and oil mist.
After flowing through the bearing points and sliding parts, the oil collects below the piston, connecting rod and crankshaft assembly in the oil pan (sump). The sump is a reservoir where the oil cools while the foam dissipates and settles. Engines subject to high loads are also fitted with an oil cooler. Engine service life can be prolonged dramatically by keeping the oil clean.

Oil filters

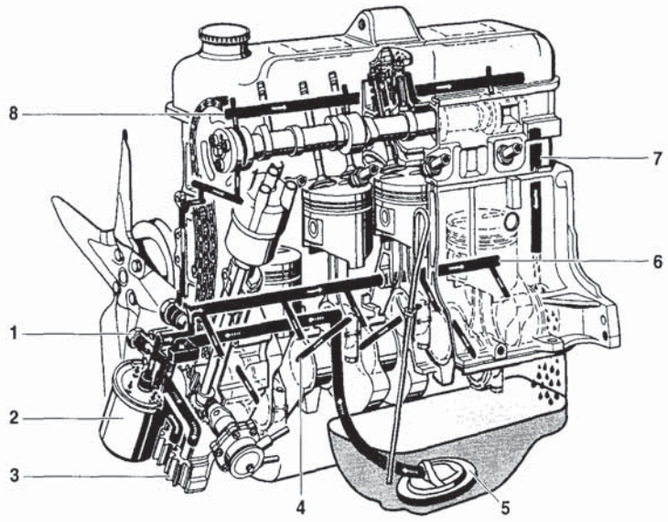
Oil filters remove solid particles from the engine oil (combustion residue, metal particles and dust) and maintain the lubricating capability of the oil in the intervals between changes. Oil-filter dimensions are governed by the degree to which impurities are generated in the engine and by the maintenance intervals prescribed by the engine manufacturer. Filter maintenance and oil changes should be performed at the same time.

Full-flow filters protect the entire oil circuit because particles which cause wear are trapped during their first pass through the circuit. Fine-mesh paper filters have proven effective as filtration elements in full-flow circuits. Their filtration is significantly finer than that furnished by strainers or disk filters. Full-flow filters must incorporate a bypass valve to prevent interruptions in oil supply should the filter get clogged. They should always be installed in the oil circuit downstream from the pressure-relief valve. Full-flow filters usually incorporate replaceable elements.

Bypass filters remove only about 5...10 % of the oil from the engine's lubricating system for subsequent return to the oil pan after filtering. Most bypass filters are of the fiber-filling type (deep-bed filter).

Such bypass filters are recommended for use only in conjunction with full-flow filters. The bypass filter can remove extremely fine particles (primarily soot) not previously extracted by the full-flow filter, reducing the concentration of micro-contaminants in the oil.

Force-feed lubrication system
1 Pressure relief valve 2 Oil filter, 3 Gear pump, 4 From main bearing to connecting-rod bearing, 5 Suction strainer, 6 Main oil-pressure line to crankshaft bearings, 7 Return flow from timing-gear case to crankcase, 8 To camshaft bearings.

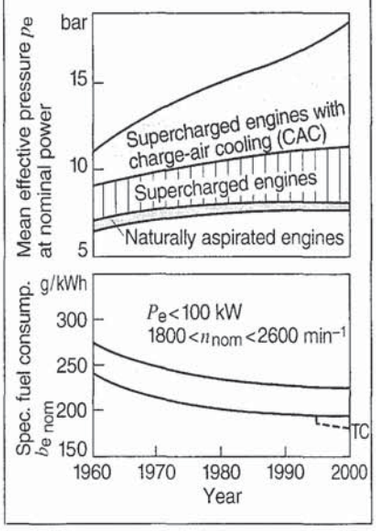


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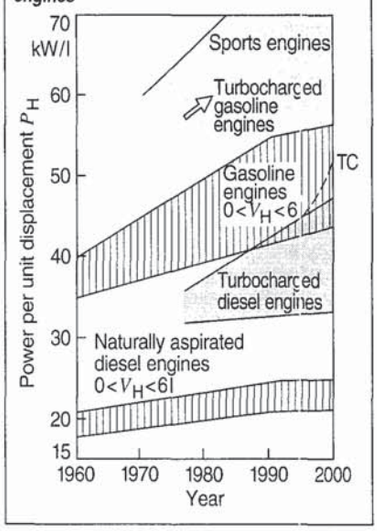
Empirical values and data for calculation

Comparisons

Mean effective pressure and fuel consumption[1]) trends for truck engines

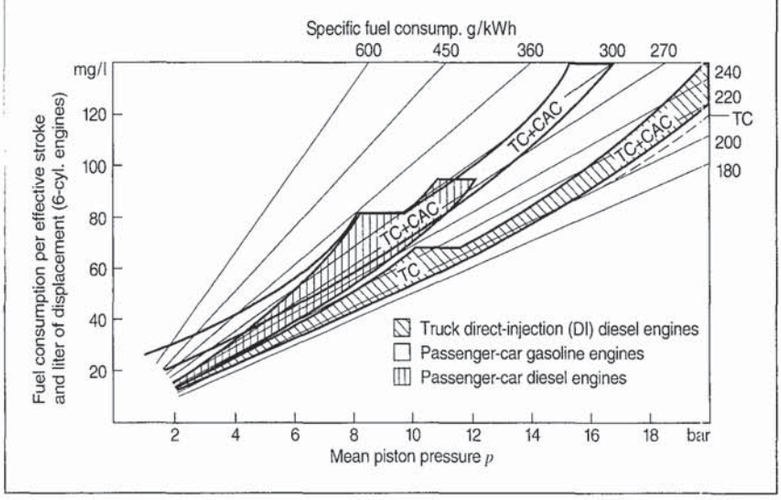


Trend in power per unit displacement for diesel and spark-ignition passenger-car engines



[1]) See P.337 for on-the vehicle measures influencing fuel consumption. TC = Turbo Compound.

Fuel consumption of diesel and spark-ignition engines[1])



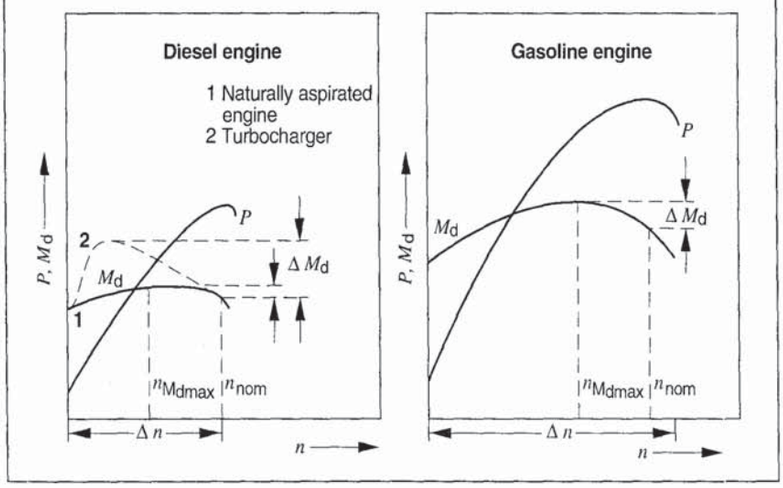
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Comparative data

Engine type		Engine speed min⁻¹	Compression ratio	Mean pressure bar	Power per liter kW/l	Weight-to-power ratio kg/kW	Fuel consumption g/kWh	Torque increase %
SI engine for Motor-cycles	2-stroke	4500...12000	7...9	5...12	40...100	5...0.5	600...350	5...10
	4-stroke	5000...10000	8...11	7...10	30...70	4...0.5	350...270	5...25
Pass. cars	NA[1])	4500...7500	8...12	8...11	35...65	3...1	350...250	15...25
	SC[2])	5000...7000	7...9	11...15	50...100	3...1	380...280	10...30
Trucks		2500...5000	7...9	8...10	20...30	6...3	380...270	15...25
Diesel engine for Pass. cars	NA[1])	3500...5000	20...24	7...9	20...35	5...3	320...240	10...15
	SC[2])	3500...4500	20...24	9...12	30...45	4...2	290...240	15...25
Trucks	NA[1])	2000...4000	16...18	7...10	10...20	9...4	240...210	10...15
	SC[2])	2000...3200	15...17	10...13	15...25	8...3	230...205	15...30
	CAC[3])	1800...2600	14...16	13...18	25...40	5...2,5	225...195	20...40
Special types Rotary engine		6000...8000	7...9	8...1	35...45	1.5...1	380...300	5...15
Stirling engine		2000...4500	4...6	–	–	10...7	300...240	20...40
Gas turbine		8000...70000	4...6	–	–	3...1	1000...300	50...100

[1]) Naturally aspirated engine, [2]) with supercharging, [3]) with charge air cooling/intercooling.

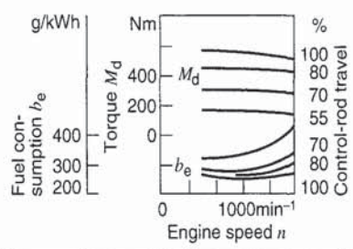
Power and torque curves



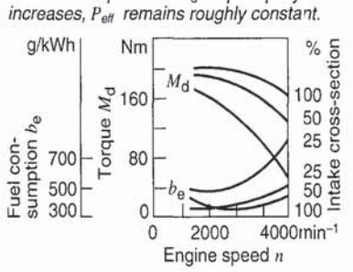
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Performance curves *(part-load behavior) for a specific control-rack travel or specific accelerator-pedal position*

Diesel engine with constant control-rack travel. M_d remains roughly constant with n.



Carburetor SI engine (4-stroke) with constant throttle position. M_d drops rapidly as n increases, P_{eff} remains roughly constant.



Torque position

The position on the engine-speed curve (relative to min^{-1} for max. output) at which maximum torque is developed, specified in % ($n_{Mdmax}/n_{nenn} \cdot 100$).

Useful speed range

(minimum full-load speed/nominal speed)

Engine type		Useful speed range Δn_N	Torque position %
Diesel engine for	pass. cars	3.5...5	15...40
	trucks	1.8...3.2	10...60
Spark-ignition engine		4...7	25...35

Torque increase

Engine type		Torque increase M_d in %
Diesel engine Pass. cars	Nat. asp. engine	15...20
	SC[1])	20...30
	SC[1]) + CAC[2])	25...35
Diesel engine Trucks	Nat. asp. engine	10...15
	SC[1])	15...30
	SC[1]) + CAC[2])	25...40
Spark-ignition engine	Nat. asp. engine	25...30
	SC[1]) + CAC[2])	30...35

[1]) with supercharging, [2]) with charge-air cooling/intercooling.

Engine output, atmospheric conditions

The torque and thus the power output of an internal-combustion engine are essentially determined by the calorific content of the cylinder charge. The amount of air (or, more precisely, of oxygen) in the cylinder charge provides a direct index of calorific content. The change which the engine will display at full power can be calculated as a function of variations in the condition of the ambient air (temperature, barometric pressure, humidity), provided that engine speed, air/fuel (A/F) ratio, volumetric efficiency, combustion efficiency, and total engine power loss remain constant. The A/F mixture responds to lower atmospheric density by becoming richer. The volumetric efficiency (pressure in cylinder at BDC relative to pressure in ambient atmosphere) only remains constant for all atmospheric conditions at maximum throttle-valve aperture (full-throttle). Combustion efficiency drops in cold thin air as vaporization rate, turbulence, and combustion speed all fall. Engine power loss (friction losses + gas-exchange work + boost power drain) reduces the indicated power.

Effect of atmospheric conditions

The quantity of air which an engine draws in, or is inputted to the engine by supercharging, depends upon the ambient atmosphere's density; colder, heavier, denser air increases engine output. Rule of thumb: Engine power drops by approximately 1% for each 100 m increase in altitude. Depending upon engine design, the cold intake air is normally heated to some degree while traversing the intake passages, thereby reducing its density and thus the engine's ultimate output. Humid air contains less oxygen than dry air and therefore produces lower

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engine power outputs. The decrease is generally modest to the point of insignificance. The warm humidity of air in tropical regions can result in a noticeable engine power loss.

Definitions of power

The effective power is the engine's power as measured at the crankshaft or ancillary mechanism (such as the transmission) at the specified min^{-1}. When measurements are made downstream from the transmission, the transmission losses must be factored into the equation. Rated power is the maximum effective power of the engine at full throttle. Net power corresponds to effective power.

Conversion formulas are used to convert the results of dynamometer testing to reflect standard conditions, thereby negating the influences of such factors as time of day and year while simultaneously allowing the various manufacturers to provide mutually comparable data. The procedure converts atmospheric density – and thus the effective volume of air in the engine – to defined "standard conditions" for air mass.

The comparison data in the following table show the most important standards used in power correction.

