largest proportion of a vehicle's operating time takes place at this ratio. The corresponding curve should be as close as possible to the point of minimal specific fuel consumption.

If the track fuel consumption for the single transmission levels is applied over the driving speed, the qualitatively presented consumption diagram shown in Fig. 4-25 results. Driving in the highest possible gear can also be considered fuel efficient as depicted in this figure.

In contrast to stepped transmissions, CVTs offer the possibility to freely choose the engine operating points for a prescribed power demand on the lines of constant engine power. Thus the optimum value can be adjusted as per requirement. In this way, using appropriate transmission control, it is possible to transform the demand curve such that the engine always operates in a fuel-efficient range. A deviation from this control line is made only in the upper speed range in order to make an excess torque available up to the nominal point.

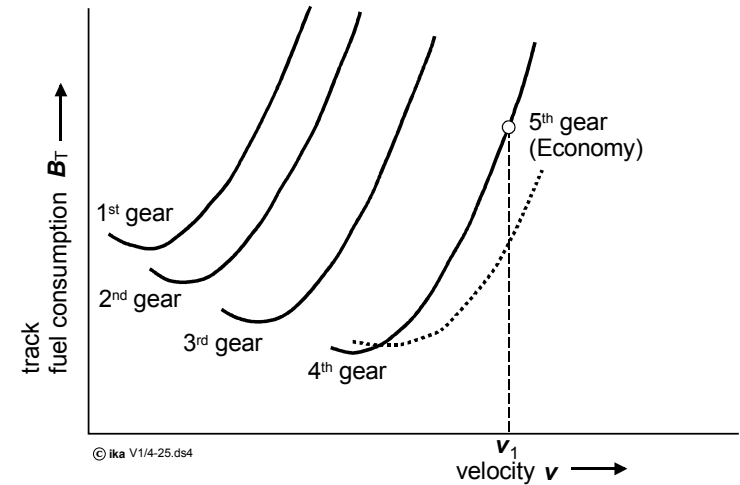


Fig. 4-25: Track fuel consumption (constant velocity drive)

The indicated reduction possibilities can be estimated from a comparison diagram of specific energy demand. In this case the applied energy demand is to be referred to a specific target application. Fig. 4-26 shows the results of a parameter study conducted for types of local public transportation involving busses, passenger cars, and trains. The number of transported passengers along with the distance covered are considered as application targets here.

For the passenger car, the ECE-cycle was assumed here as the representative driving course. For buses and trains, respective cycles which maintained the statistical averages for acceleration, driving velocity and the distance between stations were chosen. The speeds achieved in all cycles are so small that the air resistance can be neglected ($W_L < 0.1 \cdot W_{\text{dem}}$).

In quadrant 'A', both dominant portion of energy from the wheel as well as acceleration resistance are indicated, initially considering a mass-less vehicle, meaning the proportion of energy only required for the transportation of "pay-load". Under normal conditions, in today's vehicles this net mechanical energy demand amounts to 5 Wh/Pkm.

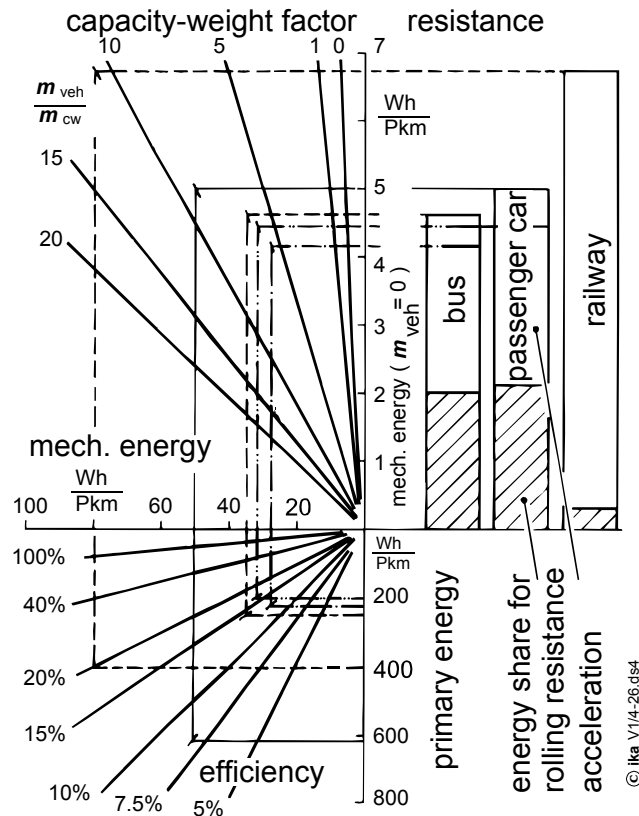


Fig. 4-26: Influences on the specific energy demand

In the quadrant 'B' this energy demand is multiplied with the ratio total mass/payload. In a passenger car, this results in an average mechanical energy demand of approx. 50 Wh/Pkm on the driven wheels.

The energy that finally has to be employed as primary energy is determined in quadrant 'C'. An efficiency η_{ges} is defined here which refers to the ratio of energy required for driving through a cycle to the supplied primary energy. By improving the payload factor and the total efficiency, the specific energy demand can be significantly reduced.

Using the example of an EC standard consumption (MVEG), Fig. 4-27 shows the effects of vehicle-related measures on the fuel consumption of a passenger car. It

also becomes obvious here that improvements particularly in the drivetrain efficiency and weight reduction lead to decreasing fuel consumption. Optimizing aerodynamics and road resistances only have a minor influence on this cycle.

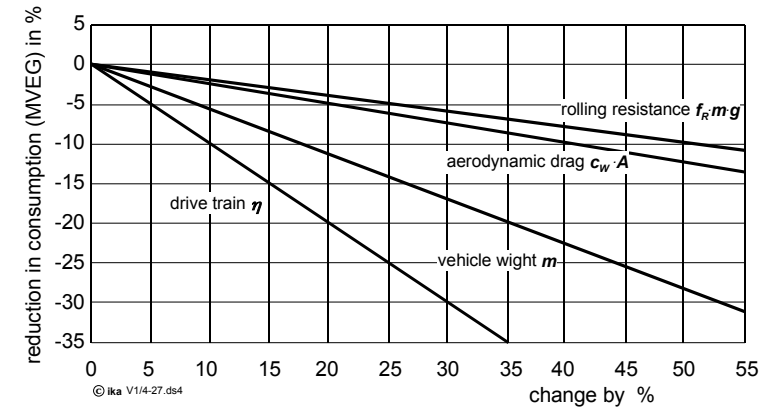
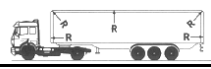
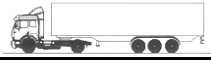
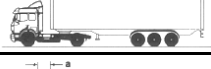
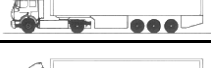
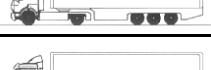


Fig. 4-27: Methods of reducing fuel consumption (Audi)

In Chapter 2.5 it has already been explained that the weight in commercial vehicles has a decisive influence on road resistance and in turn on fuel consumption. A reduction of the vehicle weight, primarily serves to increase the payload capacity. The total weight in the loaded condition is prescribed only by statutory regulations and not by vehicle construction. In contrast to passenger cars, there is a significant potential for reducing fuel consumption through aerodynamic measures. In particular, this is valid for long-distance trucks that are frequently driven in the range of allowable maximum speed.

Fig. 4-28 shows different design variations of semi trailers which were compared against each other in an experiment. The starting point was represented by an uncovered production vehicle. It was generally optimized over several stages using aerodynamic add-ons.

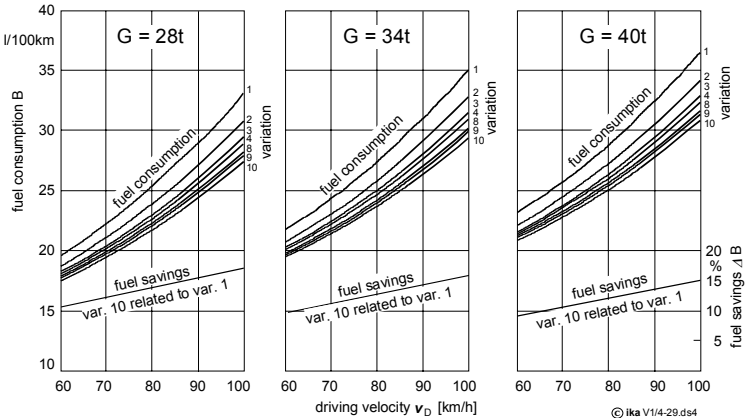
Fig. 4-29 indicates the fuel consumption dependent on speed for the different optimization levels at three different load conditions.

Var. No.	figure	description of variations	
1		starting position without LKK and EKK, body edges rectangular	clearance space $a = 830 \text{ mm}$
2		see 1 but body edges rounded	$R = 150 \text{ mm}$
3		see 1 but with cab fairing and flaps (body edges rectangular)	
4		see 2 but with cab fairing and flaps (body edges rounded)	
5		see 4 but clearance space $a = 630 \text{ mm}$	
6		see 5 but side covering wheel cutouts unfaired	
7		see 6 but wheel cutouts partly faired	
8		see 6 but wheel cutouts fully faired	
9		see 8 but with underbody covering	
10		see 9 but with rear pulling-in	

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Fig. 4-28: Aerodynamic measures at semi trailer train (ATZ, 1993)

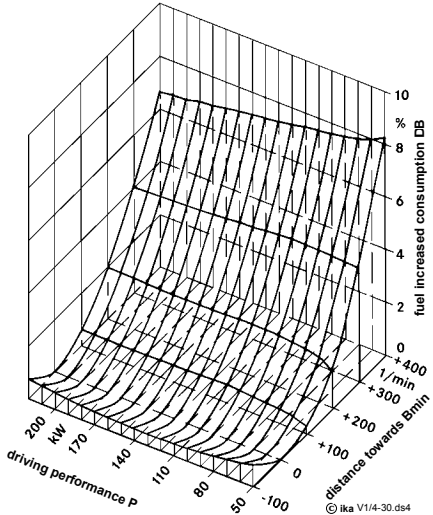
Also in the fully-loaded condition, a drop in fuel consumption of approx. 2l/100 km can be achieved only using an air-guiding body. It is also noticed that expensive full-body fairing on the semi trailer only indicates minor improvements. The particular area of application of the truck determines whether such measures are justified or not.



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Fig. 4-29: Influence of aerodynamic measures on the fuel consumption (ATZ, 1993)

The influence of correct layout of the transmission ratio or the driver's correct choice of gears on the fuel consumption has already been mentioned before. Using the example of a 40 t semi-trailer Fig. 4-30 shows how the fuel consumption increases if the engine is not operated at the optimum operating point.



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Fig. 4-30: Influence of the change of engine speed on fuel consumption (ATZ, 1993)

In actual driving, the influence of side aggregates on the fuel consumption is often underestimated. At the institute experiments were carried out on an individual example in order to quantify this influence (Table 4-1).

Test vehicle: BMW 525 iX – Consumption [l/100km]			
Cycle	Without accessories	With air conditioner	With air conditioner and generator
90 km/h constant	7.89	8.94	9.76
120 km/h constant	9.80	10.75	11.22
ECE-cycle	10.95	12.43	14.10

Table 4-1: Influence of aggregates on the fuel consumption

It becomes obvious that especially at low speeds the increase in fuel consumption is significant, up to 29%. In future, consumption-optimized side aggregate concepts can be positively used inspite of a probably higher technical expense.

4.3 Drivetrain Layouts

There are generally three possibilities to transfer the driving power on to the road:

- Rear-wheel drive (RWD),
- Front-wheel drive (FWD),
- All-wheel drive (AWD).

The two-axle motor vehicle forms the basis for the description of the drivetrain concepts. The possible drivetrain layouts in commercial vehicles with more than two axles, can generally be assigned to one of the three basic concepts of the two-axle vehicle.

4.3.1 Rear-Wheel Drive

Until the 30s, the rear-wheel drive layout was almost exclusively used. The concept of front-wheel steering almost inevitably led to a driven rear axle since the torque transfer to the steered wheels caused problems. Today only a third of the vehicles produced in Germany are equipped with rear-wheel drive. Fig. 4-31 shows the possible engine layouts.