Power correction standards (Comparison)

Standard (Date of publication)	EEC 80/1269 (4/81)	ISO 1585 (5/82)	JIS D 1001 (10/82)	SAE J 1349 (5/85)	DIN 70 020 (11/76)
Barometric pressure during testing (* vapor pressure subtracted)					
Dry p_{PT}* kPa	99	99	99	99	–
Absolute p_{PF} kPa	– –	–	–	–	101.3
Temperature during testing					
Absolute T_p K	298	298	298	298	293
Engines with spark ignition, naturally-aspirated and turbo/supercharged					
Correction factor α_a	$\alpha_a = A^{1.2} \cdot B^{0.6}$ $A = 99/p_{PT}$ $B = T_p/298$				$\alpha_a = A \cdot B^{0.5}$ $A = 101.3/p_{PF}$ $B = T/293$
Corrected power: $P_0 = \alpha_a \cdot P$ (kW) (P measured power)					
Diesel engines, naturally-aspirated and turbo/supercharged					
Atmospheric correction factor f_a	$f_a = A \cdot B^{0.7}$ ($A = 99/p_{PT}$; $B = T_p/293$) (naturally-aspirated and mechanically-supercharged engines).				as α_a for SI engines
	$f_a = A^{0.7} \cdot B^{1.5}$ ($A = 99/p_{PT}$; $B = T_p/293$) (turbocharged engines with/without charge-air cooling)				
Engine correction factor f_m	$40 \le q/r \le 65$: $f_m = 0.036 \cdot (q/r) - 1.14$ $q/r < 40$: $f_m = 0.3$ $q/r > 65$: $f_m = 1.2$				$f_m = 1$
$r = p_L/p_E$ Boost pressure response, with p_L absolute boost pressure, p_E absolute pressure before compressor, q spec. fuel consumption (SAE J 1349). 4-stroke engines: q = 120,000 F/DN, 2-stroke engines: q = 60,000 F/DN, with F Fuel flow (mg/s); D Effective stroke volume (l); N Engine speed (min^{-1}).					
Corrected power: $P_0 = P \cdot f_a^{fm}$ (kW) (P measured power).					
Prescribed accessories					
Fan	Yes, with electric/viscous-drive fan at max. slip				Not defined
Emissions control system	Yes				
Alternator	Yes, loaded with engine-current draw				Yes
Servo pumps	No				No
Air conditioner	No				No

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Calculation

Quantity		Unit
a_K	Piston acceleration	m/s^2
B	Fuel consumption	kg/h; dm^3/h
b_e	Spec. fuel consumption	g/kWh
D	Cylinder diameter $2 \cdot r$	mm
d_v	Valve diameter	mm
F	Force	N
F_G	Gas force in the cylinder	N
F_N	Piston side thrust	N
F_o	Oscillating inertial force	N
F_r	Rotating inertial force	N
F_s	Rod force	N
F_T	Tangential force	N
M	Torque	N · m
M_o	Oscillating moments	N · m
M_r	Rotating moments	N · m
M_d	Engine torque	N · m
m_p	Weight-to-power ratio	kg/kW
n	Engine speed	min^{-1}
n_p	Injection-pump speed	min^{-1}
P	Power	kW
P_{eff}	Net horsepower[1])	kW
P_H	Power output per liter	kW/dm^3
p	Pressure	bar
p_c	Final compression pressure	bar
p_e	Mean effective pressure (mean pressure, mean working pressure)	bar
p_L	Boost pressure	bar
p_{max}	Peak cylinder pressure	bar
r	Crank radius	mm
s_d	Injection cross section of the nozzle	mm^2
S, s	Stroke, general	mm
s	Piston stroke	mm
s_t	Suction stroke of a cylinder (2-stroke)	mm
s_F	Suction stroke, 2-stroke engine	mm
S_k	Piston clearance from TDC	mm
S_s	Slot height, 2-stroke engine	mm
T	Temperature	°C, K
T_c	Final compression temperature	K
T_L	Boost-air temperature	K
T_{max}	Peak temperature in combustion chamber	K
t	Time	s
V	Volume	m^3

Symbol		Unit
V_c	Cylinder compression volume	dm^3
V_E	Injected quantity per pump stroke	mm^3
V_f	Charge volume of a cylinder (2-stroke)	dm^3
V_F	Charge volume of a 2-stroke engine	dm^3
V_h	Displacement of a cylinder	dm^3
V_H	Displacement of the engine	dm^3
v	Velocity	m/s
v_d	Mean velocity of the injected spray	m/s
v_g	Gas velocity	m/s
v_m	Mean piston velocity	m/s
v_{max}	Max. piston velocity	m/s
z	Number of cylinders	–
α_d	Injection period (in °crankshaft at injection pump)	°
β	Pivot angle of connecting rod	°
ε	Compression ratio	–
η	Efficiency	–
η_e	Net efficiency	–
η_{th}	Thermal efficiency	–
ν, n	Polytropic exponent of real gases	–
ϱ	Density	kg/m^3
φ, α	Crank angle (φ_o = top dead center)	°
ω	Angular velocity	rad/s
λ	$= r/l$ Stroke/connecting-rod ratio	–
λ	Air/fuel (A/F) ratio	–
$\varkappa$	$= c_p/c_v$ Adiabatic exponent of ideal gases	–

Superscripts and subscripts

0, 1, 2, 3, 4, 5	Cycle values/main values
o	Oscillating
r	Rotating
1st, 2nd	1st, 2nd order
A	Constant
′, ″	Subdivision of main values, derivations

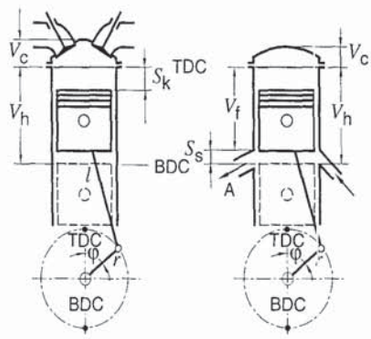
Conversion of units
(See PP. 19...28)

1 g/hp · h	= 1.36 g/kW · h
1 g/kW · h	= 0.735 g/hp · h
1 kp · m	= 9.81 N · m ≈ 10 N · m
1 N · m	= 0.102 kp · m ≈ 0.1 kp · m
1 hp	= 0.735 kW
1 kW	= 1.36 hp
1 at	= 0.981 bar ≈ 1 bar
1 bar	= 1.02 at ≈ 1 at

[1]) Effective power P_{eff} is the effective horsepower delivered by the internal-combustion engine, with it driving the auxiliary equipment necessary for operation (e.g., ignition equipment, fuel-injection pump, scavenging-air and cooling-air fan, water pump and fan, supercharger) (DIN 1940). This power is called net engine power in DIN 70 020 (see P. 415).

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Calculation equations

Mathematical relationship between quantities	Numerical relationship between quantities
Swept volume (displacement) Swept volume of a cylinder $V_h = \frac{\pi \cdot d^2 \cdot s}{4}$; $V_f = \frac{\pi \cdot d^2 \cdot s_F}{4}$ (2-stroke)	$V_H = 0.785 \cdot 10^{-6}\, d^2 \cdot s$ V_h in dm³, d in mm, s in mm
Swept volume of engine $V_H = V_h \cdot z$; $V_F = V_f \cdot z$ (2-stroke)	$V_H = 0.785 \cdot 10^{-6}\, d^2 \cdot s \cdot z$ V_h in dm³, d in mm, s in mm
Compression Compression ratio $\varepsilon = \frac{V_h + V_c}{V_c}$ (see P. 420 for diagram)	 ***4-stroke engine*** ***2-stroke engine***
Final compression pressure $p_c = p_o \cdot \varepsilon^v$	
Final compression temperature $T_c = T_o \cdot \varepsilon^{v-1}$	
Piston movement (see P. 421 for diagram) Piston clearance from top dead center $S_k = r\left[1 + \frac{l}{r} - \cos\varphi - \sqrt{\left(\frac{l}{r}\right)^2 - \sin^2\varphi}\right]$	
Crankshaft angle $\varphi = 2 \cdot \pi \cdot n \cdot t$ (φ in rad)	$\varphi = 6 \cdot n \cdot t$ φ in °, n in min⁻¹, t in s
Piston velocity (approximation) $v \approx 2 \cdot \pi \cdot n \cdot r\left(\sin\varphi + \frac{r}{2l}\sin 2\varphi\right)$	$v \approx \frac{n \cdot s}{19\,100}\left(\sin\varphi + \frac{r}{2l}\sin 2\varphi\right)$ v in m/s, n in min⁻¹, l, r and s in mm
Mean piston velocity $v_m = 2 \cdot n \cdot s$	$v_m = \frac{n \cdot s}{30\,000}$ (see P. 422 for diagram) v_m in m/s, n in min⁻¹, s in mm
Maximum piston velocity (approximate, if connecting rod is on a tangent with the big-end trajectory; $a_k = 0$) l/r | 3.5 | 4 | 4.5 v_{max} | $1.63 \cdot v_m$ | $1.62 \cdot v_m$ | $1.61 \cdot v_m$ (see P. 422 for diagram)	
Piston acceleration (approximation) $\alpha_k \approx 2 \cdot \pi^2 \cdot n^2 \cdot s\left(\cos\varphi + \frac{r}{l}\cos 2\varphi\right)$	$a_k \approx \frac{n^2 \cdot s}{182\,400}\left(\cos\varphi + \frac{r}{l}\cos 2\varphi\right)$ α_k in m/s², n in min⁻¹, l, r and s in mm

[1]) See P. 34 for definitions of "mathematical relationships between quantities" and "numerical relationships between quantities".

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Calculation equations (continued)

Mathematical relationship between quantities	Numerical relationship between quantities
Gas velocity Mean gas velocity in the valve section $v_g = \frac{d^2}{d_v^2} \cdot v_m$	$v_g = \frac{d^2}{d_v^2} \cdot \frac{n \cdot s}{30\,000}$ v_g in m/s, d, d_v, and s in mm, n in min^{-1}
The highest volumetric efficiency values are achieved at mean gas velocities of 90 ... 110 m/s (empirical values).	
Fuel supply Injected quantity per injection pump stroke $V_E = \frac{P_{eff} \cdot b_e}{\varrho \cdot n_p \cdot z}$	$V_E = \frac{1000 \cdot P_{eff} \cdot b_e}{60 \cdot \varrho \cdot n_p \cdot z}$ V_E in mm^3, P_{eff} in kW, b_e in g/kW · h (or also P_{eff} in hp, b_e in g/hp · h), n_p in min^{-1}, ϱ in kg/dm^3 (for fuels $\varrho \approx 0.85$ kg/dm^3)
Mean velocity of injection spray $v_d = \frac{2 \cdot \pi \cdot n_p \cdot V_E}{S_d \cdot \alpha_d}$ (α_d in rad)	$v_d = \frac{6\, n_p \cdot V_E}{1000 \cdot S_d \cdot \alpha_d}$ v_d = in m/s, n_p in min^{-1}, V_E in mm^3, S_d in mm^2, α_d in °
Engine power $P = M \cdot \omega = 2 \cdot \pi \cdot M \cdot n$	$P = M \cdot n/9549$ P in kW, M in N · m (= W · s),
$P_{eff} = V_H \cdot p_e \cdot n/K$ K = 1 for 2-stroke engine K = 2 for 4-stroke engine	$P_{eff} = \frac{V_H \cdot p_e \cdot n}{K \cdot 600} = \frac{M_d \cdot n}{9549}$ P_{eff} in kW, p_e in bar, n in min^{-1}. M_d in N · m
Power per unit displacement (power output per liter) $P_H = \frac{P_{eff}}{V_H}$	$P = M \cdot n/716.2$ P in hp, M in kp · m, n in min^{-1}
Weight-to-power ratio $m_p = \frac{m}{P_{eff}}$	

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Calculation equations (continued)

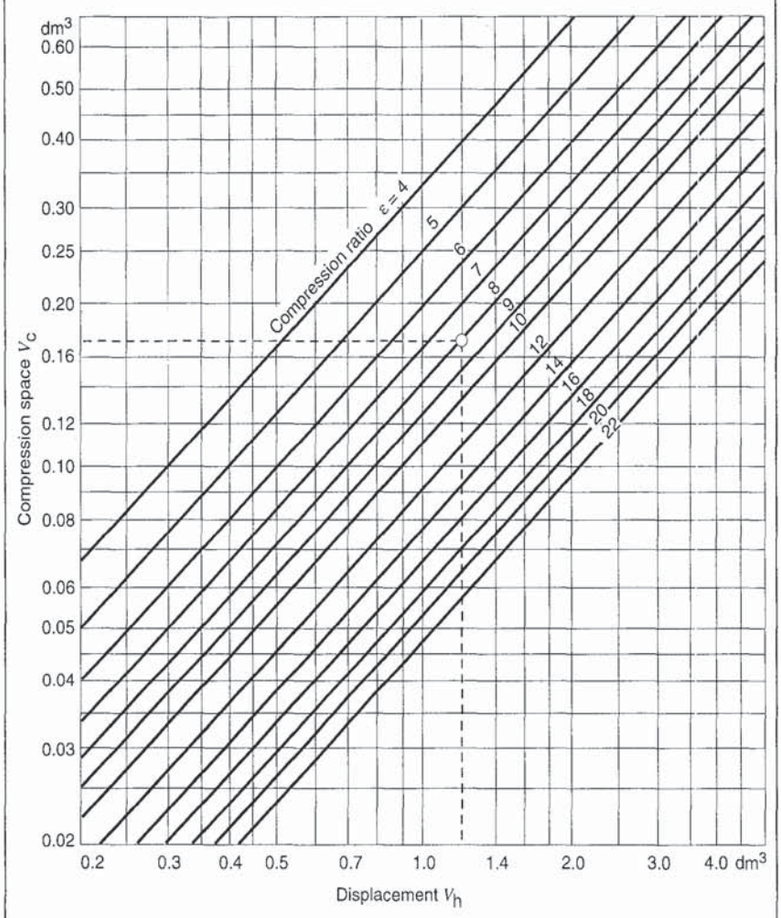
Mathematical relationship between quantities		Numerical relationship between quantities	
Mean piston pressure (mean pressure, mean working pressure)			
4-stroke engine $p = \frac{2 \cdot P}{V_H \cdot n}$	2-stroke engine $p = \frac{P}{V_H \cdot n}$	4-stroke engine $p = 1200 \frac{P}{V_H \cdot n}$	2-stroke engine $p = 600 \frac{P}{V_H \cdot n}$
		p in bar, P in kW, V_H in dm³, n in min⁻¹	
		$p = 833 \frac{P}{V_H \cdot n}$	$p = 441 \frac{P}{V_H \cdot n}$
		p in bar, P in hp, V_H in dm³, n in min⁻¹	
$p = \frac{4 \cdot \pi \cdot M}{V_H}$	$p = \frac{2 \cdot \pi \cdot M}{V_H}$	$p = 0.1257 \frac{M}{V_H}$	$p = 0.0628 \frac{M}{V_H}$
		p in bar, M in N · m, V_H in dm³	
Engine torque			
$M_d = \frac{V_H \cdot p_e}{4 \pi}$	$M_d = \frac{V_H \cdot p_e}{2 \pi}$	$M_d = \frac{V_H \cdot p_e}{0.12566}$	$M_d = \frac{V_H \cdot p_e}{0.06284}$
		M_d in N · m, V_H in dm³, p_e in bar	
		$M_d = 9549 \cdot P_{eff}/n$	
		M_d in N · m, P_{eff} in kW, n in min⁻¹	
Fuel consumption[1]			
B = Measured values in kg/h $b_e = B/P_{eff}$ $b_e = 1/(H_u \cdot \eta_e)$		B in dm³/h or kg/h V_B = Measured volume on test dynamometer t_B = Elapsed time for measured volume consumption $b_e = \frac{V_b \cdot \varrho_B \cdot 3600}{t_B \cdot P_{eff}}$ ϱ_B = Fuel density in g/cm³, t_B in s, V_B in cm³, P_{eff} in kW.	
Efficiency			
$\eta_{th} = 1 - \varepsilon^{1-\nu}$ $\eta_e = P_{eff}/(B \cdot H_u)$		$\eta_e = 86/b_e$ where H_u = specific calorific value 42,000 kJ/kg b_e in g/(kW · h)	

[1]) See P. 337 for the effect of vehicle design measures on fuel consumption.

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Displacement and compression area *See P. 417 for diagram and equation.*

The diagram below applies to the displacement V_h and compression space V_c of the individual cylinder, and to the total displacement V_H and total compression space V_C.

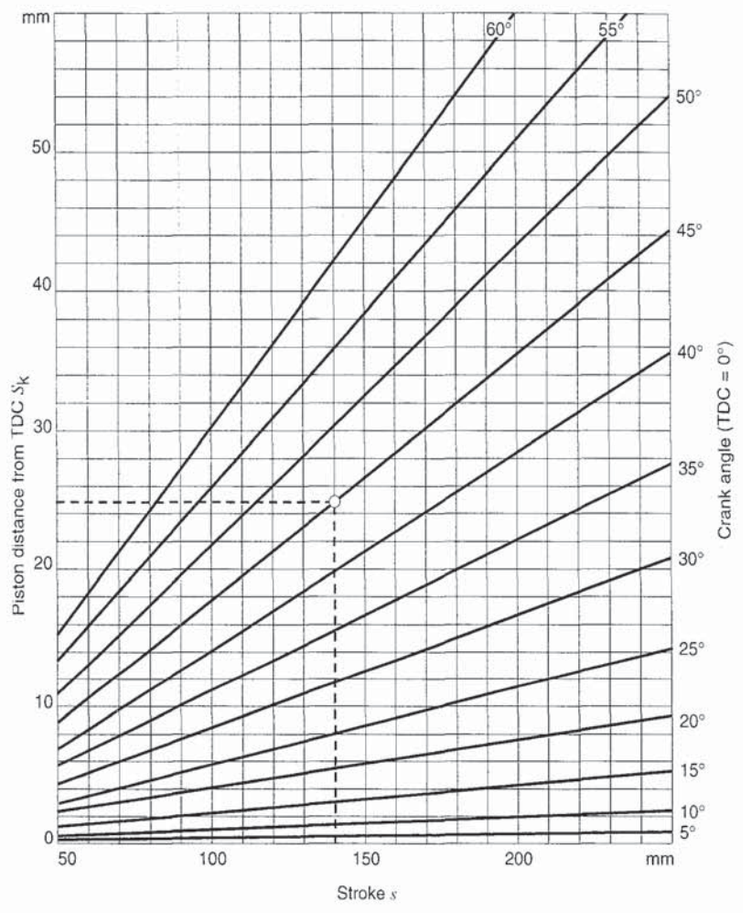


Example:
An engine with a displacement of 1.2 dm³ and a compression ratio ε = 8 has a compression area of 0.17 dm³.

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Piston clearance from top dead center *See P. 417 for equation.*

Conversion of degrees crank angle to mm piston travel.

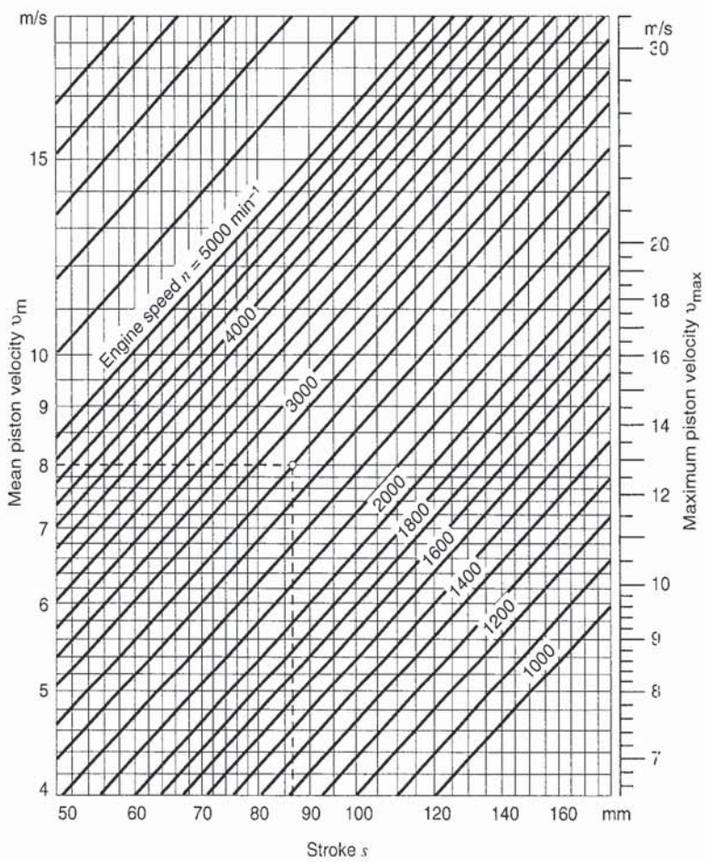


Example:
The piston clearance from top dead center is 25 mm for a stroke of 140 mm at 45° crankshaft.

The diagram is based on a crank ratio $l/r = 4$ (l connecting-rod length, r one half of the stroke length). However, it also applies with very good approximation (error less than 2%) for all ratios l/r between 3.5 and 4.5.

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Piston velocity *See P. 417 for equation.*



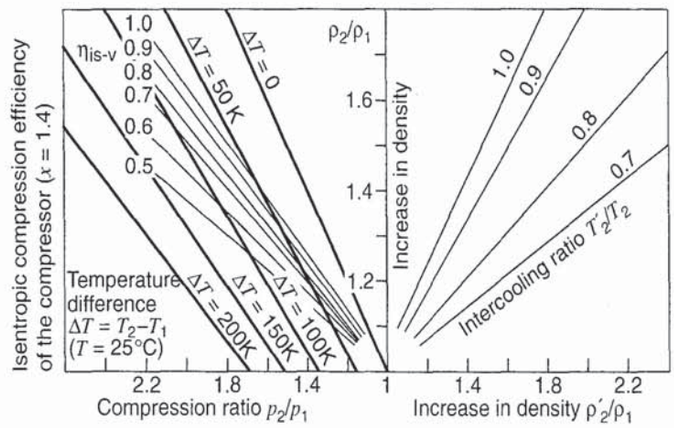
Example:
Mean piston velocity v_m = 8 m/s and maximum piston velocity v_{max} = 13 m/s for stroke s = 86 mm and at an engine speed of n = 2800 min^{-1}.

The diagram is based on $v_{max} = 1.62\ v_m$ (see P. 417).

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Charge density increase in the cylinder with turbo/supercharging

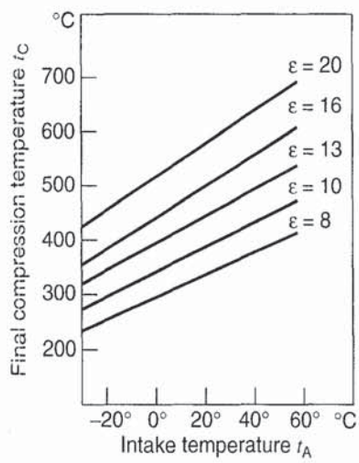
Increase in density on supercharging as a function of the pressure ratio in the compressor, the compressor efficiency and the intercooling rate for charge-air cooling (CAC).



$p_2/p_1 = \pi_c$ = *Pressure ratio during crankcase compression,* ϱ_2/ϱ_1 = *Increase in density,*
ϱ_1 = *Density upstream of compressor,* ϱ_2 = *Density downstream of compressor in kg/m³*
T_2'/T_2 = *Intercooling rate,* T_2 = *Temperature before CAC,* T_2' = *Temperature after CAC in K*
η_{is-v} = *Isentropic compressor efficiency*

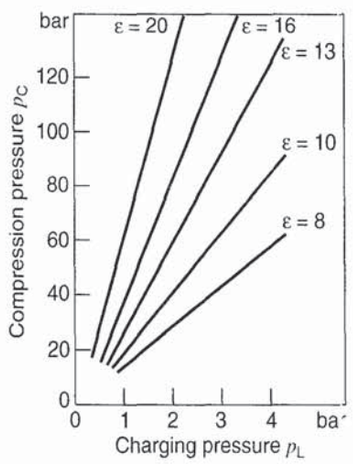
Final compression pressure and temperature

Final compression temperature as a function of the compression ratio and the intake temperature.



$t_c = T_c - 273.15K$, $T_c = T_A \cdot \varepsilon^{n-1}$, $n = 1.35$

Final compression pressure as a function of the compression ratio and boost pressure.



$p_c = p_L \cdot \varepsilon^n$, $n = 1.35$

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Reciprocating-piston engine with external combustion (Stirling engine)

Operating concept and efficiency

Cycle sequence: In phase I, the power piston (bottom), or working piston, is at its lowest position and the displacer (top piston) is in its highest; all of the working gas is expanded in the "cold" area between the two pistons. During the transition from phase I to phase II, the power piston compresses the working medium in the cold chamber. The displacer remains in its uppermost position. During the transition from phase II to phase III, the displacer moves down to push the compressed working medium through the heat exchanger and into the regenerator (where it absorbs the stored heat) and from there into the heater (where it is heated to maximum working temperature). Because the power piston remains in its lowermost position, the volume does not change.

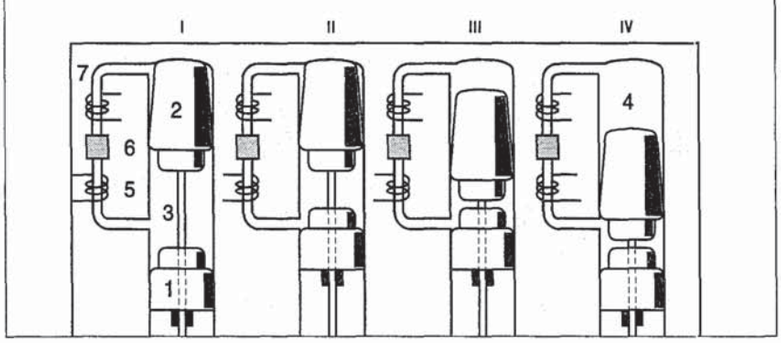
After being heated the gas enters the "hot" area above the displacer. During the transition from phase III to phase IV the hot gas expands; the power piston and displacer are pushed into their lowermost positions and power is produced. The cycle is completed with the transition from phase IV back to phase I, where the displacer's upward motion again propels the gas through the heater and into the regenerator, radiating substantial heat in the process. The residual heat is extracted at the heat exchanger before the gas reenters the cold area.

Thus the theoretical cycle largely corresponds to isothermal compression (the working gas is cooled back down to its initial temperature in the heat exchanger after adiabatic compression), isochoric heat addition via the regenerator and heater, quasi-isothermal expansion (the working gas is reheated to its initial condition in the heater after adiabatic expansion) and isochoric heat dissipation via the regenerator and heat exchanger.

The ideal cycle shown in the p-V and T-S diagrams could only be achieved if – as described – the movement of the power and displacer systems were discontinuous.

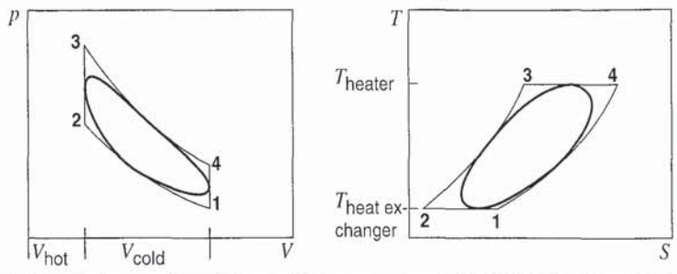
If both pistons are connected to a shaft, i.e., via a rhombic drive, they carry out phase-shifted sinusoidal movements leading to a rounded work diagram with the same cycle efficiency – similar to the efficiency of the Carnot cycle – but with sacrifices in power and net efficiency.

The Stirling engine cycle
Four states of discontinuous power-piston and displacer movement. 1 Power piston, 2 Displacer, 3 Cold space, 4 Hot space, 5 Heat exchanger, 6 Regenerator, 7 Heater.



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Theoretical Stirling engine cycle as shown in the p-V and T-S diagrams.



Design and operating characteristics

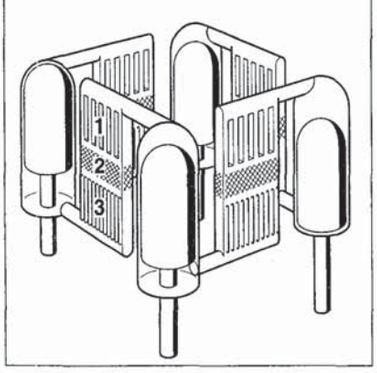
Modern Stirling engines are double-acting engines with (for instance) 4 cylinders operating with a defined phase shift. Each cylinder has only one piston whose top surface acts as a power piston, and whose bottom surface acts as a displacer for the following cylinder. The heat exchanger, regenerator and heater are located between the cylinders.

In order to maintain an acceptable ratio between power and displacement, the engines run at high pressures of 50 to 200 bar, variable for purposes of load control. Gases with low flow losses and high specific heats (usually hydrogen) must be used as working fluids. Because the heat exchanger must transfer all of the heat which is to be extracted from the process to the outside air, Stirling engines require considerably larger heat exchangers than IC-engines.