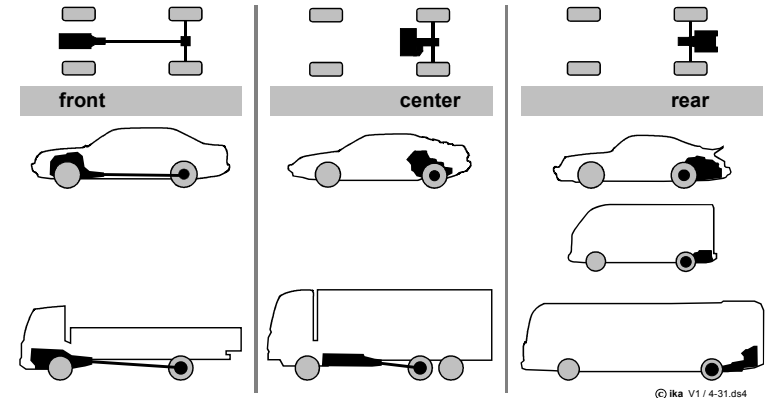


Fig. 4-31: Engine layout in passenger cars and commercial vehicles with RWD

The most widespread RWD type, the standard drive, consists of the engine being installed in front of the front axle (front engine). The torque transfer between transmission and driven axle (in the transaxle design, between engine and transmission located in front of the rear axle) takes place via the cardan shaft. This results in a negligibly higher weight and in passenger cars requires a tunnel in the passenger compartment.

Further engine layouts associated with RWDs include the centrally mounted engine located between front and rear axles and the rear engine layout behind the rear axle.

The centrally mounted engine layout highly impairs the amount of space available in the passenger compartment and also reduces the transporting capacity of a vehicle. In passenger cars, this layout is almost exclusively used in sports cars and in racing. In commercial vehicles the centrally mounted engine is applied in underfloor layout.

Particularly in passenger cars, the rear engine layout also creates limitations in the maximum pay-load carrying capacity since the trunk is displaced to the area above the front axle. The necessity for larger wheel houses in this area (steering angle) reduces the available space. However, this layout is used widely in busses where the standard drive becomes unfavourable due to long wheel bases involved.

The RWD in passenger cars proves to be a disadvantage from the point of view of economy of space in each of its designs. The standard drive, which is the most frequently used design, is however advantageous in average and large vehicles as a result of the body stiffening effect produced by the tunnel for the cardan shaft.

The inherently oversteering tendency of rear-wheel driven vehicles in the absence of appropriate measures (refer AE-II) can largely be optimised for drivability specific to safety using modern chassis construction.

The main advantage of rear-wheel driven vehicles lies in the total absence of drivetrain influences on steering. In addition, modern rear-wheel driven vehicles have improved traction since in addition to their often excellent axle load distribution (50:50), the rear axle is additionally loaded during acceleration or by a pay-load (Chapter 4.4).

In commercial vehicles, which are primarily designed for high pay-loads and the high rear-axle loads associated with it, RWD is used since it provides sufficient traction also when fully loaded. In this case, the rear axle of heavy commercial vehicles is often replaced by a twin axle. In case of the latter, either one or also both axles are driven.

4.3.2 Front-Wheel Drive

More and more vehicles with front-wheel drive have seen application since the 60s. It is primarily used in association with the front engine.

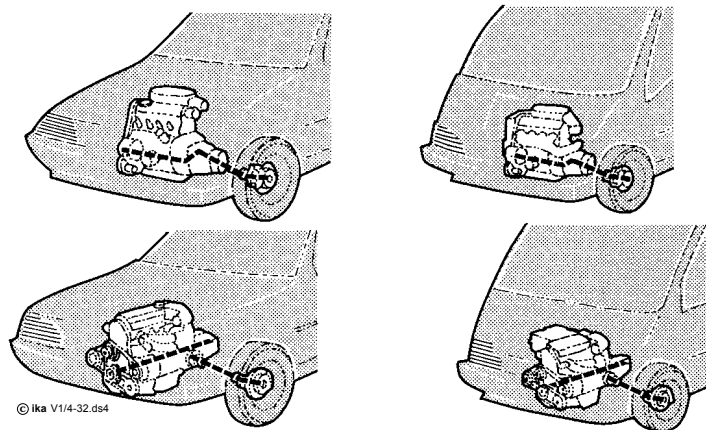


Fig. 4-32: Engine layout in passenger cars and lightweight van-type trucks with FWD

In order to transfer the driving power on to the driven front wheels, constant-velocity (free of fluctuations in angular velocity and torque) joints are required. Spherical

constant-velocity joints which transfer high torques over large angular ranges are generally used in today's vehicles.

The compact drivetrain unit composed of the engine, transmission and differential, installed primarily in the form of a transversely mounted engine-transmission assembly and a simple rear axle design, provide additional space in FWDs. This is of particular significance in smaller passenger cars.

The engine's positioning in the area of the driven axle results in front-wheel driven designs having a better traction at small pay-loads and accelerations compared to the standard drive.

At higher pay-loads combined with higher acceleration values on roads with good grip, the load on the driven axle drops, hence leading to a reduction of the utilisable traction. This disadvantage is accepted in lightweight van-type trucks in order to realize as even and as large a loading space as possible. The advantage of the FWD with respect to weight is negligible compared to the standard drive.

Front-wheel driven vehicles are inherently more drivable, showing a good-natured and easily controlled driving behavior.

The assembly of engine, transmission, and front axle as a single block brings it closer to automated installation which leads to a reduction in production costs.

4.3.3 All-Wheel Drive

The desire for optimum traction creates the need for AWD, particularly in vehicles which are driven off-road. AWD in the form of engageable AWD has been used in the commercial vehicle sector and in off-road vehicles for a long time. Success in sport led to an increased involvement on the part of many automobile manufacturers in AWDs in the early 80s resulting in the appearance of a number of concepts.

AWD is used in combination with all engine layouts. In commercial vehicles, both the front as well as rear axles can be designed in the form of driven tandem axles.

Fig. 4-33 shows the possible engine layouts in AWD for different vehicles.

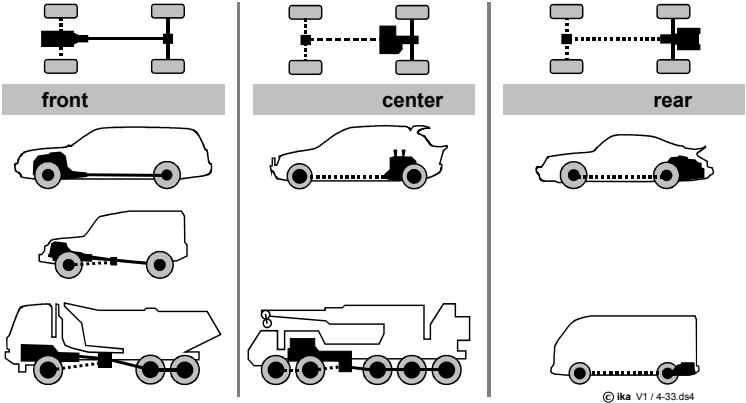


Fig. 4-33: Engine layout in passenger cars and commercial vehicles with AWD

Table 4-2 shows possible all-wheel drive concepts that essentially differ according to driving and braking behavior and convenience of operation.

Passenger Car All-Wheel Drive Concepts					
Concept	Manually engaged	Automatically engaged	Permanent		
Type	Rigid clutch	Multi-disk clutch, center differential	Center differential	Multi-disk clutch	Visco clutch
Torque distribution FA/RA	Variable (friction dependent)	Constant	Constant	Variable (selectable/slip-dependent)	Variable (slip-dependent)
Locking characteristic	Complete	Slip-dependent	Manual slip-dependent	Slip-dependent	Slip-dependent

Table 4-2: All-wheel drive concepts

As examples, some all-wheel drive systems are presented as follows:

a) Manually Engaged AWD

In the manually engaged AWD, a connection between the differentials of front and rear axles is realized over a manually operated clutch. If the clutch is opened, the vehicle is then driven by the front or rear axle as the case may be (Fig. 4-34).

A simple design with positive-engagement between the drive shafts leading to the front and rear axles does not allow for speed differences between the axles and is mostly operated manually.

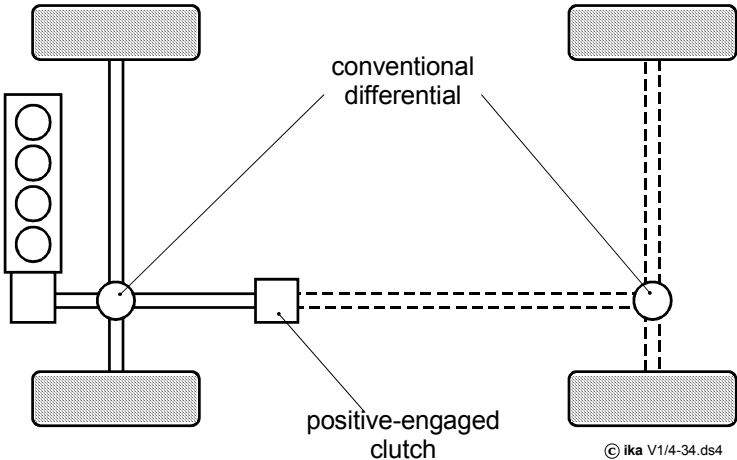


Fig. 4-34: Manually overdriven AWD (e.g. Fiat Panda 4x4)

b) Electronic Shift-on-Fly AWD

The shifting process can be automated using a frictional clutch (e.g. multi-disk clutch). Fig. 4-35 shows the transfer case of such an all-wheel drive system consisting of a spur-gear planetary differential and two multi-disk clutches.

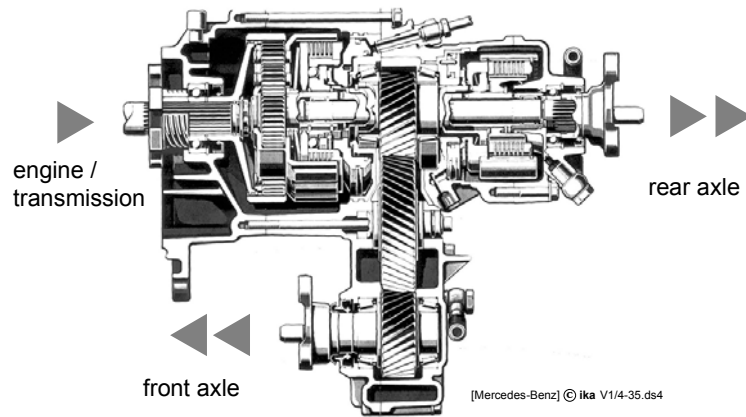


Fig. 4-35: Friction clutch transfer case (Daimler-Benz 4-Matic)

In case of the four-wheel drive “4-Matic” of Mercedes-Benz, the vehicle is driven on pure rear-wheel drive in normal operation. The differential is locked by the first spring-loaded multi-disk clutch while the opened second multi-disk clutch separates sun wheel and front-wheel drive. Based on demand, three operating steps which add on to each other may be recalled.

Step 1: The front-wheel drive is engaged:

Using electronics the first multi-disk clutch which locks the differential is hydraulically released, while the second one is closed. In this way, the differential distributes the driving torque to the front and rear axles at a definite torque ratio (35:65).

Step 2: In extreme circumstances, the differential can additionally be locked by closing the first clutch. If the hydraulics fail, the spring load guarantees a functional RWD.

Step 3: The rear-axle differential is locked.

During braking, all locks are released in order to allow the ABS system to operate without any constraint. The control of the locks and clutches takes place electronically. Fig. 4-36 shows the components of such a four-wheel drive.

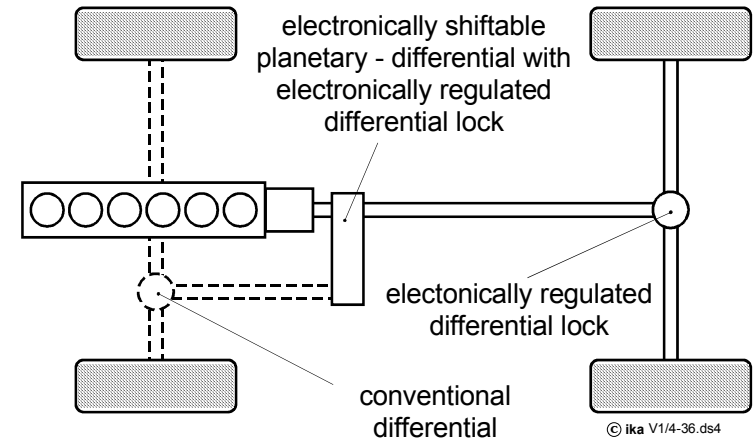


Fig. 4-36: Electronic shift-on-fly all-wheel drive (Daimler Benz 4-Matic)

c) Permanent AWD

In contrast to the shiftable variants, all-wheel drive can also have a permanent effect. In this case however, the associated stresses and slip have to be compensated by speed compensation between the axles. The various concepts which have been realized so far, differ according to the method of torque distribution and the differential locks used. Using a transfer case, the driving torque can be distributed to the front and rear axles in a definite ratio. Bevel-gear differentials are employed for symmetric torque ratios while spur-gear planetary differentials for an unequal distribution of the torques to the axles. Speed compensation at the differentials can be impeded or put a stop to using locking concepts already discussed in Chapter 3.5.3 (Fig. 4-37).

A visco-hydraulic clutch represents an alternative to a lockable transfer case as a link between the driven axles. In this concept, the vehicle is driven only by one axle connected directly with the engine. A speed balance is achieved at low values of slip as a result of the second axle being driven freely on the road. The visco-hydraulic clutch produces a torque on the second slower running axle only at differing speeds between front and rear axles, e.g. at increased slip on the driven axle (Fig. 4-38).