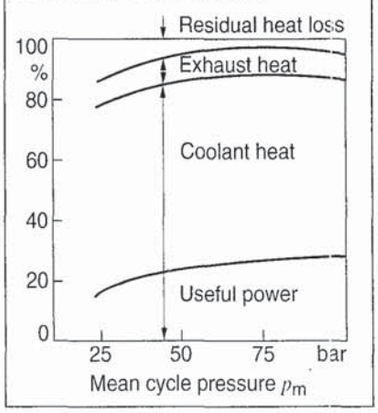
Advantages of the Stirling engine: very low concentrations of all the pollutants which are subject to legislation (HC, CO and NO_x); quiet operation without combustion noise; burns a wide variety of different fuels (multifuel capability); fuel consumption (in program map) roughly equivalent to that of a direct-injection diesel engine at comparable speeds.

Disadvantages: high manufacturing costs due to complicated design; very high operating pressures with only moderate power output relative to the unit's volume and weight; expensive load-control system required; large cooling surface and/or ventilation power required.

Double-action Stirling engine
1 Heater, 2 Regenerator, 3 Heat exchanger.



Heat balance of Stirling engine



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Wankel rotary engine

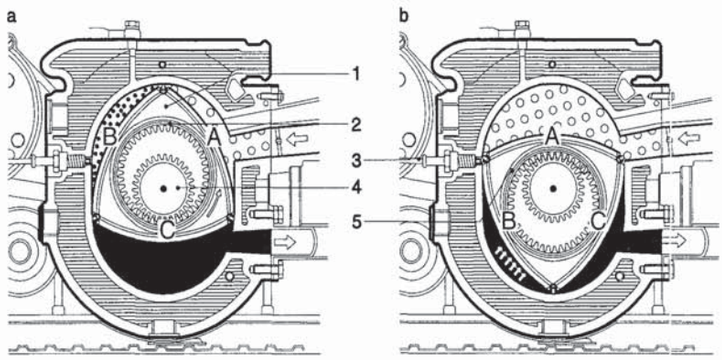
The rotary engine is an unconventional piston powerplant in which the crankshaft mechanism is replaced by an eccentric drive unit operated by a rotary piston. The piston forms the combustion chambers as it proceeds through its trochoidal rotation pattern.

Viewed from the side, the rotor is a triangle with convex sides. Located within the water-cooled housing is the oval – or more precisely: hourglass shaped – piston chamber (epitrochoid). As the rotor turns, its three apices follow the wall of the housing to form three mutually-sealed, variable displacement chambers (A, B and C) spaced at 120° intervals. Each of these chambers hosts a complete four-stroke combustion cycle during each full rotation of the rotor, i.e. after one full rotation of the triangular rotor, the engine has completed the four-stroke cycle three times and the eccentric shaft has completed an equal number of rotations.

The rotor is equipped with both face and apex seals. It incorporates a concentric internal ring gear and the bearings for the engine shaft's eccentric. The rotary piston's internal ring gear turns against a housing-mounted gear: this gear runs concentrically relative to the eccentric shaft. This gear set transmits no force. Instead, it maintains the rotary piston in the trochoidal orbit pattern required to synchronize piston and eccentric shaft.

The teeth of the gear set act in accordance with a 3:2 conversion ratio. The rotor turns at two thirds of the shaft's angular velocity and in the opposite direction. This arrangement produces a relative rotor velocity that is only one third of the shaft's angular velocity relative to the housing.

Design and operating concept of the Wankel rotary engine
1 Rotor, 2 Internal gearing in rotor, 3 Spark plug, 4 Fixed pinion, 5 Running surface of eccentric.
a) Cell **A** *takes in air-fuel mixture, cell* **B** *compresses the mixture, and the combustion gases* **C** *are exhausted from cell C. (Depressions in the rotor flanks allow gas to pass by the trochoidal restriction.)*
b) Cell **A** *is filled with fresh gas, the combustion gases expand in cell* **B**, *thereby turning the eccentric shaft via the rotor, combustion gases continue to be exhausted from cell* **C**. *The next phase of combustion is again that shown in figure a), whereby cell* **C** *has taken the place of cell* **A**. *Thus the rotor, by turning through 120° of one rotation, has carried out the complete four-stroke process at its three flanks. During this process, the eccentric shaft has made one complete rotation.*



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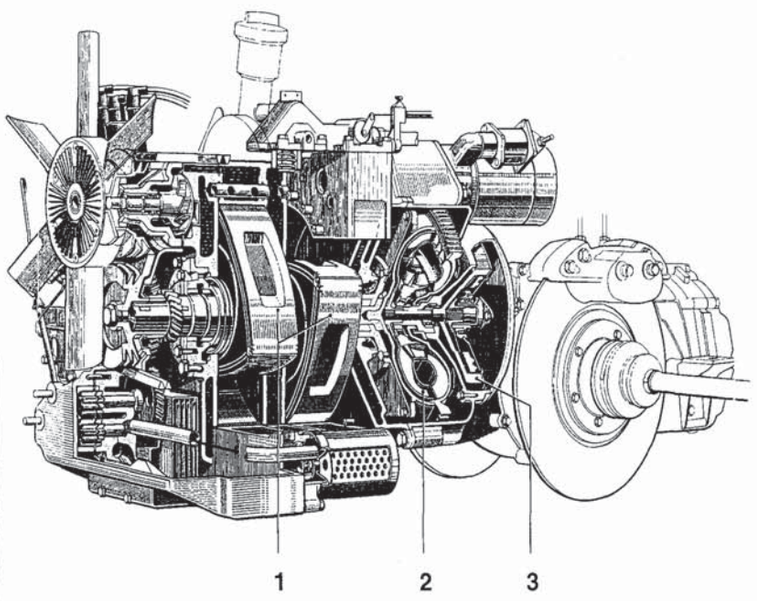
Gas exchange is regulated by the piston itself as it moves past slots in the housing. An alternative to this arrangement – with peripheral intake ports located along the apices' trochoidal path – is represented by intake ports in the side of the block (side ports).

Considerably higher gas velocities and engine speeds are made possible by the lack of restrictive gas passages and the absence of reciprocating masses. Every rotary engine can be completely balanced mechanically. The only remaining irregularity is the uneven torque flow, a characteristic of all internal-combustion engines. However, the torsional force curve of a single-rotor engine is considerably smoother than that of a conventional single-cylinder engine, due to the fact that the power strokes occur over 270° of the eccentric shaft's rotation. Power-flow consistency and operating smoothness can both be enhanced by joining several rotary pistons on a single shaft. In this context, a three-rotor Wankel corresponds to an eight-cylinder, reciprocating-piston engine. The torque curve can be made to assume the characteristics of a throttled engine or a racing engine by changing the timing (location) and shape of the intake ports.

Advantages of the rotary engine: complete balancing of masses; favorable torsional force curve; compact design; no valve-gear assembly; excellent tractability.

Disadvantages: less than optimal combustion chamber shape with long flame paths, high HC emissions, increased fuel and oil consumption, higher manufacturing costs, diesel operation not possible, high location of output shaft.

Design of a twin rotary engine
1 Rotor, 2 Hydraulic torque convertor, 3 Automatic clutch.



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Gas turbine

In the gas turbine, the individual changes of state during the cycle take place in spatially separate components (compressors, burners and turbines), which communicate with one another via flow-conducting components (diffusers, spirals and the like). These changes of state therefore occur continuously.

Operating concept, comparative cycle and efficiency

In automotive gas turbines, the intake air is drawn in continuously through a filter and noise attenuator prior to condensation in a radial compressor and subsequent warming in a heat exchanger. The heat exchanger in current automotive units is usually designed as a rotating regenerator.

The compressed and preheated air then flows into the burner where it is directly heated through injection and combustion of gaseous, liquid or emulsified fuels. Energy from the compressed and heated gases is then transmitted to one, two or three turbine stages on one to three shaft assemblies. The radial- or axial-flow turbines initially drive compressors and auxiliary assemblies, which then relay the remaining power to the driveshaft via a power turbine, reduction gear and transmission.

The turbine usually incorporates adjustable guide vanes (AGV turbine) designed to reduce fuel consumption at idle and in part-load operation whilst simultaneously enhancing tractability during acceleration. In single-shaft machines, this necessitates an adjustment mechanism at the transmission.

After partial cooling in the expansion phase the gases flow through the gas section of the heat exchanger, where most of their residual heat is discharged into the air. The gases themselves are then expelled through the exhaust passage, whereby they can also supply heat for the vehicle's heating system.

The thermal efficiency and with it the fuel consumption of the gas turbine are largely determined by the maximum possible operating temperature (burner exit temperature). The temperatures that can be achieved using highly heat-resistant cobalt- or nickel-based alloys do not allow fuel consumption that is comparable with present-day piston engines. It will need the changeover to ceramic materials before similar or even better fuel efficiency can be achieved.

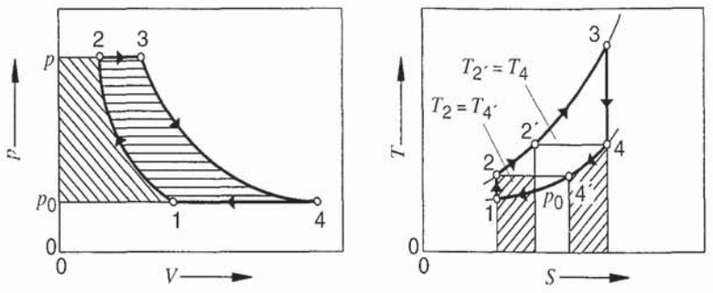
The comparative thermodynamic cycle for the gas turbine is the constant-pressure or Joule cycle. It consists of isentropic compression (process 1 → 2), isobaric heat addition (process 2 → 3), isentropic expansion (process 3 → 4) and isobaric heat dissipation (process 4 →1). High levels of thermal efficiency are only available when the temperature increase from T_2 to $T_{2'}$, supplied by the heat exchanger, is coupled with a thermal discharge (4 → 4'). If heat is completely

Characteristic operating temperatures (orders of magnitude) at various positions in metallic and ceramic automotive gas turbines at full load

Measuring point	Metal turbine	Ceramic turbine
Compressor exit	230 °C	250 °C
Heat-exchanger exit (air side)	700 °C	950 °C
Burner exit	1000 °C...1100 °C	1250 °C...1350 °C
Heat-exchanger inlet (gas side)	750 °C	1000 °C
Heat-exchanger exit	270 °C	300 °C

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Thermodynamic comparative cycle as shown in the p-V and T-S diagrams.



exchanged, the quantity of heat to be added per unit of gas is reduced to

$$q_{add} = c_p \cdot (T_3 - T_{2'}) = c_p \cdot (T_3 - T_4)$$

and the quantity of heat to be removed is

$$q_{rem} = c_p \cdot (T_{4'} - T_1) = c_p \cdot (T_2 - T_1).$$

The maximum thermal efficiency for the gas turbine with heat exchanger is:

$$\eta_{th} = 1 - Q_{rem}/Q_{add} = 1 - (T_2 - T_1)/(T_3 - T_4)$$

Where $p_2/p_1 = (T_2/T_1)^{\frac{\chi}{\chi-1}} = (T_3/T_4)^{\frac{\chi}{\chi-1}}$
and $T_4 = T_3 \cdot (T_1/T_2)$ it follows that

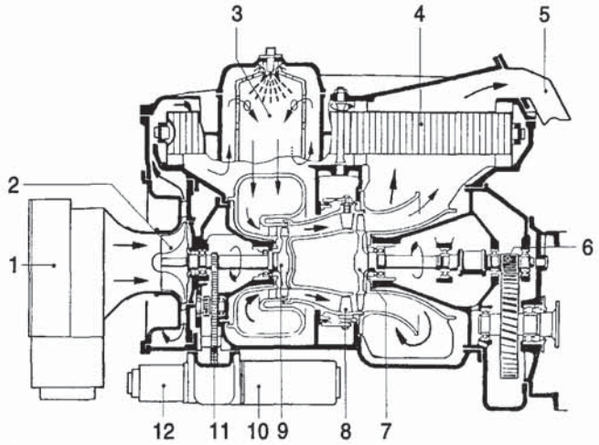
$$\eta_{th} = 1 - (T_2/T_3).$$

Current gas-turbine powerplants achieve thermal efficiencies of up to 35%.

Advantages of the gas turbine: clean exhaust without supplementary emissions-control devices; extremely smooth running; multifuel capability; good static torque curve; extended maintenance intervals.

Disadvantages: manufacturing costs still high, poor transitional response, higher fuel consumption, less suitable for low-power applications.

Gas turbine
1 Filter and silencer, 2 Radial-flow compressor, 3 Burner, 4 Heat exchanger, 5 Exhaust port, 6 Reduction gearset, 7 Power turbine, 8 Adjustable guide vanes , 9 Compressor-turbine, 10 Starter, 11 Auxiliary equipment drive, 12 Lubricating-oil pump.



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Engine cooling

Air cooling

Cooling air is routed by dynamic pressure and/or a fan around the finned external walls of the cylinder casing. Flow restrictors and fan-speed control, etc., can be employed to regulate the current in response to variations in temperature and load factor. Power consumption is 3...4 % of total engine output. Suitable soundproofing measures can be employed to obtain both consistent engine temperatures and noise levels comparable to those of liquid-cooled engines. Heat absorbed by the engine's oil is dispersed by an air-cooled oil cooler mounted at a suitable position in the air stream.

Water cooling

Water cooling has become the standard in both passenger cars and heavy-duty vehicles.

Pure water is no longer employed as coolant; today's coolants are a mixture of water (drinking quality), antifreeze (generally ethylene glycol), and various corrosion inhibitors selected for the specific application. An antifreeze concentration of 30...50 % raises the coolant mixture's boiling point to allow operating temperatures of up to 120 °C at a pressure of 1.4 bar in passenger cars.

Radiator designs and materials

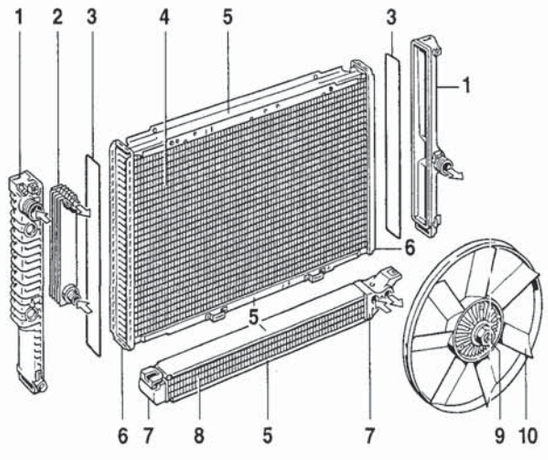
The cores of the coolant radiators in modern passenger cars are almost always made of aluminum, which is also being used in an increasing number of heavy-vehicle radiators throughout the world. There are two basic assembly variations: brazed and mechanically joined radiators.

For cooling high-output engines, or when space is limited, the best solution is a brazed, high-performance flat-tube and corrugated-fin radiator layout with minimal aerodynamic resistance on the air-intake side.

The less expensive, mechanically-assembled finned-tube system is generally employed for applications with less powerful engines or when more space is available.

When the radiator is assembled mechanically, the cooling grid is formed by mounting stamped fins around round, oval and flat-oval tubes. The fins are corrugated and/or slotted at right angles to the direction of air flow.

Turbulators are applied in all types of radiators to enhance the thermal transfer on the coolant side (i. e. in the pipes) provided the attendant pressure losses remain within acceptable limits. On the cooling-air side, corrugations and gills provide



Passenger-car cooling system
1 Radiator tank,
2 Transmission-fluid cooler,
3 Gasket,
4 Engine-coolant radiator,
5 Side brace,
6 Base,
7 Oil tank,
8 Engine-oil cooler,
9 VISCO® coupling,
10 Fan.

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improved thermal transfer in the fins.

The radiator tank ensures that the coolant is distributed throughout the block. These tanks are made of fiberglass-reinforced polyamides, and are injection molded with all connections and mountings in a single unit. They are flange-mounted to the radiator core.

Radiator design

Regardless of operating and environmental conditions, the radiator must continue to provide reliable thermal transfer by discharging engine heat into the surrounding air. Different methods can be applied to determine radiator capacity. The cooling capacity required for a specific radiator can be determined empirically, using comparisons with reference units of the same design, or calculations based on correlation equations for thermal transfer and flow-pressure loss can be employed.

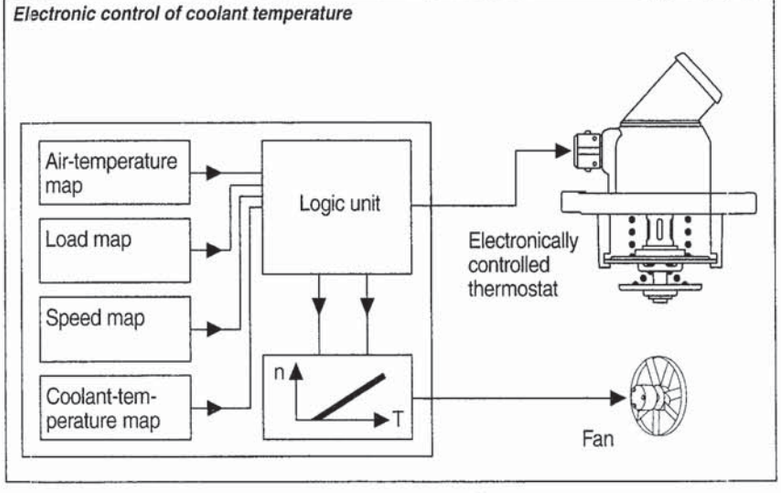
Aside from reliable cooling, other priorities in radiator design include minimizing the power required to operate the fan and maintaining low aerodynamic drag. The mass of the cooling air stream is a decisive factor, as there is an inverse relationship between fan and radiator capacities: A more powerful fan with higher energy consumption allows a smaller radiator, and vice versa. In addition, the temperature differential between the surrounding air and the coolant should be as large as possible, an objective that can only be achieved by maximizing coolant temperature, which in turn entails a corresponding increase in system pressure.

Regulation of coolant temperature

A motor vehicle's engine operates in a very wide range of climatic conditions and with major fluctuations in engine load factors. The temperature of the coolant -- and with it that of the engine – must be regulated if they are to remain constant within a narrow range. An efficient way to compensate for varying conditions is to install a temperature-sensitive thermostat incorporating an expansion element to regulate temperature independent of pressure variations in the cooling system. The thermostat responds to drops in coolant temperature by activating a valve to increase the amount of coolant bypassing the radiator. This method provides consistent operating temperatures, good vehicle-heater performance and helps lower emissions while also reducing engine wear.

Further possibilities are permitted when a map thermostat is used. An electronically controlled thermostat differs from purely expansion-element-regulated thermostats in that it has a larger logic content. When the wax element is heated in the map thermostat, which can be controlled by the engine-management sys-

Electronic control of coolant temperature



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tem, an increased coolant temperature is simulated such that an optimum temperature level is set.

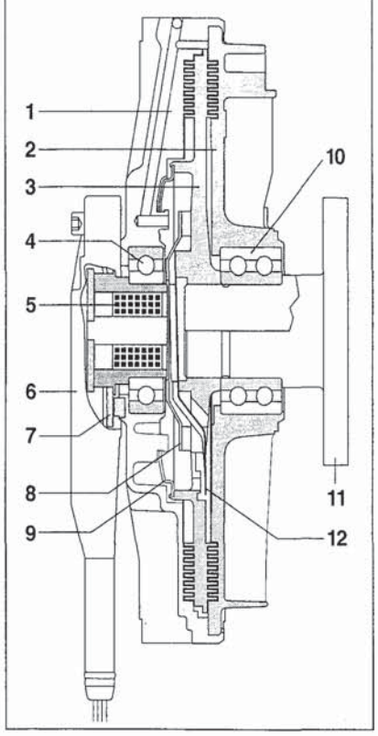
Coolant expansion tank

The expansion tank provides a reliable escape channel for pressurized gases, preventing cavitation of the kind that tends to occur on the suction side of the water pump. The expansion tank's air volume must be large enough to absorb the coolant's thermal expansion during rapid pressure buildup and prevent the coolant from boiling over.

Expansion tanks are injected-molded in plastic (generally polypropylene), although simple designs can also be inflated to shape. The expansion tank can form a single unit with the radiator tank, or the two can be joined in a flange or plug connection. It is also possible to install the expansion tank at a remote location.

The position and shape of the filler opening can be used to limit capacity, thus preventing overfilling. A sight glass or an electronic level sensor can be employed to monitor the level of the coolant, or the expansion tank can be manufactured in undyed, transparent plastic. However, colorless polypropylene is sensitive to ultraviolet rays; it is thus important that the expansion tank not be exposed to direct sunlight.

Visco coupling TE20-K2
1 Basic body, 2 Cover, 3 Primary disk, 4 Magnet bearing, 5 Solenoid coil, 6 Solenoid, 7 Permanent magnet, 8 Stop plate, 9 Intermediate disk, 10 Coupling bearing, 11 Flanged shaft, 12 Valve-lever spring.



Cooling-air fan

Because motor vehicles also require substantial cooling capacity at low speeds, force-air ventilation is required for the radiator. Single-piece injection-molded plastic fans are generally employed in passenger cars; injection-molded fans with drive-power ratings extending up to 20 kW are likewise now used in commercial vehicles. Fans with more modest power ratings are mostly driven electrically by DC and EC motors (up to 600 W). Although blade design and arrangement can be selected to provide relatively quiet operation, the noise levels of such fans remain substantial due to their consistently high rotation speeds. Using electric motors for cooling in mid-size cars and larger vehicles would entail excessive costs. On these vehicles, the fan is powered directly by the engine, via a drive belt, or, in heavy trucks, the fan is attached directly to the crankshaft, dispensing with an intermediate drive element.

The fan-control arrangement requires particular attention. Depending upon vehicle and operating conditions, the unassisted air stream can provide sufficient cooling up to 95% of the time. It is thus possible to economize on the fuel which would otherwise have to provide the energy to drive the fan. Electric fans use a multistage or continuous control system for this purpose, i.e. the fan is only activated above precisely defined coolant temperatures by means of electric temperature switches or by the engine electronics.

The fluid-friction or viscous-drive fan

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(VISCO® coupling) is a mechanical-drive arrangement of proven effectiveness for application in both passenger cars and heavy vehicles. It basically consists of three sections: the engine-powered primary (or input) disk, the internally-activated secondary (or output) section, and the control mechanism.

Control can be effected by two methods:

- Firstly, the pure temperature-dependent, self-regulating coupling, which varies its speed infinitely through a bimetallic element, an operating pin and a valve lever by means of the amount of silicone oil located in the working chamber. The controlled variable is the temperature of the air leaving the radiator, and thus indirectly the temperature of the coolant.
- Secondly, the electrically activated coupling; this coupling is electronically controlled and electromagnetically actuated. Instead of just one controlled variable, a wide range of input variables is used for control purposes. These are usually the temperature limits of the various cooling media.

An intermediate disk divides the secondary section into a supply chamber and a working chamber through which the fluid circulates. There is no mechanical connection between the working chamber and the primary disk, which rotates freely within it. Torque is transmitted through the internal friction of the highly-viscous fluid and its adhesion to the inner surfaces. There is a degree of slippage between input and output.

Intercooling (charge-air cooling)

Cooling the boost air reduces both the thermal loads placed on the engine and the exhaust-gas temperatures, with attendant benefits in NO_X emissions and fuel consumption. It also inhibits preignition in spark-ignition engines. Basically speaking, both the engine coolant and the ambient air can be employed to cool the boost air. An air-to-coolant intercooler can be installed in virtually any location, a benefit associated with this water-cooled unit's modest dimensions. However, without an auxiliary cooling circuit, this type of system can only cool the charge air down almost to the temperature of the engine coolant.

For these reasons air-to-air intercoolers have become the configuration of choice in both passenger cars and heavy vehicles. These intercoolers can be mounted in front of, beside or above the engine radiator, or at a completely separate location. A separately-mounted intercooler can utilize either the unassisted vehicle air stream or its own fan. Extra effort is required to ensure adequate air supply when the intercooler is to be located to the front of the engine radiator. The advantage of this location lies in the fact that the fan ensures sufficient air flow across the intercooler at low vehicle speeds. A disadvantage is that the cooling air is itself heated in the process: The capacity of the engine radiator must therefore be increased accordingly.

The system of corrugated aluminum fins and tubes employed for the intercooler core is similar to that used in the radiator for the engine coolant. Wide tubes with internal fins provide superior performance and structural integrity in actual practice. Owing to the high level of thermal-transfer resistance on the charge-air side it is possible to hold the fin density on the cooling-air side to a minimum. The diffusion rate Φ is a particularly important intercooler property. It defines the relationship between boost-air cooling efficiency and the boost-air/cooling-air temperature differential:

$$\Phi = (t_{1E} - t_{1A}) / (t_{1E} - t_{2E})$$

The equation's elements are:
Φ Diffusion rate
t_{1E} Boost-air intake temperature,
t_{1A} Boost-air exit temperature,
t_{2E} Cooling-air intake temperature.
For passenger cars: $\Phi = 0.4...0.7$;
For commercial vehicles: $\Phi = 0.65...0.85$.
Whenever possible, the plenum chamber is injection-molded in fiberglass-reinforced polyamide as a single casting incorporating all connections and mounts. Plenum chambers which are subject to increased stresses, e.g. the charge-air inlet

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system, are injection-molded from highly heat-resistant PPA or PPS. The plenum chamber is flange-mounted on the radiator core. Plenum chambers which feature undercut shapes or are intended for high-temperature applications are die-cast in aluminum, and welded to the core.

Oil and fuel cooling

Oil coolers are often needed in motor vehicles to cool both engine oil and gear oil. They are used when the heat losses from the engine or transmission can no longer be dissipated via the surface of the oil pan or the transmission with the result that the permitted oil temperatures are exceeded.

Fuel coolers are installed in modern diesel assemblies in order to cool down to a permissible level the excess diesel fuel from the return which heats up during the injection process.

Oil coolers generally take the form of aluminum oil-to-air or oil-to-coolant coolers, which can be installed either adjacent to the engine-coolant radiator in the cooling module or separately. Separately mounted units depend upon the unassisted air stream or an extra fan for cooling.

Oil-to-air coolers mostly consist of a system of flat tubes and corrugated fins with a high power density, or of a system of round tubes and flat fins. Turbulence inserts are soldered into flat-tube systems for strength reasons (high internal pressures).

Stainless-steel disk coolers and aluminum forked-pipe coolers are used to cool lubricating oil and engine coolant in passenger cars. However, aluminum stack designs have entered the market and proven successful over the last few years.

Disk coolers have their own casing and are mounted between the engine block and the oil filter. Forked-pipe coolers have no casing and must therefore be integrated in the oil-filter housing or in the oil pan.

Disk-stack oil coolers are made up of individual disks which are turned and arranged on top of each other. Turbulence inserts are inserted between the individual disks.

If only a modest cooling output is required (e.g. cooling of transmission fluid in automatic transmissions), aluminum flat-tube coolers can be used for passenger cars and commercial vehicles. They are installed in the outlet tank of the engine radiator.

An oil-to-air cooler is used to cool transmission fluid in more powerful heavy vehicles. The unit is mounted in front of the engine radiator in order to ensure good ventilation.

Lube oil in heavy vehicles is generally cooled by stainless-steel disk packs or aluminum disk-stack coolers which are accommodated in an extended coolant duct in the engine block. If conditions are favorable, neither a casing nor additional lines are required.

Cooling-module technology

Cooling modules are structural units which consist of various cooling and air-conditioning components for a passenger vehicle, and include a fan unit complete with drive, e.g. a hydrostatic or electric motor or a Visco® coupling.

In principle, module technology features a whole range of technical and economic advantages:

- optimum layout and alignment of the components to the fan's power output,
- thus improved efficiency in the passenger car, or smaller and more inexpensive components can be realized.

The optimum layout of the individual components and their coordination with each other are made possible by exact knowledge of the characteristic curves for the fan, the fan-drive and the heat-exchanger. This prevents e.g. the backflow of cooling air, which would reduce the effective cross-sectional area of the cooler.

Intelligent thermomanagement

Future developments are heading in the direction of operationally optimized regulation of different heat flows.