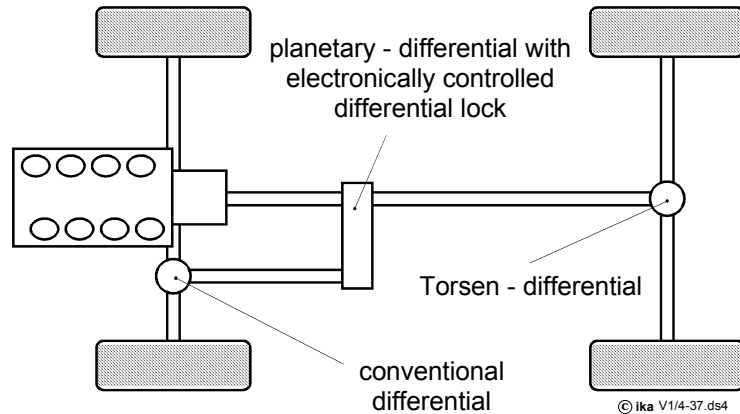


Fig. 4-37: Permanent all-wheel drive (Audi V8)

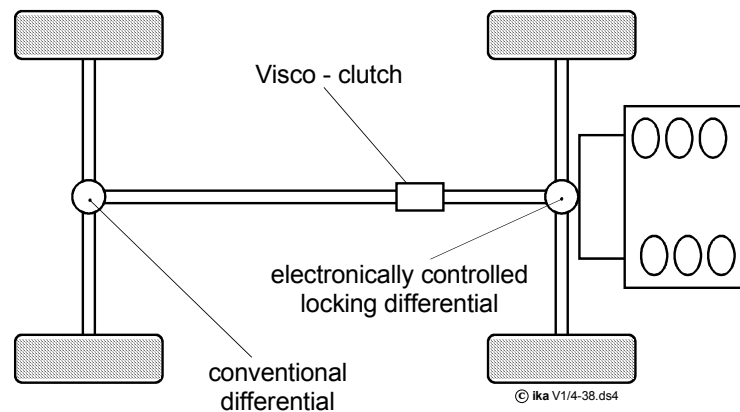


Fig. 4-38: Permanent all-wheel drive (Porsche Carrera 4)

A hydraulic multi-disk clutch which can be controlled using the clutch pressure replaces the transfer case as a simple (small mechanical expense) but effective all-wheel drive concept. Control electronics take over the control of the locking effect between both axles. Using various sensors the system can react to a specific driving condition. Based on the amount of effort invested, the locking effect can be controlled step-wise or continuously variable. Special requirements such as the complete release of the lock during braking, can be taken into consideration.

d) Assessment of Concepts

In contrast to single-axle drives, all AWD concepts are associated with significantly higher production costs.

The lowest additional expense is associated with mechanically engaged all-wheel drive. It distinguishes itself through its simple and robust design. At poor road contact, the rigid engagement of the front and rear axles produce satisfactory traction. The rigid engagement of the axles, however leads to stress on the drivetrain, slip on the wheels (loss in side forces) and to a change in the brake-force distribution in curved roads with high coefficients of friction. If dealt with properly, large benefits can however be achieved with these systems.

In permanent AWD, the possibility of balancing the speeds between the axles using intermediate differentials exists. The differences between the various concepts of permanent all-wheel drive cannot be noticed in normal driving. Only extreme situations expose the advantages of one or the other systems. The electronic-hydraulic control is currently considered the most optimum of all production all-wheel drive systems. A safe driving behavior during acceleration is common to all permanent all-wheel drives, since even at low coefficients of friction, the slip of traction forces on the wheels and hence resulting loss in cornering force is smaller compared to a single-axle driven concept. At high locking values in the differentials, the braking behavior is influenced. Even the use of ABS cannot compensate the discrepancy between driving and braking behavior. This is also applicable to self-regulating differential locks such as the Torsen-differential or the visco-hydraulic clutch, whose locking effect hinders independent wheel braking which is necessary for ABS. Integrated free-wheeling or adjusted "soft" locking characteristics are required.

The disadvantages of additional weight and increased friction losses compared to rear and front-wheel drives is common to all all-wheel drive concepts. This leads to an increased fuel consumption, which is however negligible in modern concepts equipped with electronically controlled locks. Hence Porsche quotes the identical 1/3-mix consumption for the Carrera 4 as well as the Carrera 2. The additional space requirement has to be created by reducing the size of the fuel tank and trunk or by locating the vehicle higher. In addition, the second driven axle requires a considerably more constructional expense compared to its undriven counterpart, particularly in originally front-wheel driven vehicles combined with a simple twist-beam rear axle (AE-II).

4.4 Driving Limits

In the consideration of possible driving performances, it has so far been assumed that the circumferential force acting in the tire tread area can be infinitely transferred onto the road. Indeed, according to the law of friction, the transferable traction force between the tires and road is limited by the coefficient of friction and the axle loads.

$$F_{X_{r(f)}} = \mu \cdot F_{Z_{r(f)}} \quad \text{Eq. 4-35}$$

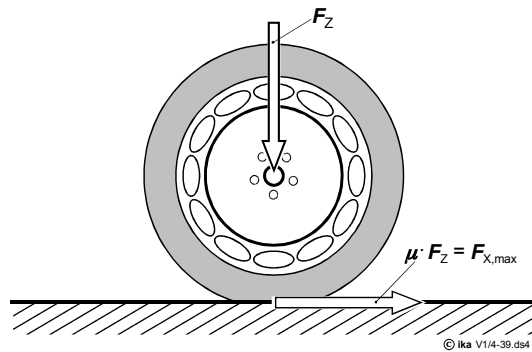


Fig. 4-39: Forces on the wheel

This restriction results in friction induced driving limits. Besides the road and tire condition, the coefficient of friction μ also depends on the slip between tires and road. In this case, Slip is defined the difference between wheel speed and vehicle speed relative to the reference speed (Fig. 4-40).

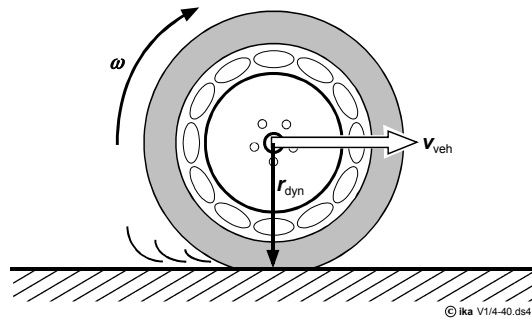


Fig. 4-40: Vehicle-/ wheel speed

According to the driving condition, slip can be distinguished into traction slip s_A and the brake slip s_B :

$$s_A = \frac{v_R - v}{v_R} \quad \text{Eq. 4-36}$$

$$s_B = \frac{v - v_R}{v} \quad \text{Eq. 4-37}$$

where: $v_W = W_{dyn} \cdot \omega_W =$ wheel speed

$v =$ longitudinal speed of the vehicle

Fig. 4-41 shows the characteristic of the coefficient of friction over the slip. The maximum coefficient of friction is reached at the so-called critical slip s_{krit} . The coefficient of friction then drops to the sliding friction coefficient μ_G at a slip of $s=1$.

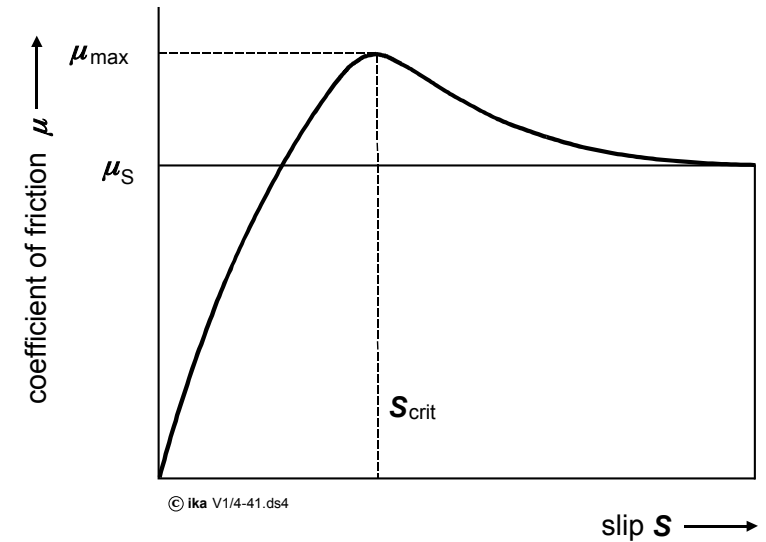


Fig. 4-41: Coefficient of friction-slip-curve

In Fig. 4-42 illustrates the friction-slip-curves for different road surfaces. Moreover, the tires used and the driving speed have an essential influence on the curve (Fig. 4-43).

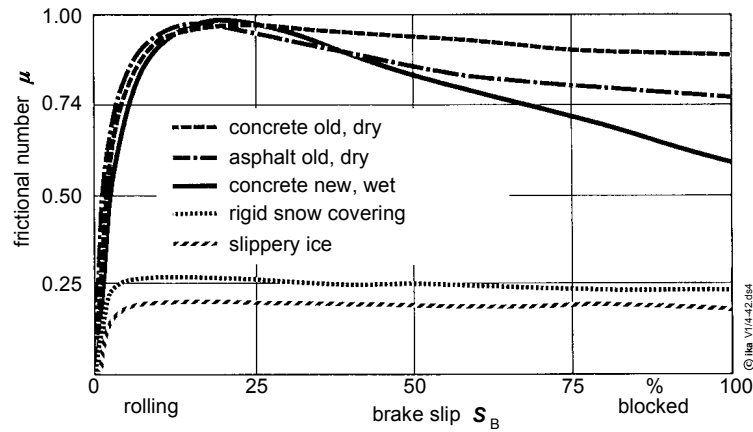


Fig. 4-42: Coefficient of friction for different roadway conditions

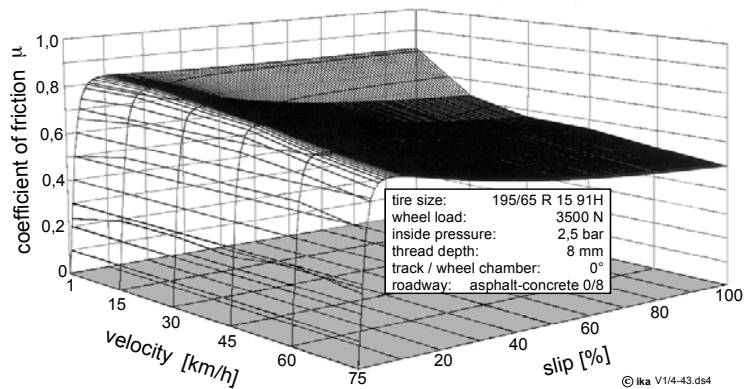


Fig. 4-43: Coefficient of friction for different vehicle speeds

Apart from the coefficient of friction, for detecting the driving limits it is necessary to know the forces acting on the tires and the roadway.

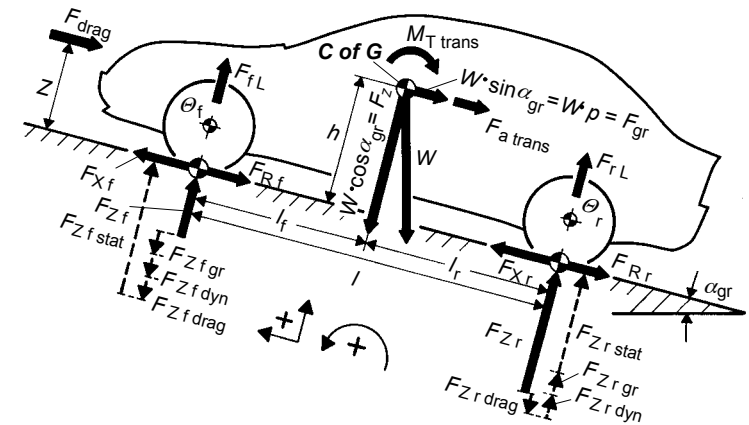
4.4.1 Vertical forces

The vertical forces (axle loads) consist of four components:

- Static component,

- Gradient component,
- Aerodynamic aerodynamic,
- Dynamic component.

Fig. 4-44 shows the forces acting on a vehicle accelerated on an upward gradient. The equations associated with the vertical loads are summarized in Table 4-3.