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Thermomanagement involves operationally optimized cooling by means of the demand-compatible regulation and allocation of material and heat flows. Thus the cooling-air flow must be regulated by the fan and radiator-blind actuators in such a way that only the minimum throughput that is actually required for heat dissipation passes through the cooling-air system (demand-triggered regulation). This minimum throughput must then be made available by suitable routing elements to the heat exchangers which require cooling air. This philosophy is to be applied to all material flows such as coolant, cooling air, engine and gear oil, charge air and fuel. For this concept to be successfully implemented, it is essential that new actuators be used in the cooling-air system and fluid circuits such as e.g. controllable fans, pumps, valves, flaps, throttling and routing elements, and that they be incorporated in a microprocessor-controlled control system.

Exhaust-gas cooling

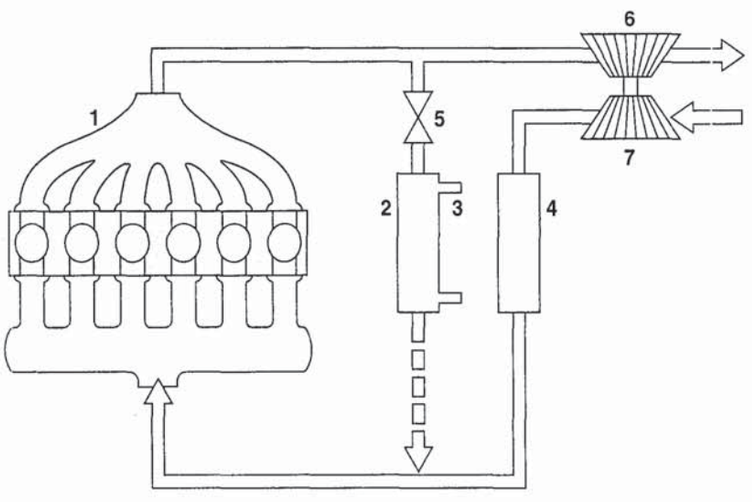
Because of the introduction of new, stricter exhaust-emission regulations for diesel engines, new technologies for reducing emissions have become the focus of engineers' attention.

One such technology is cooled exhaust-gas recirculation (EGR), which allows the reduction of emissions at the cost of only a minimum rise in fuel consumption.

The EGR system is accommodated in the high-pressure area of the engine. The exhaust gas to be recirculated is removed from the main flow between the cylinder and turbine, cooled by the engine coolant and then reintroduced to the fresh air after the intercooler. The EGR system consists of a valve which regulates the amount of exhaust gas to be recirculated, the exhaust lines, and the exhaust-gas-to-coolant heat exchanger.

Because of its location in the high-pressure area, the exhaust-gas heat exchanger is subject to extreme operating conditions. For instance, the exhaust-gas temperature can reach up to 400 °C for passenger cars and up to 700 °C for commercial vehicles, a fact which makes it imperative to use heat-resistant materials.

Cooled exhaust-gas recirculation *(schematic)*
1 Engine, 2 Exhaust-gas heat exchanger, 3 Water connection, 4 Intercooler, 5 EGR valve, 6 Turbine, 7 Compressor.



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Fuel filters

Injection systems for spark-ignition and diesel engines are sensitive to the smallest impurities in the fuel. Damage can be caused above all by particulate erosion, abrasion and water corrosion. The service life of the fuel-injection system is only guaranteed by a specific minimum purity of the fuel supplied to the components exposed to wear.

The function of the fuel filter is to reduce particulate impurities. The necessary filter fineness is defined by the requirements of the fuel-injection system. As well as guaranteeing protection against wear, the fuel filter must feature an adequate particulate retention capacity. Filters with insufficient capacity to retain contaminants are liable to become clogged before the end of the official replacement interval. This will result in a reduction in fuel delivery and thus a drop in engine power. It is therefore essential to install a fuel filter that is custom-tailored to the relevant injection system. Using unsuitable filters has at best unpleasant and at worst very costly consequences (replacement of components up to and including the entire injection system).

Fuel filters for gasoline injection systems

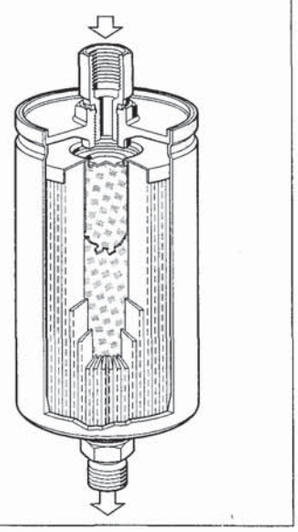
Fuel filters for spark-ignition engines are located between the fuel tank and the fuel pump and/or on the pressure side downstream of the fuel pump. The preferred design is the in-line filter. In addition, easy-change filters screwed to a base, and housing filters with non-metal filter elements are used. Components such as the pressure control valve can be integrated in the filter head.

The filter element consists of radial or spiral vee-shaped filter media. Modern filter media for gasoline consist of mixtures of superfine pulp and polyester fibers. The filter media are always deep-bed filters where particulates are predominantly retained on the inside of the medium. The filtration efficiency in once-through operation for a particulate fraction of between 3 and 5 µm (ISO/TR 13353: 1994, Part 1) is 20...50% depending on the injection system. The structure of the mixed-fiber medium and the filter area determine the particulate retention capacity and thus the maintenance interval. The design of new fuel filters for spark-ignition engines is increasingly tending towards maintenance-free service-life concepts (e.g. filter in the in-tank fuel-pump unit).

Fuel filters for diesel injection systems

Because of the significantly higher injection pressures, considerably higher demands are placed on the wear-protection performance of modern diesel injection systems compared with gasoline injection systems. This ensures reliability, low fuel consumption and compliance with emission limits over the entire service life of the vehicle (up to 1,000,000 km for commercial vehicles). Depending on the fuel contamination and application, this requires a filtration efficiency in once-through operation for a particulate fraction of between 3 and 5 µm of 70...>95% (ISO/TR 13353: 1994, Part 1). Less se-

In-line gasoline filter
1 Radial vee-shaped filter element.



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vere filtration stipulations apply to in-line injection pumps since they are connected to the engine lube-oil circuit.

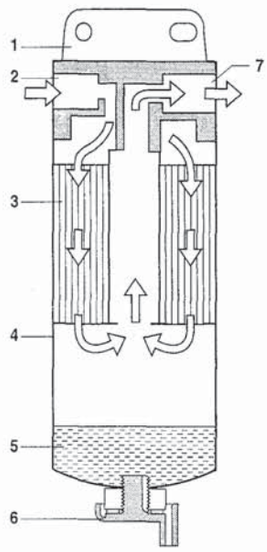
As well as high levels of superfine particulate separation, the fuel filter is expected to feature a particulate-retention capability which will permit compliance with the maintenance intervals as stipulated by the vehicle manufacturer. This can only be achieved by using special filter mediums, e.g. in multilayer form with synthetic microfiber layers. These filter mediums apply a fine prefilter effect, and guarantee maximum particulate-retention capability by separating the particulates inside the particular filter layer.

A second essential function of the diesel filter is to separate emulsified and free water in order to prevent corrosion damage. An effective water separation of >93 % at rated flow (ISO 4020) is absolutely essential for distributor injection pumps and common-rail systems. Water separation takes the form of coalescence on the filter medium. The separated water collects in the water chamber in the bottom of the filter housing. Conductivity sensors are used to monitor the water level, and the water is drained off manually via a drain plug.

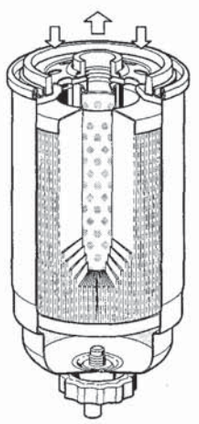
Diesel fuel is more heavily contaminated than gasoline (particularly due to products which result from the aging process). Diesel filters are therefore designed as easy-change filters. Screw-on easy-change filters with radial or spiral vee-shaped filter elements are widely used. Fuel filters with aluminum/plastic and fully plastic housings are being increasingly used. Only a non-metal filter element remains as a replacement part.

If particularly exacting demands are placed on wear protection and/or maintenance intervals, there are filter systems which have a prefilter that is adapted to the fine filter. In addition to the "microfiltration" and "water separation" functions, modern diesel filters combine "fuel preheating" (using electrical, coolant, or fuel-recirculation methods) to prevent clogging due to paraffin during winter operation, maintenance indication by way of a differential-pressure measurement, and manual pumping for filling/venting after the filter is replaced.

Diesel filter with water drain
1 Filter cover, 2 Fuel inlet, 3 Paper filter element, 4 Housing, 5 Water collecting chamber, 6 Water drain plug, 7 Fuel outlet.



Easy-change diesel filter



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Air supply

Air filters

The air filter serves to inhibit internal wear by preventing air-borne dust from being drawn into the engine.

On paved roads, the dust content of the air averages 1 mg/m³, however, on unpaved roads and on construction sites the dust content can be high as 40 mg/m³. This means that – depending on roads and operating conditions – a medium-sized engine can draw in up to 50 g of dust over 1,000 km.

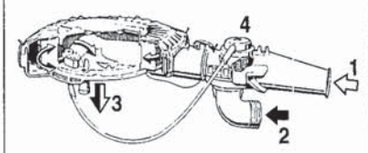
Passenger-car air filters

In addition to filtering the air, passenger-car air filters preheat the intake air and regulate its temperature, as well as damping the air-intake noise. Intake-air temperature regulation is important for the operation of the vehicle and for the composition of the exhaust gases. The temperature of the intake air may differ under part-load and full-load operating conditions.

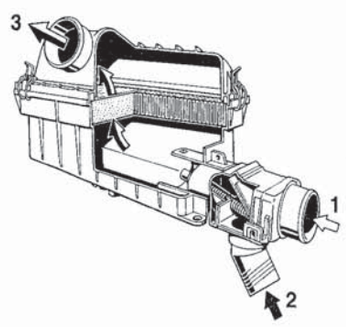
The required amount of hot air is drawn in in the vicinity of the exhaust and added to the cold intake air at the filter inlet by means of a flap-valve mechanism. The regulating mechanism is usually an automatic arrangement employing either a pneumatic vacuum unit connected to the intake manifold or an expansion element. The constant regulated intake-air temperature improves engine performance and fuel consumption, and decreases the percentage of pollutants in the exhaust gases as a result of better fuel management and distribution of the air-fuel mixture. Preheating the intake air also shortens the warm-up phase of the engine after it is started, particularly in cold weather. It also prevents ice from forming in the carburetor.

Passenger-car air filters employ paper cartridges and can be mounted either centrally or at the side of the engine compartment. This type of filter is characterized by a high retention factor which, to a large extent, remains insensitive to fluctuations in load. Cartridge replacement is a simple operation performed at the intervals specified by the vehicle's manufacturer. Passenger-car air filters must be

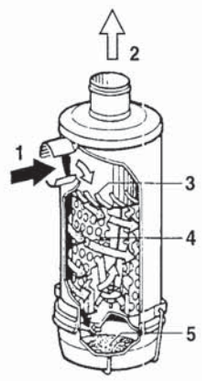
Central passenger-car air filter
1 Fresh-air intake, 2 Warm-air intake,
3 Outlet for warm/fresh air mixture,
4 Vacuum unit.



Side-mounted passenger-car air filter
1 Fresh-air intake, 2 Warm-air intake,
3 Outlet for warm/fresh air mixture.



Paper air filter with cyclone for commercial vehicles
1 Air inlet,
2 Air outlet,
3 Cyclone vanes,
4 Filter element,
5 Dust bowl.



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specially matched to each engine type in order to optimize power output, fuel consumption, intake-air temperature and damping.

Commercial-vehicle air filters

Most of the air filters used in heavy vehicles are of the paper-element type, although oil-bath filters are employed in some applications. Paper filters exhibit high filtering efficiency in all load ranges and increased flow resistance as the amount of dirt retained by the filter increases. To save space, a cyclone prefilter is often integrated in the air-filter housing fitted with a paper element. The filter is serviced by either replacing the filter element or emptying the dust cup.

Paper air filters often incorporate maintenance indicators to show when the filter needs servicing.

The servicing information provided by the manufacturer of the vehicle or the equipment must be followed. Servicing can be simplified, according to the intensity of engine-air pulsation, by using specially matched automatic dust-unloading valves.

Cyclones increase filter service life and extend the maintenance intervals. The cyclone vanes rotate the air, causing the majority of dust particles to be separated from the air before reaching the downstream air filter. Cyclones can be installed upstream of both paper air filters and oil-bath air filters. They cannot be used alone as engine air filters because their filtering efficiency is inadequate.

Intake-noise damping

The intake noise of passenger-car and commercial-vehicle air filters must be damped in order to comply with legal regulations pertaining to the overall vehicle noise level. Noise damping is achieved almost exclusively by designing the air filter to act as a reflection sound absorber having the specialized shape of a Helmholtz resonator; see P. 451.

Assuming that the air filter is of sufficient size, intake noise can generally be damped by 10 to 20 dB (A). A good empirical value for 4-stroke engines is 15 to 20 times the displacement of one cylinder. In special cases in which the noise at particular frequencies is excessive, supplemental dampers must be used.

The resonance frequency of an intake damper is

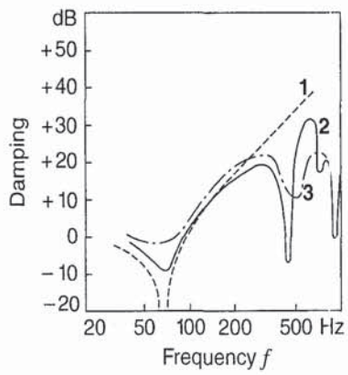
$$f_0 = \frac{c}{2\pi} \cdot \sqrt{\frac{A_m}{l \cdot V}}$$

Where

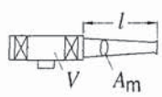
- c Speed of sound in air
- l Length of the intake manifold
- A_m Mean cross section of the intake manifold
- V Filter volume

Intake-noise damping

Damping curves of an intake-noise damper
Damper resonance f_0 = 66 Hz.
1 Theoretical damping curve without taking into account pipe resonances.
2 Curve of measured damping response with low sound energy density and without parallel flow (loudspeaker measurement).
3 Measured damping response with high sound energy density and with parallel flow (measurement at the engine).