W	wight-force vehicle	$F_{Zr drag}$	aerodynamic share, RA
F_{Zf}	axle load front	$F_{Zr dyn}$	dynamic share, RA
$F_{Zf stat}$	statistic share, FA	$F_{Xf(r)}$	traction force, front (rear)
$F_{Zf gr}$	upgrade share, FA	$F_{Rf(r)}$	rolling resistance force, front (rear)
$F_{Zf drag}$	aerodyn. share, FA	$F_{f(r) L}$	lift force, front (rear)
$F_{Zf dyn}$	dynamic share, FA	F_{drag}	air resistance force
F_{Zr}	axle load rear	$F_{a trans}$	inertia force
$F_{Zr stat}$	statistic share, RA	$\theta_{f(r)}$	red. inertia moment FA (RA)
$F_{Zr gr}$	upgrade share, RA	α_{gr}	angle of gradient

Fig. 4-44: Forces on a vehicle accelerated in an upward gradient

Vertical Forces	
Front axle	Rear axle
$F_{Zf} = (F_{Zf \text{ stat}} - F_{Zf \text{ gr}}) - F_{Zf \text{ drag}} - F_{Zf \text{ dyn}}$ $= m \cdot g \cdot \left(\frac{l_f}{l} \cdot \cos \alpha_{gr} - \frac{h}{l} \cdot \sin \alpha_{gr} \right)$ $- c_{Af} \cdot A \cdot \frac{\rho_a}{2} \cdot (v \pm v_w)^2$ $- m \cdot a \cdot \left(\frac{h}{l} + \frac{\Theta_{\text{trans red}}}{m \cdot r_{\text{dyn}} \cdot l} \right)$	$F_{Zr} = (F_{Zr \text{ stat}} - F_{Zr \text{ gr}}) - F_{Zr \text{ drag}} - F_{Zr \text{ dyn}}$ $= m \cdot g \cdot \left(\frac{l_r}{l} \cdot \cos \alpha_{gr} - \frac{h}{l} \cdot \sin \alpha_{gr} \right)$ $- c_{Ar} \cdot A \cdot \frac{\rho_a}{2} \cdot (v \pm v_w)^2$ $- m \cdot a \cdot \left(\frac{h}{l} + \frac{\Theta_{\text{trans red}}}{m \cdot r_{\text{dyn}} \cdot l} \right)$

Table 4-3: Vertical loads

The static ratio of axle loads on the front axle can be determined by a moment balance around the rear tire contact point. The static ratio of axle load on the rear axle can be similarly determined from a moment balance around the front tire contact point. In this case, aerodynamic and dynamic shares are neglected. In a gradient, the additional shift of the axle load dependent on the angle of the gradient has to be considered in the calculation of statistic axle loads.

The aerodynamic component produced by the air flow around the moving vehicle has an effect on the vertical loads in the form of lift. Similar to the force of air resistance, lift is also calculated using a coefficient of lift c_A . Lift can either be positive (lift) or negative (negative lift).

Acceleration or deceleration generates a dynamic axle load ratio as a result of inertia forces and moments.

The force of inertia acting at the center of gravity of the vehicle is produced by the translational acceleration of the vehicle mass, while the moment $M_{T\text{quer}}$, by the rotational acceleration of the masses. In order to calculate the axle load, the mass inertia forces of these parts (wheels, drive shafts, also engine and transmission in a transversally installed engine) are reduced to the driven axle.

If only the forces of inertia acting on the accelerated vehicle are considered, the moment balance around the front tire tread point results in:

$$F_{Zr \text{ dyn}} \cdot l - F_{a \text{ trans}} \cdot h - \Theta_{\text{trans red}} \cdot \ddot{\varphi} = 0 \quad \text{Eq. 4-38}$$

With $\ddot{\varphi} = \frac{a}{r_{\text{dyn}}}$ and $F_{a \text{ trans}} = m \cdot a$ for the dynamic component at the rear axle:

$$F_{Zr \text{ dyn}} = \left(m \cdot \frac{r}{l} + \Theta_{\text{trans red}} \cdot \frac{1}{r_{\text{dyn}} \cdot l} \right) \cdot a \quad \text{Eq. 4-39}$$

From the vertical force balance, the dynamic axle load ratios are found to be equal and opposite:

$$F_{Zr \text{ dyn}} = -F_{Zf \text{ dyn}} \quad \text{Eq. 4-40}$$

4.4.2 Horizontal Forces

In order to set the vehicle in motion, a horizontal force has to be created at the tire tread points of the driven axle. This horizontal force has to be in equilibrium with the acting forces. The horizontal forces acting in the vehicle coordinate system are also shown in Fig. 4-44. The following equations show the required horizontal force in the tire tread points at the front or rear axle in FWD and RWD.

Front-wheel drive:

$$F_{Xf} = F_a + F_{gr} + F_{a \text{ trans}} + F_{Rr} + F_{a \text{ rot r}}$$

$$F_{Xf} = c_w \cdot A \cdot \frac{\rho_a}{2} \cdot (v \pm v_w)^2 + m \cdot g \cdot \sin \alpha_{gr} + m \cdot a +$$

$$+ m \cdot g \cdot f_R \cdot \left(\frac{l_r}{l} \cdot \cos \alpha_{gr} - \frac{h}{l} \cdot \sin \alpha_{gr} \right) + \frac{\Theta_{red r}}{r_{\text{dyn}}^2} \cdot a \quad \text{Eq. 4-41}$$

Rear-wheel drive:

$$F_{Xr} = F_a + F_{gr} + F_{a \text{ trans}} + F_{Rf} + F_{a \text{ rot f}}$$

$$F_{Xr} = c_w \cdot A \cdot \frac{\rho_a}{2} \cdot (v \pm v_w)^2 + m \cdot g \cdot \sin \alpha_{gr} + m \cdot a +$$

$$+ m \cdot g \cdot f_R \cdot \left(\frac{l_f}{l} \cdot \cos \alpha_{gr} - \frac{h}{l} \cdot \sin \alpha_{gr} \right) + \frac{\Theta_{red f}}{r_{\text{dyn}}^2} \cdot a \quad \text{Eq. 4-42}$$

The reduced mass moment of inertia $\Theta_{\text{redh(v)}}$ at the non-driven wheels only consists of the mass moments of inertia of the wheels and the parts which rotate along with them, such as brake disks. The mass moments of inertia Θ_{red} of the rotating parts of the remaining drive train are also reduced to the driven wheels and similar to the road resistance force $F_{Rh(v)}$ of the driven wheels represents an “internal force” or an “internal moment”.

It has however to be considered here that the “internal forces and moments” have to be overcome from the point of view of power by the powertrain. Since the horizontal force in the tire tread points considered here are limited to a definite driving condition, the inner forces and moments need not be taken into consideration.

4.4.3 Friction Loads

In order to relate the vertical and horizontal forces to the utilisable coefficient of friction of the road, the co-efficient of friction on the front and rear axle is determined by ratio of the respective horizontal force to the vertical force.

$$f_r = \frac{F_{Xr}}{F_{Zr}}, \quad f_f = \frac{F_{Xf}}{F_{Zf}} \quad \text{Eq. 4-43}$$

The ratios resulting for different drive concepts and driving situations are summarized in Table 4-4.

4.4.4 Friction induced Accelerating and Climbing Ability

If the entire friction potential is utilised for acceleration or climbing, one receives the equilibrium of forces by setting up a force balance in the x and y direction and the moment balance around the y-axis which is indicated in Table 4-5 for various drive concepts.

The following conditions are considered in the derivation:

- For a maximum friction induced accelerating ability from standstill, the rotating masses are neglected
- For climbing ability, low driving speeds are assumed so that air resistance can be neglected; besides only deceleration is considered

Front-Wheel Drive		
	Front wheels	Rear wheels
Friction force for straight-ahead driving at constant speed $\ddot{x} = 0, \quad p = 0, \quad f_R = 0$	$f_f \approx \frac{F_L}{F_{Zf \text{ stat}} - F_{Zf L}}$	$f_r \approx 0$
Friction force at constant-speed uphill driving at low speed $\ddot{x} = 0, \quad f_R = 0, \quad F_L = 0$	$f_f \approx \frac{W \cdot p}{W \cdot (\frac{l_f}{l} - \frac{h}{l} \cdot p)}$ $= \frac{p}{\frac{l_f}{l} - \frac{h}{l} \cdot p}$	$f_r \approx 0$
Friction force for accelerated drive in plain at low speed	$f_f \approx \frac{W \cdot \frac{a}{g}}{W \cdot (\frac{l_f}{l} - \frac{h}{l} \cdot \frac{a}{g})}$	$f_r \approx 0$
Rear-Wheel Drive		
	Front wheels	Rear wheels
Friction force for straight-ahead driving at constant speed $\ddot{x} = 0, \quad p = 0, \quad f_R = 0$	$f_f \approx 0$	$f_r \approx \frac{F_L}{F_{Zr \text{ stat}} - F_{Zr L}}$
Friction force at constant-speed uphill driving at low speed $\ddot{x} = 0, \quad f_R = 0, \quad F_L = 0$	$f_f \approx 0$	$f_r \approx \frac{W \cdot p}{W \cdot (\frac{l_r}{l} - \frac{h}{l} \cdot p)}$ $= \frac{p}{\frac{l_r}{l} - \frac{h}{l} \cdot p}$
Friction force for accelerated drive in plain at low speed	$f_f \approx 0$	$f_r \approx \frac{W \cdot \frac{a}{g}}{W \cdot (\frac{l_r}{l} - \frac{h}{l} \cdot \frac{a}{g})}$
All-Wheel Drive		
	Front wheels	Rear wheels
Friction coefficient i: proportion of driving torque at rear axle	$f_f = \frac{(1-i) \cdot (F_{a \text{ trans}} + F_{gr} + F_R + F_{drag})}{F_{Zf \text{ stat}} - F_{Zf \text{ drag}} - F_{Zf \text{ gr}} - F_{Zf \text{ dyn}}}$ $= \frac{\frac{\Theta_{red f}}{i_{dyn}} \cdot a + F_{Rf}}{F_{Zf \text{ stat}} - F_{Zf \text{ drag}} - F_{Zf \text{ gr}} - F_{Zf \text{ dyn}}}$	$f_r = \frac{i \cdot (F_{a \text{ trans}} + F_{gr} + F_R + F_{drag})}{F_{Zr \text{ stat}} - F_{Zr \text{ drag}} - F_{Zr \text{ gr}} - F_{Zr \text{ dyn}}}$ $= \frac{\frac{\Theta_{red r}}{i_{dyn}} \cdot a + F_{Rr}}{F_{Zr \text{ stat}} - F_{Zr \text{ drag}} - F_{Zr \text{ gr}} - F_{Zr \text{ dyn}}}$
Friction coefficient at constant-speed in plain	$f_f = \frac{(1-i) \cdot (F_R + F_{drag}) - F_{Rf}}{F_{Zf \text{ stat}} - F_{Zf \text{ drag}}}$	$f_r = \frac{i \cdot (F_R + F_{drag}) - F_{Rr}}{F_{Zr \text{ stat}} - F_{Zr \text{ drag}}}$

Table 4-4: Friction coefficient

Front-Wheel Drive	
Friction limited accelerating ability in plain	$a_{\max,\mu} = \frac{(\mu \cdot l_f - f_R \cdot l_f)}{l + (\mu + f_R) \cdot (h + \frac{\Theta_{\text{trans red}}}{m \cdot r_{\text{dyn}}})} \cdot g$ $+ \frac{(f_R \cdot c_{Ar} - \mu \cdot c_{Af} - c_w) \cdot A \cdot \frac{\rho_{\text{drag}}}{2} \cdot (v \pm v_W)^2}{l + (\mu + f_R) \cdot (h + \frac{\Theta_{\text{trans red}}}{m \cdot r_{\text{dyn}}})} \cdot \frac{l}{m}$
Friction limited accelerating ability in plain from standstill	$a'_{\max,\mu} \approx \frac{(\mu \cdot l_f - f_R \cdot l_f)}{l + (\mu + f_R) \cdot h} \cdot g$
Friction limited climbing ability	$\tan \alpha_{\text{gr max},\mu} \approx \frac{(\mu \cdot l_f - f_R \cdot l_f)}{l + (\mu + f_R) \cdot h}$
Rear-Wheel Drive	
Friction limited accelerating ability in plain	$a_{\max,\mu} = \frac{(\mu \cdot l_f - f_R \cdot l_f)}{l + (\mu + f_R) \cdot (h + \frac{\Theta_{\text{trans red}}}{m \cdot r_{\text{dyn}}})} \cdot g$ $+ \frac{(f_R \cdot c_{Af} - \mu \cdot c_{Ar} - c_w) \cdot A \cdot \frac{\rho_{\text{drag}}}{2} \cdot (v \pm v_W)^2}{l - (\mu + f_R) \cdot (h + \frac{\Theta_{\text{trans red}}}{m \cdot r_{\text{dyn}}})} \cdot \frac{l}{m}$
Friction limited accelerating ability in plain from standstill	$a'_{\max,\mu} \approx \frac{(\mu \cdot l_f - f_R \cdot l_f)}{l - (\mu + f_R) \cdot h} \cdot g$
Friction limited climbing ability	$\tan \alpha_{\text{gr max},\mu} \approx \frac{(\mu \cdot l_f - f_R \cdot l_f)}{l - (\mu + f_R) \cdot h}$
All-Wheel Drive	
Friction limited accelerating ability in plain	$a_{\max,\mu} = \mu \cdot g - \frac{(\mu \cdot c_{Af} + \mu \cdot c_{Ar} + c_w)}{m} \cdot A \cdot \frac{\rho_{\text{drag}}}{2} \cdot (v \pm v_W)^2$
Friction limited accelerating ability in plain from standstill	$a'_{\max,\mu} \approx \mu \cdot g$
Friction limited climbing ability	$\tan \alpha_{\text{gr max},\mu} \approx \mu$