Air filter with intake pipe
l Length of intake pipe,
A_m Mean cross-section of intake pipe,
V Filter volume.



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Turbochargers and superchargers for internal-combustion engines

(self-charging)

By compressing the air inducted for combustion in the internal-combustion (IC) engine, and thereby increasing its mass, charging systems also increase the output obtained for a given displacement at a given engine speed. The "compressors" generally used for IC engines are of three basic types; the mechanically-driven supercharger, the exhaust-gas turbocharger and the pressure-wave supercharger.

Mechanical superchargers compress the air using power supplied by the engine crankshaft (mechanical coupling between engine and supercharger), while the turbocharger is powered by the engine's exhaust gases (fluid coupling between engine and turbocharger).

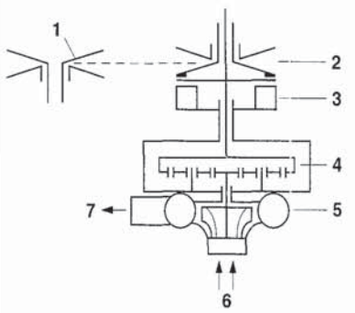
Although the pressure-wave supercharger also derives its compression force from the exhaust gases, it requires a supplementary mechanical drive (combination of mechanical and fluid coupling).

Superchargers (mechanically driven)

These fall into two categories: mechanically-driven centrifugal superchargers (MKL) and mechanically-driven positive-displacement superchargers (MVL).

The turbo-type supercharger for the MKL corresponds to the exhaust-gas turbocharger in its essential configuration. This type of device is very efficient, providing the best ratio between unit dimensions and flow volume. However, the extreme peripheral velocities required to generate the pressure mean that drive speeds must be very high. As the secondary drive pulley (2:1 conversion ratio relative to primary drive) does not rotate fast enough to drive a centrifugal supercharger, a single-stage planetary gear with a 15:1 speed-increasing ratio is employed to achieve the required peripheral speeds. In addition, a transmission unit must be included to vary the rotational speeds if the pressure is to be maintained at a reasonably constant level over a wide range of flow volumes (~ engine speed).

Mechanical centrifugal supercharger (MKL) *(schematic)*
1 Variable-speed primary pulley, 2 Variable-speed secondary pulley, 3 Solenoid clutch, 4 Step-up planetary-gear set, 5 Compressor, 6 Air intake, 7 Air outlet.



The necessity of using extreme rotational speeds, and the technical limits imposed on the transmission of drive power, mean that the centrifugal supercharger's range of potential applications is limited to medium- and large-displacement diesel and gasoline passenger-car engines. This design has not been extensively employed for mechanical superchargers.

Positive-displacement superchargers (MVL) operate both with and without internal compression. Internal-compression superchargers include the reciprocating-piston, the screw-type, the rotary-piston and the sliding-vane compressor. The Roots supercharger is an example of a unit without internal compression. All of these positive-displacement superchargers share certain characteristics as shown in the graphic illustration for a Roots supercharger.

- The curves for the constant rotational speed n_{LAD} = in the graph of p_2/p_1 against $\dot{V}$ are extremely steep, indicating that increases in the pressure ratio p_2/p_1 are accompanied by only slight reductions in the mass-flow volume $\dot{V}$. The precise extent of the drop in flow volume is basically determined by the efficiency of the gap seal (backflow losses). It is a function of the pressure ratio p_2/p_1 and of time, and is not influenced by rotational speed.

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- The pressure ratio p_2/p_1 does not depend upon the rotational speed. In other words, high pressure ratios can also be generated at low mass-flow volumes.
- The mass-flow volume $\dot{V}$ remains independent of the pressure ratio, and is, roughly formulated, directly proportional to rotational speed.
- The unit retains stability throughout its operating range. The positive-displacement compressor operates at all points of the p_2/p_1-$\dot{V}$ graph as determined by supercharger dimensions.

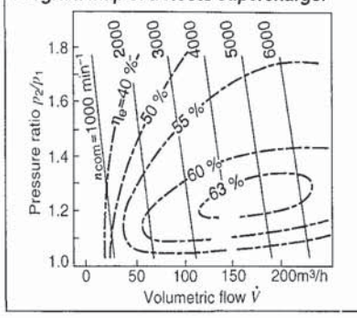
The two twin-bladed rotary pistons of the Roots supercharger operate without directly contacting each other or the housing. The size of the sealing gap thus created is determined by the design, the choice of materials and the manufacturing tolerances. An external gear set synchronizes the motion of the two rotary pistons.

In the sliding-vane supercharger, an eccentrically-mounted rotor drives the three centrally mounted sliding vanes; the eccentric motion provides the internal compression. The extent of this internal compression can be varied for any given eccentricity by altering the position of the outlet edge A in the housing.

The spiral-type supercharger employs an eccentrically-mounted displacement element which is designed to respond to rotation of the input shaft by turning in a double-eccentric oscillating pattern. In sequence, the working chambers open for charging, close for transport and open once again for discharge at the hub. The spirals can be extended beyond the length shown in the illustration to provide internal compression.

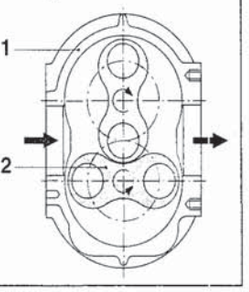
The displacement element is driven by a belt-driven, grease-lubricated auxiliary shaft, while the input shaft is lubricated by the engine's oil circuit. Radial sealing is via gaps, while lateral sealing strips provide the axial seal.

Program map of a Roots supercharger



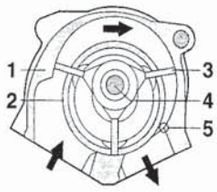
Cross-section through a Roots supercharger

1 Housing, 2 Rotary piston.



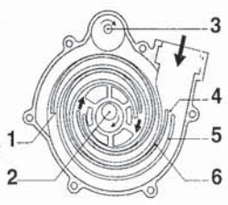
Cross-section through a sliding-vane supercharger

1 Housing, 2 Rotor, 3 Vanes, 4 Shaft, 5 Outlet edge A.



Cross-section through a spiral-type supercharger

1 Air intake into second working chamber, 2 Drive shaft, 3 Displacer guide, 4 Air intake into primary working chamber, 5 Housing, 6 Displacer.



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The rotary-piston supercharger incorporates a rotary piston moving about an internal axis. The driven inner rotor (rotary piston) turns through an eccentric pattern in the cylindrical outer rotor. The rotor ratios for rotary-piston superchargers are either 2:3 or 3:4. The rotors turn around fixed axes without contacting each other or the housing. The eccentric motion makes it possible for the unit to ingest the maximum possible volume (chamber I) for compression and discharge (chamber III). The internal compression is determined by the position of the outlet edge A.

A ring and pinion gear with sealed grease lubrication synchronizes the motion of the inner and outer rotors. Permanent lubrication is also employed for the roller bearings. Inner and outer rotors use gap seals, and usually have some form of coating. Piston rings provide the seal between working chamber and gear case.

Superchargers on IC engines are usually belt-driven (toothed or V-belt). The coupling is either direct (continuous engagement) or via clutch (e.g., solenoid-operated clutch, actuation as required). The step-up ratio may be constant, or it may vary according to engine speed.

Mechanical positive-displacement superchargers (MVL) must be substantially larger than their centrifugal counterparts (MKL) in order to produce a given mass flow. The mechanical positive-displacement supercharger is generally applied to small- and medium-displacement engines, where the ratio between charge volume and space requirements is still acceptable.

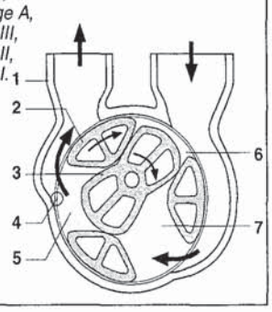
Exhaust-gas turbochargers

The exhaust-gas turbocharger (ATL) consists of two turbo elements: a turbine and a compressor installed on a single shaft. The turbine uses the energy of the exhaust-gas to drive the compressor. The compressor, in turn, draws in fresh air which it supplies to the cylinders in compressed form. The air and the mass flow of the exhaust gases represent the only coupling between the engine and the compressor. Turbocharger speed does not depend upon engine speed, but is rather a function of the balance of drive energy between the turbine and the compressor.

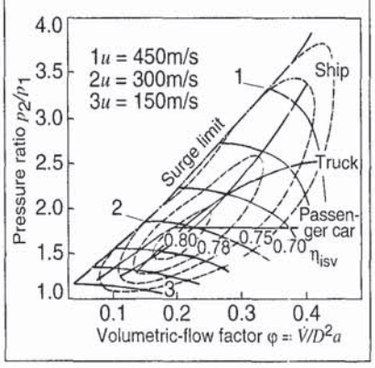
Exhaust-gas turbochargers are used on engines in passenger cars, trucks and heavy-duty engines (marine and locomotive power plants, stationary power generators).

The typical engine-performance curves for this type of application are illustrated in a compression graph (P. 441), valid for all displacements, in which the surge line separates the stable operating range on its right from the instable range. It is obvious that the instable range presents no difficulties provided that the correct turbocharger is selected, as all of the points representing potential operating conditions lie either on the engine operating curves (full load) or below them (part-load operation).

Cross-section through a rotary-piston supercharger
1 Housing,
2 Outer rotor,
3 Inner rotor,
4 Outlet edge A,
5 Chamber III,
6 Chamber II,
7 Chamber I.



Compression graph with typical engine operation curves valid for all displacements



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Different applications require various configurations. However, all exhaust-gas turbochargers have practically the same major components: the turbocharger rotor and shaft assembly, which combine with the bearing housing to form the so-called core assembly, and the compressor housing. Other components such as turbine housing and control elements vary according to the specific application.

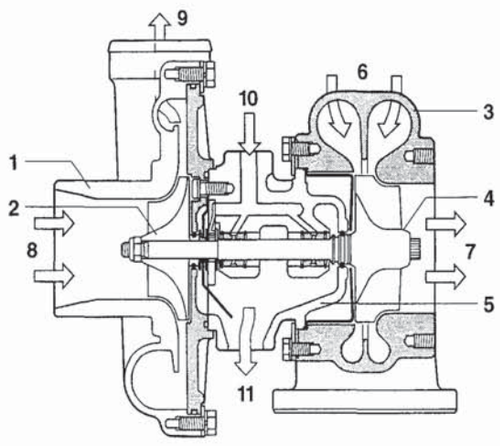
Piston rings are installed on both the exhaust and intake sides to seal off the bearing housing's oil chamber. In some special applications sealing is enhanced by trapped air or a compressor-side carbon axial face seal. Friction bearings are generally used, installed radially as either floating double plain bushings or stationary plain-bearing bushings, while multiple-wedge surface bushings provide axial support. The turbocharger is connected to the engine's lube-oil circuit for lubrication, with oil supply and return lines located between the compressor and turbine housings. No additional cooling arrangements are provided for the bearing housing on standard units. The temperatures can be maintained below critical levels using devices such as a heat shield, and by thermally isolating the bearing housing from the hot turbine housing, supplemented by incorporating suitable design elements in the bearing housing itself. Water-cooled bearing housings are employed for exhaust-gas temperatures in excess of 850 °C. The rear wall of the compressor seals the compressor side of the bearing housing.

The housing of the radial compressor is generally made of cast aluminum. A bypass valve can be integrated in the housing for special applications. This bypass valve is used exclusively in SI-engine supercharging and prevents pumping on the compressor side when load is swiftly removed from the engine.

Turbine housings differ substantially according to intended use. Casting materials for turbine housings range from GGG 40 to NiResist D5 (depending upon exhaust-gas temperature). Exhaust-gas turbochargers for trucks incorporate a twin-flow turbine housing in which the two streams join just before reaching the impeller. This housing configuration is employed to achieve pulse turbocharging, in which the pressure of the exhaust-gas is supplemented by its kinetic energy.

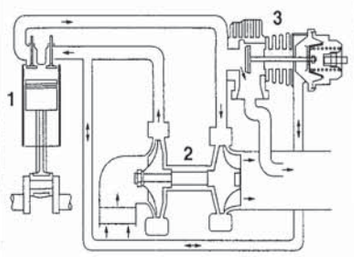
In contrast, in the case of constant-pressure turbocharging, only the pressure energy of the exhaust-gas is utilized, and single-flow turbine housings can be employed. This configuration has become especially popular for use in conjunction with water-cooled turbine housings on marine engines. The exhaust-gas tur-

Truck exhaust-gas turbocharger with twin-flow turbine housing
1 Compressor housing,
2 Compressor wheel,
3 Turbine housing,
4 Rotor,
5 Bearing housing,
6 Incoming exhaust gas,
7 Exhaust-gas discharge,
8 Atmospheric fresh air,
9 Compressed fresh air,
10 Oil supply,
11 Oil return.

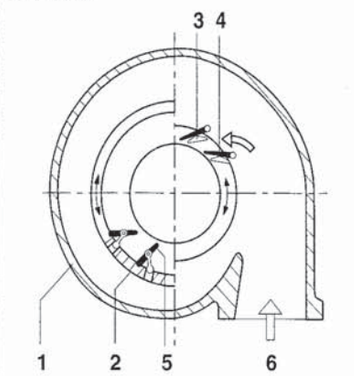


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Boost-pressure regulation via exhaust-side boost-pressure control valve (wastegate)
1 Engine, 2 Exhaust-gas turbocharger, 3 Wastegate.



Variable turbine geometry
(schematic diagram)
1 Turbine housing, 2 Adjusting ring, 3 Control cams, 4 Adjustable guide blades, 5 Guide blades with adjusting lever, 6 Air intake.



bochargers on heavy-duty engines often incorporate a nozzle ring upstream from the turbine. The nozzle ring provides a particularly smooth and consistent flow to the impeller while allowing fine adjustment of the flow through the turbine.

Exhaust-gas turbochargers for passenger cars generally use single-flow turbine housings. However, the car engine's wide min^{-1} range means that some form of turbocharger governing mechanism is required if the boost pressure is to be maintained at a relatively constant level throughout the engine's operating range. Standard practice presently favors regulating flow on the exhaust side, whereby a portion of the engine's exhaust gases is routed past the turbine (bypass) using a governing mechanism (wastegate) which can be in the form of a valve or a flap.