Table 4-5: Adherent accelerating- and climbing ability in different propulsion methods

Presuming that the wheel resistance coefficient is much smaller than the coefficient of friction ($f_R \ll \mu$) the following is applicable:

$$(\mu + f_R) \cdot h \ll l \quad \text{Eq. 4-44}$$

The following relations are obtained for driving limits in the plane:

$$a_{\max,\mu,\text{nose}} \approx \mu \cdot g \cdot \frac{l_r}{l} = \mu \cdot g \cdot \frac{F_{Zf \text{ stat},0}}{W} \quad \text{Eq. 4-45}$$

$$a_{\max,\mu,\text{tail}} \approx \mu \cdot g \cdot \frac{l_f}{l} = \mu \cdot g \cdot \frac{F_{Zr \text{ stat},0}}{W} \quad \text{Eq. 4-46}$$

and:

$$\tan \alpha_{\text{gr max},\mu,\text{nose}} \approx \mu \cdot \frac{l_r}{l} = \mu \cdot \frac{F_{Zf \text{ stat},0}}{W} \quad \text{Eq. 4-47}$$

$$\tan \alpha_{\text{gr max},\mu,\text{tail}} \approx \mu \cdot \frac{l_f}{l} = \mu \cdot \frac{F_{Zr \text{ stat},0}}{W} \quad \text{Eq. 4-48}$$

where $F_{Zf \text{ stat},0}$, $F_{Zr \text{ stat},0}$ = statistic axle loads in the plain.

4.4.5 Traction Coefficient and Legal Regulations

The driving friction limits are essentially determined by the ratio of the static axle load of the driven axle to the total weight which is called the traction coefficient τ .

$$\tau = \frac{F_{Z \text{ driven stat},0}}{W} \quad \text{Eq. 4-49}$$

Its value is proportional to the position of the center of gravity and hence most of all dependent on the engine position inside the vehicle and the load condition.

Fig. 4-45 shows a representative cross-section of vehicles for which the traction coefficients dependent on load condition for a front-wheel drive, standard-drive, and rear engine, rear-wheel drive (by law, the vehicle load is indicated in kg or in t) are indicated.

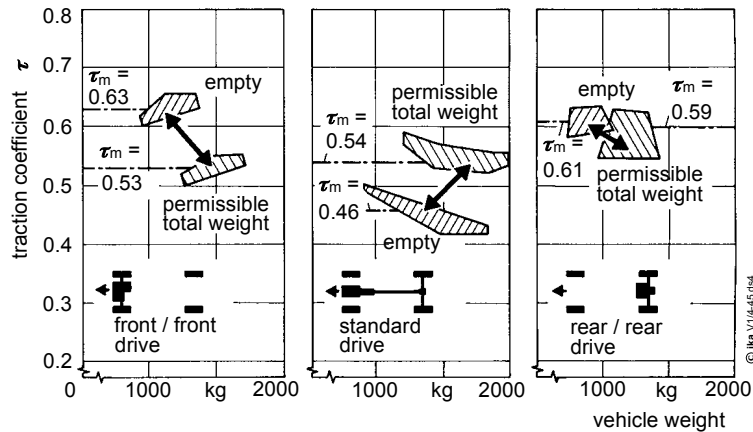
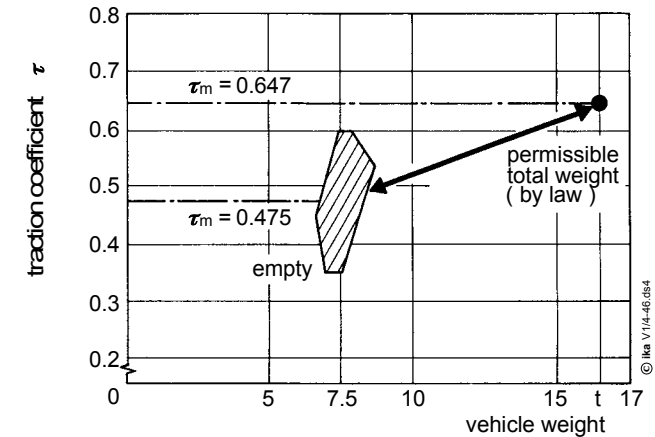


Fig. 4-45: Traction coefficients for different drive concepts

A front-driven vehicle has favorable traction conditions at unladen driving conditions since the engine is located above the driven axle. This however deteriorates with increasing pay-load since the trunk is positioned on the rear axle. In contrast, an improvement in the traction coefficient results when vehicles with standard drive are loaded such that they balance out the conceptual advantages of front-wheel driven vehicles. Furthermore, the rear axle is additionally loaded by the dynamic shift of weight to the rear axle during acceleration or climbing. In passenger cars with rear-engine rear-wheel drive, the good traction coefficient changes only insignificantly when loaded.

A change in the traction coefficient when loaded is of special significance in commercial vehicles due to the large weight differences between unladen and laden condition as a result of the high ratio of pay-load to unladen weight. Fig. 4-46 shows the traction coefficients of double-axle trucks with a permissible total weight of 17 t against the vehicle weight.

The spread is reduced to a point for the permissible vehicle weight, since the permissible axle load is limited to 11 t by law. This limit is to be considered in all cases for a driven rear axle.

Fig. 4-46: Traction coefficients of double-axle trucks with $W_{\text{perm}}=17 \text{ t}$

Since it is necessary to calculate the axle loads in order to determine the traction coefficient, legal regulations (StVZO) concerning the permissible maximum axle loads (Table 4-6) and permissible dimensions (Fig. 4-47) should be mentioned in this context. Additionally the prescribed minimum engine outputs, dependent on the maximum permissible weight are also indicated (Fig. 4-48).

Single axle	Dual axle	Triple axle
powered: 11.5 t nonpowered 10 t	$d < 1.0 \text{ m} : 11 \text{ t}$ $1.0 \text{ m} < d \leq 1.3 \text{ m} : 16 \text{ t}$ $1.3 \text{ m} < d \leq 1.8 \text{ m} : 18 \text{ t}$ $d > 1.8 \text{ m} : 20 \text{ t}$	$d \leq 1.3 \text{ m} : 21 \text{ t}$ $1.3 \text{ m} < d \leq 1.4 \text{ m} : 24 \text{ t}$

Table 4-6: Permissible axle loads (StVZO)

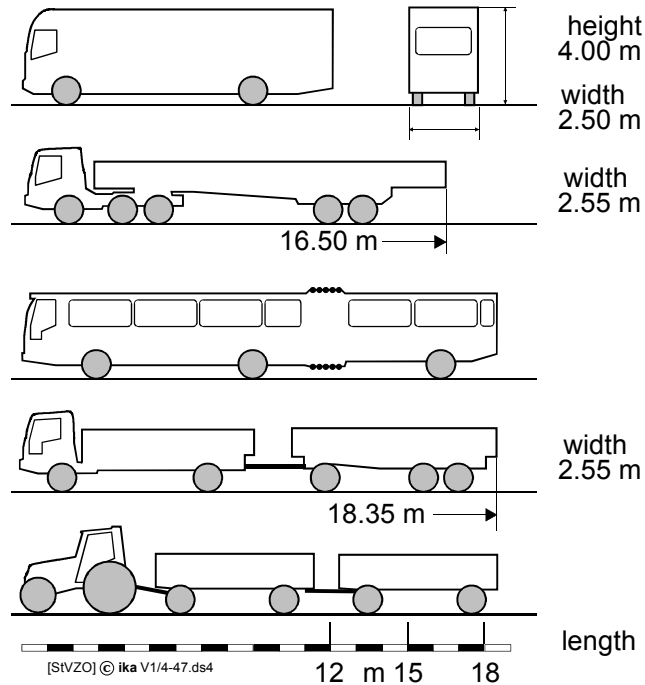


Fig. 4-47: Permissible dimensions for motor vehicles and truck trailers (StVZO)

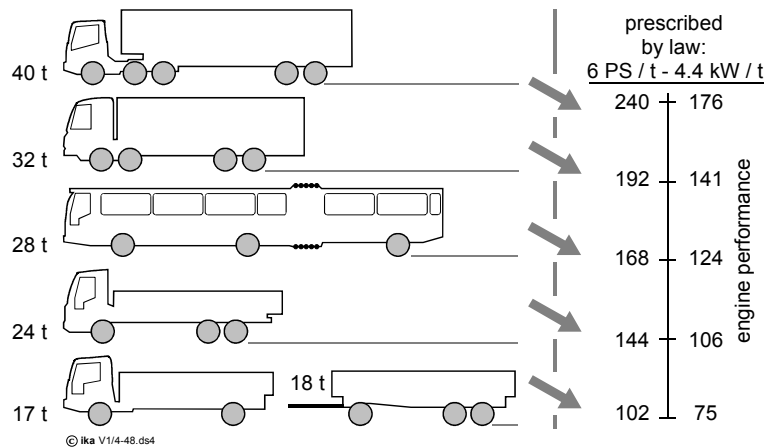


Fig. 4-48: Total weight and minimum engine output (StVZO)

Finally, the traction coefficients resulting from the permissible axle loads of driven axles at full load for some commercial vehicles concepts are indicated in the following table.

Vehicle type	Total weight [t]	Number of driven axles	Driven axles axle load [t]	Traction coefficient
Commercial vehicle, 3 axles	24	2*	18	0.750
Commercial vehicle, 3 axles	24	1**	11	0.485
Semitrailer	40	2	18	0.450
Semitrailer	40	1	11	0.275

*driven double axle

**driven axle with trailing axle

Table 4-7: Resulting traction coefficients at full loading for commercial vehicles

4.4.6 Braking Capacity limited by Friction

The decelerating road resistances and the engine brake moment are not considered in the design of braking systems. In the plane, when aerodynamic axle load influences are neglected, $F_z = G$. The following is then applicable:

$$F_B = m \cdot a_B = m \cdot g \cdot \frac{a_B}{g} = W \cdot z = F_Z \cdot z \quad \text{Eq. 4-50}$$

The total brake force corresponds to the sum of the braking forces $F_{B\text{wheel}}$ acting on the individual wheels. The braking forces on the wheels of an axle are generally of the same magnitude for the sake of driving stability.

$$F_B = 2 \cdot F_{B\text{wheel},f} + 2 \cdot F_{B\text{wheel},r} = F_{Bf} + F_{Br} \quad \text{Eq. 4-51}$$

Since the braking system in all vehicles is realized on all-wheels, a maximum deceleration can be achieved by the optimum distribution of the brake force on the axles:

$$a_{B\text{max}} = \mu \cdot g \quad \text{or} \quad z_{\text{max}} = \mu \quad \text{Eq. 4-52}$$

The necessary distribution (ideal brake-force distribution) depends on different parameters (speed, load, etc.).

Initially, in order to understand better, the coefficient of friction f (Equation 4-43) is again referred to. Here, it is defined as the ratio of the axle brake force F_{Bf} or F_{Br} to the respective axle load:

$$f_f = \frac{F_{Bf}}{F_{Zf}}, \quad f_r = \frac{F_{Br}}{F_{Zr}} \quad \text{Eq. 4-53}$$

Brake force distribution is said to be ideal, if the coefficients of friction at the front and rear axle for all the encountered friction ratios and thus for each retardation z , match. The optimum braking effect for all coefficients of friction of the road can be achieved only this way.

From the basic condition of the ideal brake-force distribution:

$$f_f = f_r = f_{id} \quad \text{Eq. 4-54}$$

the following results:

$$\begin{aligned} F_B &= z \cdot F_Z = F_{Bf} + F_{Br} \\ &= f_f \cdot F_{Zf} + f_r \cdot F_{Zr} \end{aligned} \quad \text{Eq. 4-55}$$

the brake force is given by:

$$F_B = f_{id} \cdot (F_{Zf} + F_{Zr}) \quad \text{Eq. 4-56}$$

When the aerodynamic influences on the axle loads are neglected, the following is applicable:

$$F_B = f_{id} \cdot F_Z \quad \text{Eq. 4-57}$$

such that the coefficients of friction on each axle correspond to the retardation:

$$z = f_{id} = f_f = f_r$$

Considering the axle load relations for the plane, influences of rotational masses not being considered, a relation for the ideal deceleration as a function of the front axle brake force F_{Bf} can be deduced:

$$z = \frac{F_{Bf}}{m \cdot g \cdot \left(\frac{l_r}{l} + \frac{h}{l} \cdot z \right)} \quad \text{Eq. 4-58}$$

z thus forms the solution of a quadratic equation:

$$z = -\frac{l_r}{2 \cdot h} + \sqrt{\left(\frac{l_r}{2 \cdot h} \right)^2 + \frac{F_{Bf}}{F_Z} \cdot \frac{l}{h}} \quad \text{Eq. 4-59}$$

For the brake force on the rear axle dependent on the brake force on the front axle, each relative to the vehicle weight, the following is applicable:

$$\frac{F_{Br}}{F_Z} = z - \frac{F_{Bf}}{F_Z} \quad \text{Eq. 4-60}$$

This results in:

$$\frac{F_{Br}}{F_Z} = -\frac{l_r}{2 \cdot h} + \sqrt{\left(\frac{l_r}{2 \cdot h} \right)^2 + \frac{F_{Bf}}{F_Z} \cdot \frac{l}{h}} - \frac{F_{Bf}}{F_Z} \quad \text{Eq. 4-61}$$

This parabolic ratio of the ideal brake-force distribution is determined by the coordinates of the center of gravity and a definite load.

Fig. 4-49 shows the course of the relative rear-axle brake force over the relative front-axle brake force in the so-called brake force diagram. The lines of constant retardation are also drawn in.

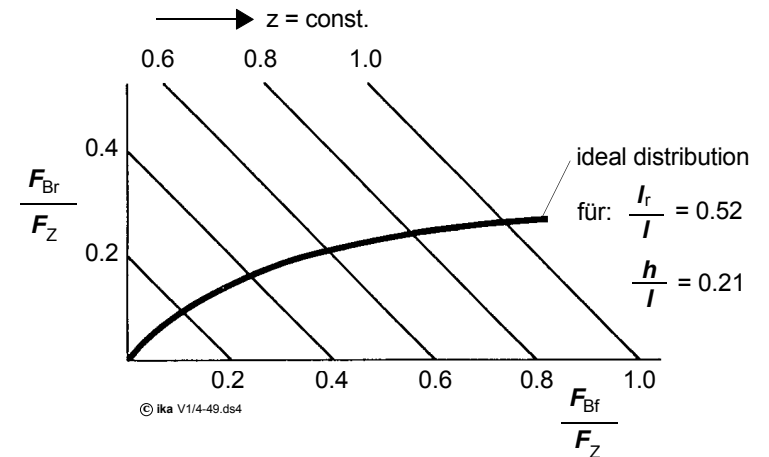


Fig. 4-49: Ideal brake-force distribution

For increasing retardation, the rear-axle brake force shows a degressive course since the load on the rear axle increasingly drops.

4.4.7 Brake-Force Distribution

The ideal brake-force distribution in vehicle braking systems can only be realized through considerable control-engineering effort. Generally systems where differing effort is required to tune the real distribution to the ideal one, are used.

These systems can be classified into:

- Rigid brake-force distribution,
- Brake-force control,
- Brake-force regulation.

4.4.7.1 Rigid brake-force distribution

In a vehicle, the brake-force distribution takes place without additional control measures over a fixed ratio dependent on the ratio of areas of the front and rear wheel brake cylinders and on the design in the case of stepped master cylinders.

A linear relation is indicated in the brake-force diagram (Fig. 4-50).

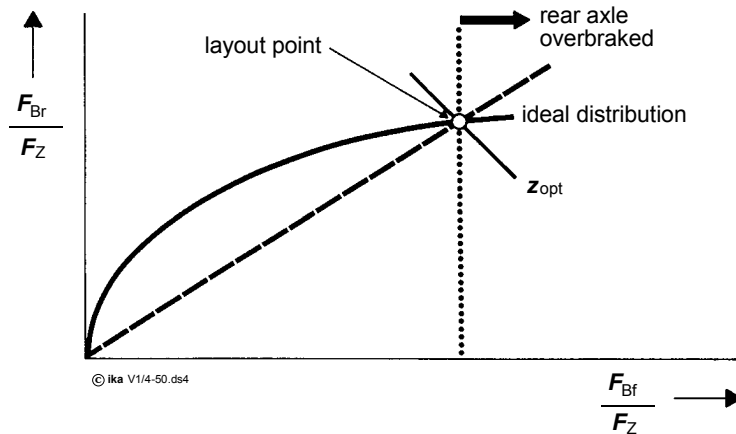


Fig. 4-50: Rigid brake-force distribution

Up to the point of intersection of the characteristic of rigid brake-force distribution with the ideal one, also the so-called layout point of the brake system, the front-axle brake force is greater than the ideal condition. Therefore at appropriate friction conditions, the front axle tends to overbrake resulting in the front wheels locking.

In retardations situated above the layout point, the rear axle tends to block. Only in the layout point there is the ideal case of even adhesion loading.

Most of all the blocking of the rear axle has to be avoided since otherwise, the vehicle begins to skid, while a vehicle with a locked front axle only loses its steering control but remains "driving-stable". Thus the EC guidelines prescribe that the layout point may only be exceeded at a retardation of $z=0.8$.

4.4.7.2 Brake-Force Control

The primary task of brake force control is to bring the characteristic of the real brake force distribution as close to the ideal one as possible. In order to achieve this, special valves are employed between the master cylinder and rear wheel brake cylinders. Based on the efficiency of these valves, one can distinguish between different layouts.

- Brake-force Limiter

The simplest possibility for brake-force control involves a brake-force limiter. It limits the rear-axle brake force to a maximum value (Fig. 4-51). In this way, the overbraking of the rear axle can be prevented.

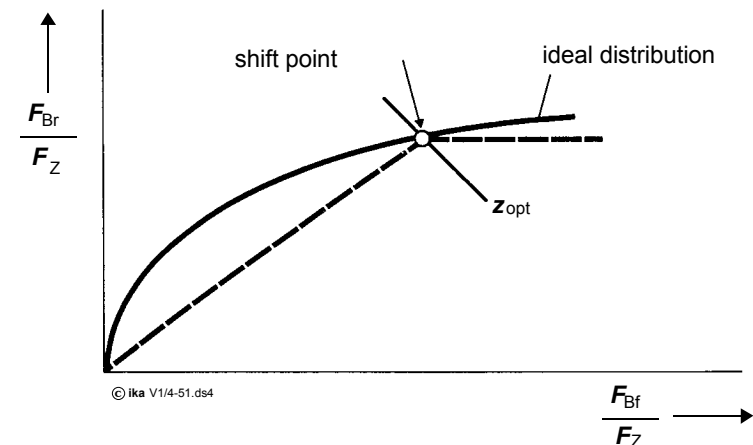


Fig. 4-51: Brake-force distribution when using a brake-force limiter

The brake-force limiter is realized by a positive pressure valve. At a certain intake pressure dependent on the spring layout, the valve closes such that the rear axle brake pressure also remains constant at further rising inlet pressures.

- Brake-force Reducer

An extensive approach of the real brake-force distribution to the ideal one is possible by the brake-force reducer. The course is shown in Fig. 4-52.

The tendency of the front axle to lock can be lowered compared to the brake-force limiter. The brake-force reduction is achieved by a valve which allows only certain increases in pressure to pass through to the outlet starting from a prescribed inlet pressure.

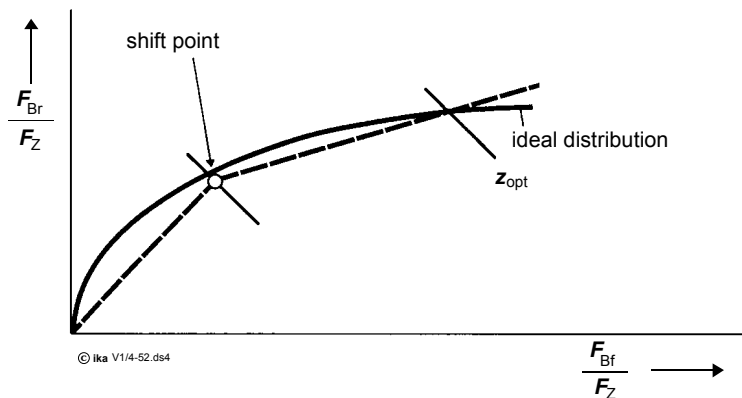


Fig. 4-52: Brake-force distribution when using a brake-force reducer

- Automatic Load-dependent Brake-Force Distribution

Changing loads lead to changes in the position of the center of gravity and in turn to a change in the ideal brake-force distribution. In order to maintain the desired approach of the characteristics of real and ideal brake-force distribution for different load conditions roughly constant, most of all in heavy commercial vehicles, automatic load-dependent brakes (ALB) are employed.

One can distinguish between “radiant” ALB and “deviating” ALB.

In the “radiant” ALB (Fig. 4-53) the basically rigid ratio of the front and rear axle brake forces is varied according to the load. In the “deviating” ALB (Fig. 4-54) the change-over point at which the brake force reduction or limitation comes into effect changes. The control of the load-dependent brakes basically takes place over the relative distance between the axle and vehicle body. In vehicles equipped with level control, pressure changes in the spring struts and bellows can be used for control.

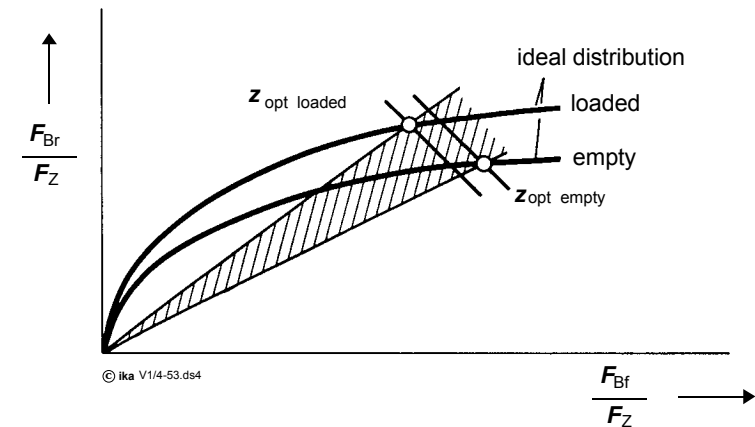


Fig. 4-53: Radiant ALB

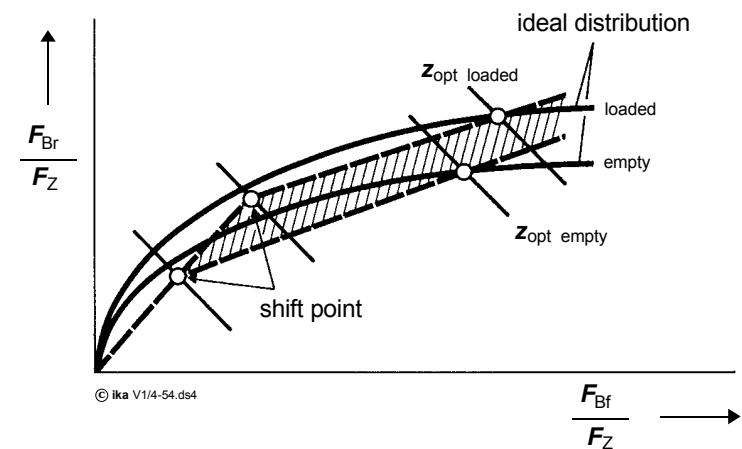


Fig. 4-54: Deviating ALB

The change in the position of the center of gravity due to load is especially significant in commercial vehicles. An automatic-load-dependent is prescribed by law (StVZO) for semitrailer trains and trailers with a permissible total weight over 7.5 t.

Fig. 4-55 shows a load-dependent brake-force reducer for a compressed-air braking system in commercial vehicles.

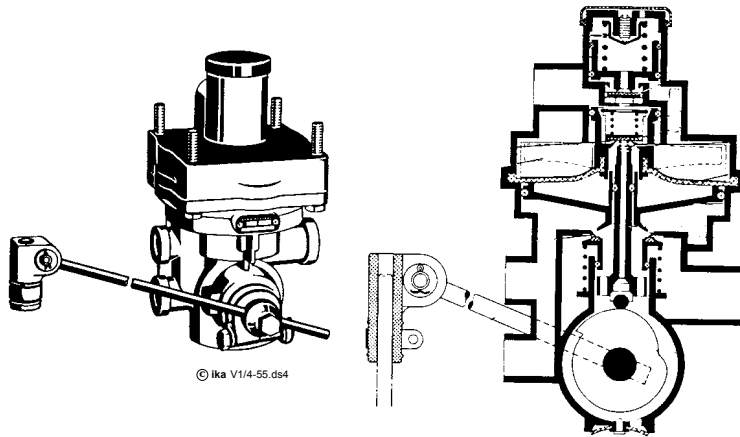


Fig. 4-55: Load-dependent brake-force reducer for compressed-air brake systems

4.4.7.3 Brake-Force Regulation (Antiskid Devices)

The above mentioned brake-force controls serve to bring the actual brake force distribution as close to the ideal one as possible, but are not however able to prevent the locking of the wheels.

The locking of a wheel ($s_B=1$) not only leads to a reduction of the possible brake force and thus to a retardation, but more importantly to a steep reduction of the transferable side force. When the front wheels lock, it implies a loss of steerability, while an overbraked rear axle implies an increased risk of spinning.

In Fig. 4-56, the characteristic of the brake force F_B as well as the maximum possible cornering force F_S on an antiskid road dependent on the brake-force slip s_B is qualitatively drawn in.

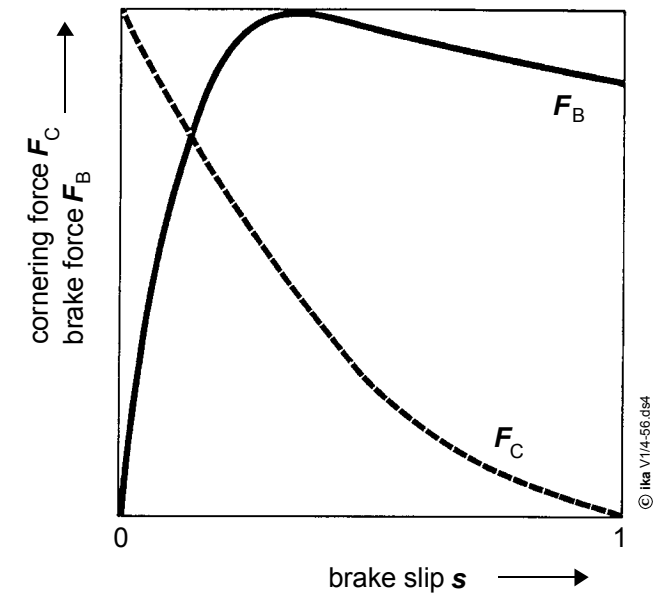


Fig. 4-56: Braking and cornering force as a function of the brake slip

Antiskid devices (ASD) limit the slip created by a pressure control when braking so that a high retardation can be achieved with a simultaneously good cornering force. The friction coefficient between tires and roadway and thus the possible brake force depends, however, on the tire model, slip angle, and to a high degree on the road condition. An antiskid device should show an adaptive control that can adjust itself to different tire and road conditions.

a) Control Variables

Since the more or less noticeable maxima of friction coefficients occur at a roughly constant brake slip, it would be the ideal control variable for antiskid devices. The slip between road and tires cannot however be directly detected. Other suitable control variables have to be found.

Fig. 4-57 illustrates the forces and moments acting on a braked wheel.

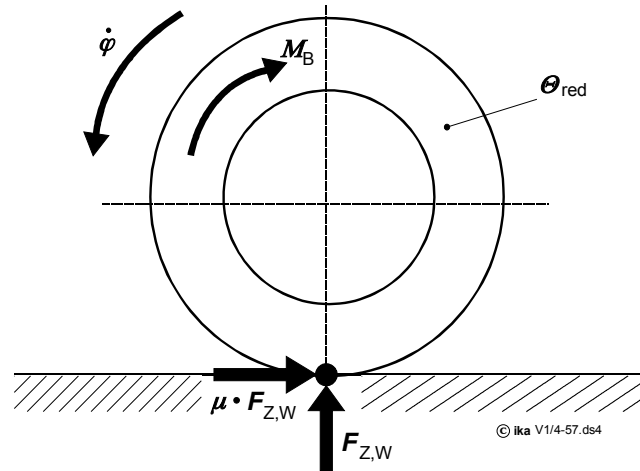


Fig. 4-57: Forces and moments on a braked wheel

From the sum of the moments, for the brake torque results:

$$M_B = \mu \cdot F_{Z,W} \cdot r_{dyn} - \Theta_{red} \cdot \ddot{\varphi}_W \quad \text{Eq. 4-62}$$

After having exceeded the maximum friction, the brake force in the tire tread area $\mu \cdot F_{Z,W}$ and also the resulting moment $\mu \cdot F_{Z,W} \cdot r_{dyn}$ reduces. At a constant braking torque, the reduction in the resultant moment can be compensated by a reduction of the angular acceleration of the wheel when braking, i.e., by an enlargement of the wheel rotational deceleration. Since the moment $\mu \cdot F_{Z,W} \cdot r_{dyn}$ is considerably larger than the moment $\Theta_{red} \cdot \ddot{\varphi}_W$, a small change of μ leads to a large change of $\ddot{\varphi}_W$. As a result, the wheel rotational deceleration $\ddot{\varphi}_W$ is a suitable control variable for antiskid devices.

Electronically controlled automatic antiskid devices generally detect the wheel rotational deceleration by a differentiation of speed signals detected on the wheel using speed sensors. In order to exactly control the retardation, additional variables are however necessary.

The "relative slip" is often used as a second control variable. From the wheel speeds of several wheels, by means of logical relationships, a reference speed is determined that corresponds to an appropriate wheel speed for optimum friction at the point of consideration. By comparing the actual wheel speed with the reference speed, the magnitude of the respective "relative" slip is determined.

Alternatively, the translational vehicle deceleration can be used as a second control variable which is used as a reference for the wheel rotational deceleration. This however creates the need for an accelerometer in the vehicle.

Special sensors that are integrated in the tires and directly measure the coefficient of friction between tires and road in the tire tread are presently in the development stage. It operates on the basis of a conclusion that the deformation of the profile elements emerging from the tire tread as a result of the tires rolling on a function of the local ratios of forces between tires and road. Fig. 4-58 shows the schematic design of such a sensor.

A displacement of the permanent magnet relative to the plate produces a change in the flow densities at the 4 hall generators. From the individual output signals, the tire deformations in x, y, and z-direction can then be determined.

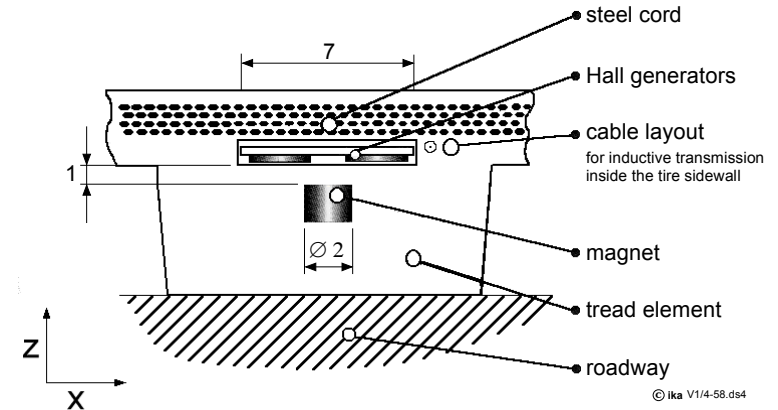


Fig. 4-58: Tire sensor for friction measurement

b) Control Procedure

In principle, the control cycle used in all brake force regulation over the wheel rotational deceleration is the same. The excess brake pressure created by the driver, is adjusted in such a way that the rotational deceleration of the wheel fluctuates between an upper and lower level. The methods used to realize as far as possible a secure maintenance of the limits required for optimum braking, free of external disturbances, are different according to the type of Anti-Skid Device (ASD) used.

As an example, Fig. 4-59 shows the control characteristic of ABS (Antilock Braking System) using the wheel rotational deceleration and relative slip as control variables.

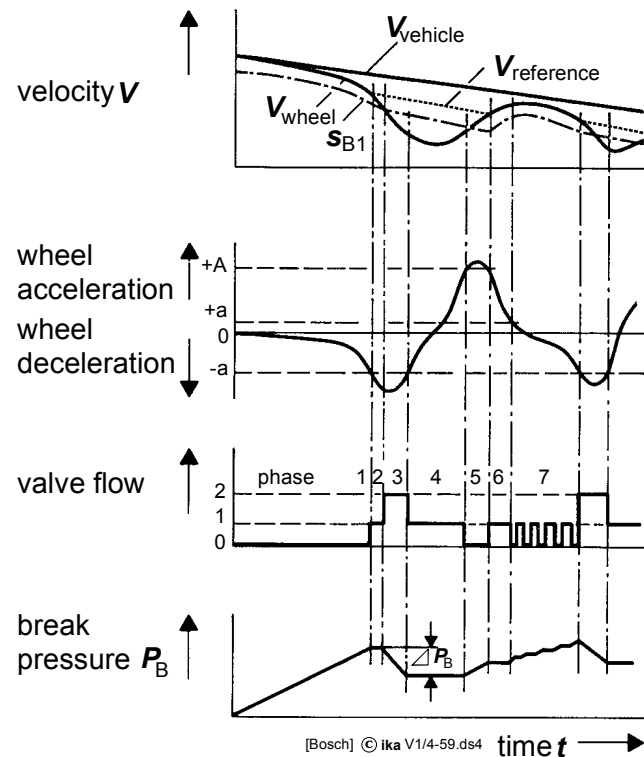


Fig. 4-59: Control process of ABS-control (Bosch)

Phase 1: Due to increasing brake pressure applied by the driver, the circumferential speed of the wheel falls steeply when compared to the vehicle speed.

Phase 2: If the wheel deceleration exceeds the step characteristic for reaching the maximum friction ('-a' in Fig. 4-59), the brake pressure is maintained at the actual value.

Phase 3: If the wheel speed falls below the slip characteristic s_{B1} , the brake pressure is lowered.

Phase 4: If step '-a' is reached again, the pressure is kept constant.

Phase 5: If the wheel acceleration exceeds the upper acceleration limit ('+A') in this phase of constant brake pressure, the brake pressure is increased again.

Phase 6: The brake pressure is kept constant between the acceleration signals '+A' and '+a', after which it is slowly increased until once again the '-a' wheel deceleration is reached.

Phase 7: A new control cycle starts, this time directly introduced by a reduction in pressure.

The pressure maintenance phase at the beginning of the control process serves to filter out eventual external disturbances. Since with large wheel moments of inertia, smaller coefficient of friction, and slow pressure increase in the wheel brake cylinder, the wheel can lock in spite of careful braking without having reached the wheel deceleration step '-a', the relative slip is included as second control variable of ABS. Reaching a certain slip value also creates a brake pressure reduction.

c) Control Strategies

Three different control strategies can presently be used in controlling the brake force on the wheels of an axle using antiskid devices. They are:

- Single-wheel control,
- Select-low control,
- Select-high control.

- Single-Wheel Control

If the maximum possible retardation of a vehicle should be realized, each wheel has to utilise the maximum possible friction. This is possible using single-wheel or individual control in which the control variable is measured using sensors at every wheel and its own brake pressure channel. The brake pressure is separately controlled for each wheel independent of the behavior of the remaining wheels of the vehicle.

On a one-sided smooth road (μ -split), due to strongly differing brake forces a high yaw moment confronts the advantage of the maximum brake capacity.

- Select-Low Control

In the select-low control, a common brake pressure is applied to both wheels on an axle. The magnitude of the brake pressure is determined by the wheel with the lower friction. This can be achieved by a sensor on each wheel or by a common sensor on the differential.

In select-low control, the friction of the wheel rolling on the road surface with a higher coefficient of friction is not entirely used. A high cornering force capability is however maintained. Yaw moments do not occur since roughly the same brake forces act on both wheels. For these reasons, the select-low control is suitable for the rear axle.

- Select-High Control

In the select-high control, the wheel with the higher friction determines the common brake pressure of both wheels on an axle. A sensor is however required for each wheel. The achievable retardation is higher than in the select-low controlled axle. The wheel rolling on the road surface with a lower coefficient of friction however, which leads to the loss of the cornering force. The forces unequally transferred on the road create a yaw moment. Due to the higher achievable retardation select-high controls are generally used for the front axle.

As a result of these control principles, numerous combinations result for a two-axle vehicle. Table 4-8 shows some variants with differing numbers of speed sensors and pressure control channels.

System Variants ☐ Control Channel ◀ Sensor	chan- nels	sen- sors	control		utilization of friction		steer- ability	driving stability
			FA	RA	FA	RA		
	4	4	single	single	++	++	+	-
	4	4	single	single	++	++	+	-
			single	select	++	-	+	-
			select high	low	+	-	-	+
	3	3	single	select	++	-	+	-
			select high	low	+	-	-	+
	2	3	select high	select low	+	+	-	-
	2	2	select low	select low	-	-	-	-
	2	2	select low		-	-	-	-
	1	1	select low		--	--	-	-

Table 4-8: System variants of Anti-Skid Devices

The single-wheel control on both axles in each case requires 4 sensors and 4 control channels. Antiskid devices with individually or select-high controlled front wheels and select-low controlled rear axle require 4 channels as well and sensors in a diagonal brake circuit distribution. In the standard distribution only three channels each are required.

The 2-channel variant with 3 sensors is usually controlled according to the select-high method. Steerability and driving stability are not guaranteed for μ -split conditions. In a diagonal brake circuit distribution, the brake pressure of a rear wheel can be controlled commonly with the diagonally opposite located and separately controlled front wheel. Also in this 2-channel system, steerability and driving stability are not guaranteed. 1-channel controls only guarantee driving stability on symmetrical road surfaces, while steering capability is not guaranteed. The braking distances are comparatively long.

d) Applied Examples

Applied examples of antiskid devices (ASD) can be classified into:

- Electronic systems,
- Mechanical systems.

Electronically controlled antiskid devices generally consist of three constructional groups, as can be seen in Fig. 4-60:

- Sensors (1) to detect the control variable,
- The Electronic Control Unit (2) which processes the sensor signals,
- The Hydraulic Control Unit (3) which converts the commands of the electronic control unit using magnetic valves into pressure changes at the wheel brakes.

Due to the high safety relevance of braking systems, the electronic control unit has to be designed as a redundant system. This system consisting of two identical microprocessors, recognizes failures and malfunctions by comparing control signals.

Mechanical antiskid devices were employed primarily from the point of view of costs. Since the control cycles are comparatively slow and the cost advantages have become smaller and smaller compared to electronically controlled systems, these systems are not used anymore.

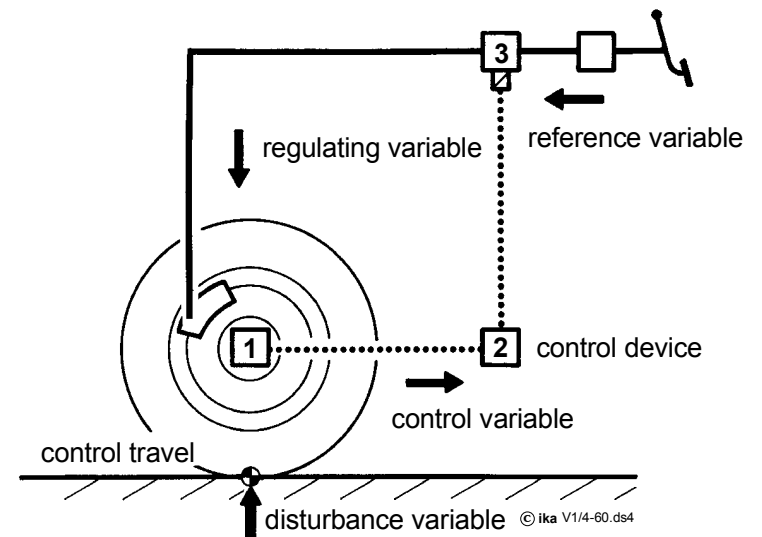


Fig. 4-60: Control loop of an electronic ASD

- Antilock Braking System

The Antilock Braking System (ABS, Bosch) works with individual front-wheel control and select-low control on the rear axle. It is installed in the brake system as an additional constructional element. According to the type of the brake circuit distribution, three or four brake valves and sensors are used. The 3-channel control consists of a common sensor for the rear wheels positioned at the differential. The sensors are speed detectors. The computer determines the rotational deceleration from the wheel speed along with the relative slip as a second variable. From the two control variables, the control commands are sent to the magnetic valves which have three possible positions: normal, pressure-hold, and pressure-reducing position (Fig. 4-61).

After every standstill, the electronic control unit and electronic parts of the control hydraulics are tested by a self-test for perfect operation. In case of failures, the ABS is turned off, the normal brake system remains completely functional. A warning lamp on the instrument panel indicates to the driver that the ABS system is out of order.

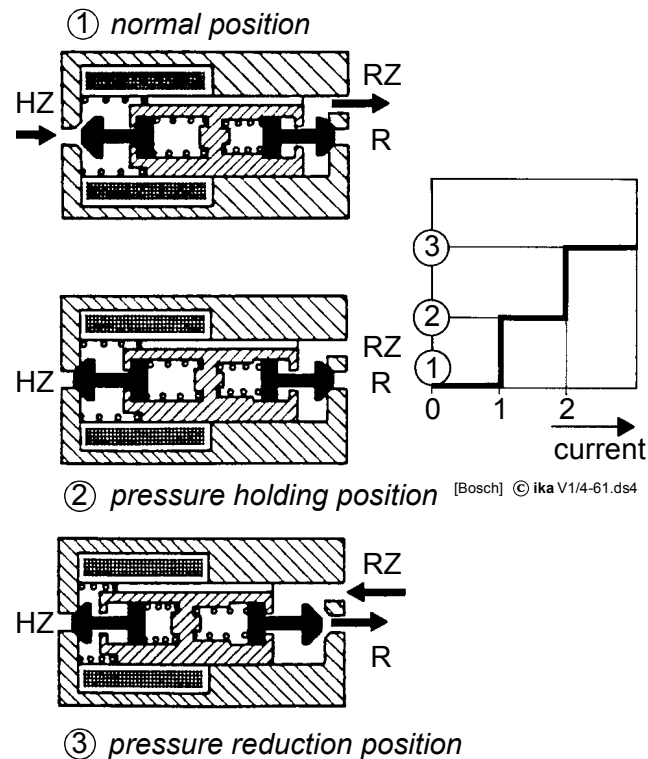


Fig. 4-61: Function of the ABS-3/3 magnet valve (Bosch)

- Anti-Skid System

In contrast to Bosch's "Add-on" system, Anti-Skid (AS, Teves) is an integrated system in which clutch operation, brake-force boost, and the antiblock control function are put together to form a compact hydraulic-aggregate (Fig. 4-62).

As in the previous case, this system is also individually controlled on the front wheels, while the wheels of the rear axle function on the select-low principle. A diaphragm reservoir is present in the energy supply which provides constant pressure and hence the possibility of quick reaction of the antilock function.

- Anti-Lock Braking System

The Anti-Lock-Brake (ALB, Honda) works with four inductive speed sensors and four magnet valves. The front axle is controlled commonly according to the select-high principle, the rear axle according to the select-low principle. This combination accepts the blocking of a front wheel in favor of higher retardation.

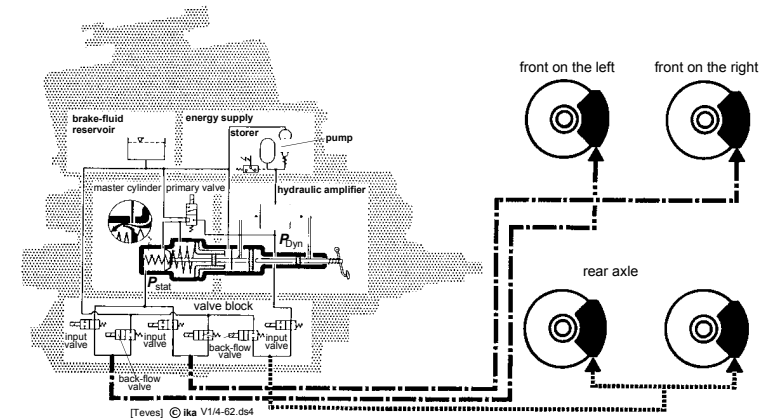


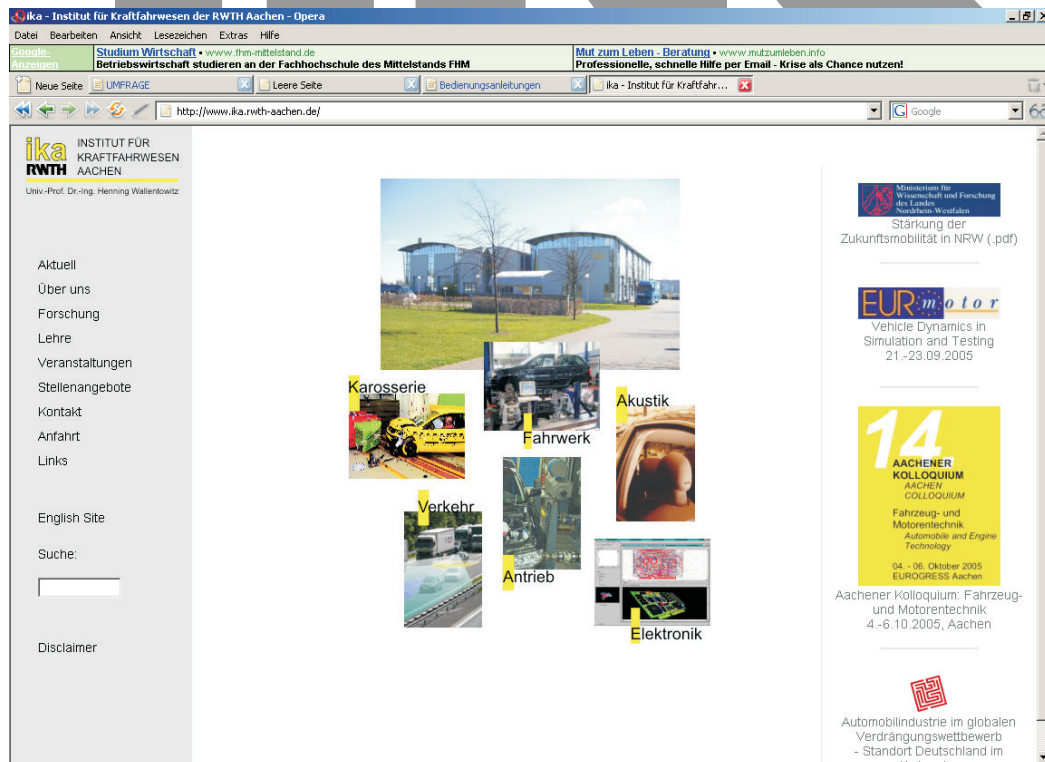
Fig. 4-62: Hydraulic-aggregate of the Anti-Skid-System (Teves)

- Anti-Skid-Braking System

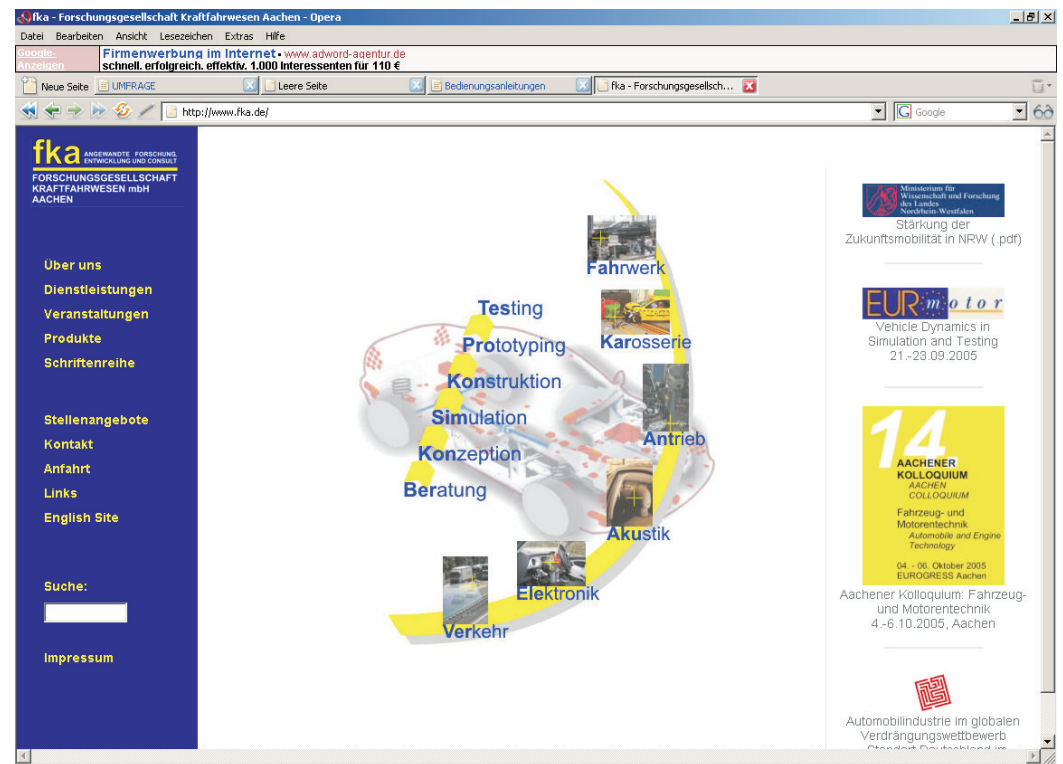
The Anti-Skid Braking System (ASBS; Mitsubishi) only has an effect on the rear wheels that are controlled commonly select-low. When it is fully braked, the front wheels block, the steerability is lost but the vehicle remains stable in its tracks. The systems has a speed sensor on the speedometer flexible shaft coupled to the transmission and an accelerometer that measures the translational vehicle deceleration as a reference variable.

In new ABS-systems, increasingly functions are integrated that serve for traction improvement and traction control. These systems are called Anti-Slip-Regulation (ASR). By a specific braking intervention, it puts a stop to individual driven wheels spinning (Fig. 4-63). According to the same principle, so-called "Driving Dynamics Controls" guarantee vehicle stability in critical areas during cornering. With the introduction of these systems the interface between longitudinal and transverse vehicle dynamics blurs.

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