In today's production turbochargers, the wastegate is actuated pneumatically, i.e. with negative pressure or overpressure. Here the necessary control pressure is tapped off directly or in timed mode from the turbocharger or as timed negative pressure from the vehicle electrical system.

With appropriate microelectronic support, boost-pressure control can be effected as a function of the engine program maps. Future wastegates will be electrically or electronically actuated.

The available energy is exploited more efficiently by governing systems incorporating turbines with variable blade geometry. With this system, the turbine's flow resistance is modified continuously to achieve maximum utilization of the exhaust energy under all operating conditions.

Of all the potential designs, adjustable guide blades have achieved general acceptance, as they combine a wide control range with high efficiency levels.

An adjusting ring is rotated to provide simple adjustment of the blade angle. The blades, in turn, are swiveled to the desired angles using adjusting cams, or directly via adjusting levers attached to the individual blades. The pneumatic actuator can operate with either vacuum or positive pressure. Microelectronic control systems can exploit the advantages of variable turbine-blade geometry by providing optimal boost pressure throughout the engine's operating range.

Aside from the variable turbine with adjustable guide blades, the turbine with adjustable control spool VST has gained acceptance in small-capacity car engines (see illustration). The mode of operation of the VST provides that, analogously to the fixed turbine, initially one flow channel determines the ram performance (spool position 1).

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When the maximum permissible boost pressure is reached, the spool opens continuously in an axial direction and exposes the second flow channel (position 2). Both channels together are configured in such a way that by far the largest part of the exhaust-gas mass flow is routed through the turbine. The remaining quantity is routed past the impeller inside the charger by further displacement of the control spool (position 3).

Multistage supercharging

Multistage supercharging is an improvement on single-stage supercharging in that the power limits can be significantly extended. The objective here is to improve the air supply on both a stationary and a non-stationary basis and at the same time to improve the specific consumption of the engine. Two supercharging processes have proven successful in this respect.

Sequential supercharging

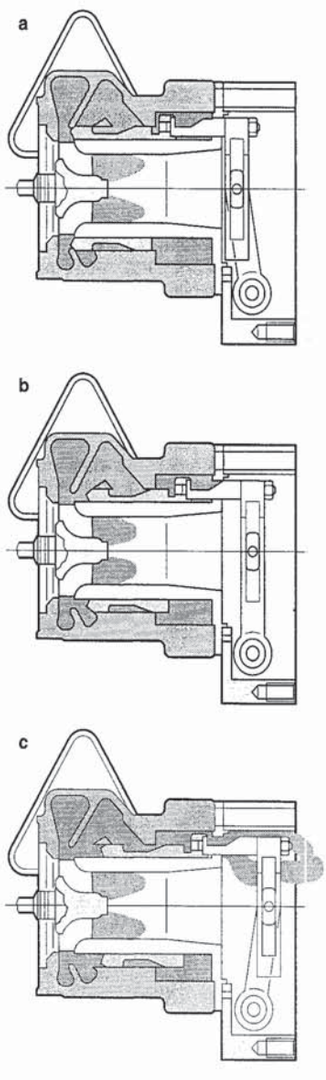
Because of the expensive charger switching equipment, sequential supercharging is predominantly used in ship propulsion systems or generator drives. In this case, as engine load and speed increases, one or more turbochargers are cut in to the basic supercharging process. Thus, in comparison with a large supercharger, which is geared to the rated power, two or more supercharging optima are achieved.

Two-stage controlled supercharging

This supercharging process is used in motor-vehicle applications on account of its simple control response. Two-stage controlled supercharging involves the serial connection of two turbochargers of different sizes with a bypass control system and ideally a second intercooler.

The exhaust-gas mass flow from the cylinders initially passes into the exhaust manifold. From this point, there is the possibility of either expanding the exhaust-gas mass flow through the high-pressure turbine (HP) or diverting a partial mass flow through the bypass line. The entire exhaust-gas mass flow is then used again by the downstream low-pressure turbine (LP).

Operation of VST supercharger with adjustable spool
a) Spool position 1 (only left channel open),
b) Spool position 2 (both channels open),
c) Spool position 3 (both channels and bypass open).



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The entire fresh-air mass flow is initially precompressed by the low-pressure stage and ideally intercooled. The flow is then further compressed and intercooled in the high-pressure stage. As a result of the precompression, the relatively small HP compressor works on a higher pressure level so it is able to generate the required air mass flow.

At low engine speeds, i.e. small exhaust-gas mass flows, the bypass remains fully closed and the entire exhaust-gas mass flow expands through the HP turbine. This results in a very rapid and high build-up of boost pressure. As engine speed increases, the expansion work is continuously switched to the LP turbine whereby the bypass cross-section is enlarged accordingly.

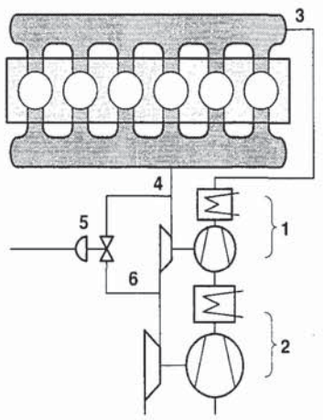
Two-stage controlled supercharging thus enables infinitely variable adaptation on the turbine and compressor sides to the requirements of engine operation.

Pressure-wave superchargers

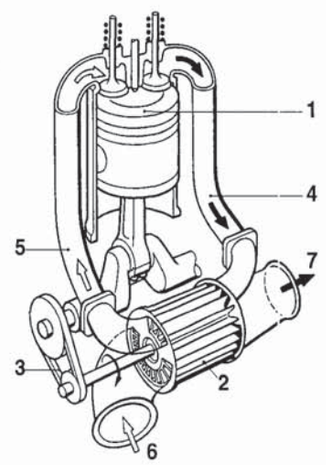
The pressure-wave supercharger exploits the dynamic properties of gases, using pressure waves to convey energy from the exhaust-gas to the intake air. The energy exchange takes place within the cells of the rotor (known as the cell rotor or cell wheel), which also depends upon an engine-driven belt for synchronization and maintenance of the pressure-wave exchange process.

Inside the cell rotor, the actual energy-exchange process proceeds at the speed of sound. This depends upon exhaust-gas temperature, meaning that it is essentially a function of engine torque, and not engine speed. Thus the pressure-wave process is optimally tailored to only a single operating point if a constant step-up ratio is employed between engine and supercharger. To get around this disadvantage, appropriately-designed "pockets" can be incorporated in the forward part of the housings. These achieve high efficiency levels extending through a relatively wide range of engine operating

Schematic structure of two-stage controlled supercharging
1 HP stage (high pressure),
2 LP stage (low pressure),
3 Intake manifold,
4 Exhaust manifold,
5 Bypass valve,
6 Bypass line.



Pressure-wave supercharger
1 Engine, 2 Cell rotor, 3 Belt drive,
4 High-pressure exhaust-gas,
5 Pressurized air, 6 Low-pressure air intake,
7 Low-pressure exhaust outlet.



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conditions and provide a good overall boost curve.

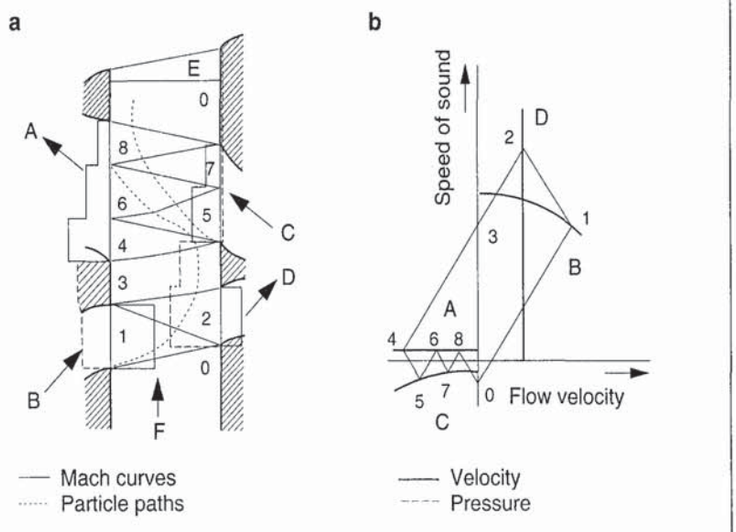
The exchange of energy occurring within the rotor at the speed of sound ensures that the pressure-wave supercharger responds rapidly to changes in engine demand, with the actual reaction times being determined by the charging processes in the air and exhaust tracts.

The pressure-wave supercharger's cell rotor is driven by the engine's crankshaft via a belt assembly. The cell walls are irregularly spaced in order to reduce noise. The cell rotor turns within a cylindrical housing, with the fresh air and exhaust-gas tracts feeding into the housing's respective ends. On one side are low-pressure air intake and pressurized air, while the high-pressure exhaust and low-pressure exhaust-gas outlet are located on the other side.

The accompanying gas-flow and state diagrams illustrate the pressure-wave process in a basic "Comprex" at full load and moderate engine speed. Developing (or unrolling) rotor and housing converts the rotation to a translation. The state diagram contains the boundary curves for the four housing openings in accordance with local conditions. The diagrams for the ideal no-loss process have been drawn-up with the assistance of the intrinsic characteristic process.

The pressure-wave supercharger's rotor is over-mounted and is provided with permanent grease lubrication, with the bearing located on the unit's air side. The air housing is of aluminum, the gas housing of NiResist materials. The rotor with its axial cells is cast using the lost-wax method. An integral governing mechanism regulates boost pressure according to demand.

Gas-flow diagram (a) and state diagram (b) for pressure-wave supercharger
A Exhaust-gas outlet, B Exhaust-gas intake, C Air intake, D Air outlet, E Residual air, fresh air, F Direction of rotor rotation.



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Exhaust systems

Exhaust-system purpose

The exhaust system reduces the pollutant constituents of the exhaust gas generated by combustion in the engine. The remaining exhaust gas is then discharged as quietly as possible at a convenient point on the vehicle. The engine power should be reduced as little as possible during the process.

Exhaust-system design

A passenger-car exhaust system serves here as our example. It consists basically of three main components (although some of these are also found in commercial-vehicle exhaust systems):

The catalytic converter serves as the exhaust-gas cleaning device for spark-ignition (SI) engines, and the oxidation catalytic converter for diesel engines. It is mounted as close as possible to the engine so that it can quickly reach its operating temperature and therefore be effective in urban driving. When installed as retrofit equipment it is fitted in place of the front muffler (front silencer), whose acoustical functions it also assumes in addition to its exhaust-gas cleaning function. The resulting acoustical changes in the vehicle must meet legal requirements.

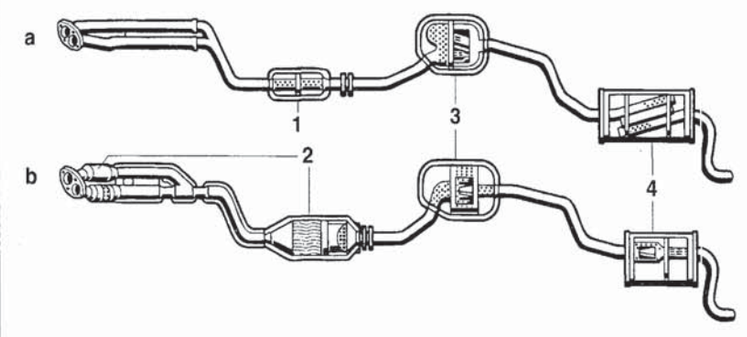
Depending on the size of the vehicle and the engine, one or several mufflers are used. In V-engines the left and right cylinder banks are frequently run separately, each being fitted with its own catalytic converter or muffler, and only brought together at the end of the vehicle in one large muffler.

The exhaust pipes are the third and last component in the exhaust system. They combine the exhaust-gas outlets in the cylinder head into one or more pipes (manifolds), and also connect the catalytic converter(s) and the mufflers to each other. The length and cross-section of the pipes, as well as the type of junction used, influence the vehicle's performance characteristics and acoustic behavior. Exhaust systems for vehicles with larger swept volumes are therefore often fitted with twin pipes. The pipes, the catalytic converter and muffler are connected to the main body of the system by means of insert connections and flanges. Many original-equipment systems are welded into one complete unit for faster mounting.

The entire exhaust system is connected with the underbody of the vehicle via flexible suspension elements. The fixing points must be carefully selected, as otherwise vibration can be transmitted to the bodywork and generate noise in the passenger compartment. The exhaust-system noise at the exhaust-emission point (tailpipe) can also cause bodywork resonances.

The total volume of the passenger-car muffler system is approximately three to eight times the engine's swept volume.

Exhaust systems
a) Without, b) With catalytic-converter system.
1 Front muffler, 2 Catalytic-converter system, 3 Center muffler, 4 Rear muffler.



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Depending on swept volume and type of muffler, the exhaust system weighs between 8 and 40 kg.

Catalytic converters

The catalytic-converter housing is of heat-resistant, high-quality steel. It contains actively-coated ceramic monoliths. In order to compensate for the differing coefficients of thermal expansion of steel and ceramics, and to protect the sensitive monolith against bumps and vibrations, a flexible mounting is used. Two different types of mounting have been developed:

The wire-knit mounting, of highly heat-resistant stainless steel, is insensitive to extreme exhaust-gas temperatures and to pronounced gas pulsations in the high-speed range. Because of its poor heat-insulation qualities, the pipes and the body of the catalytic converter must often also be insulated.

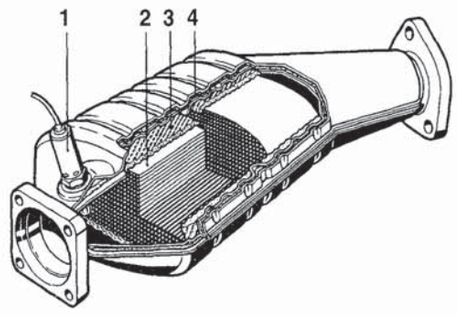
The swelling-mat mounting has come to the forefront in application. This mounting is made of ceramic fiber felt, composed of aluminum silicate fibers and expanding mica particles. The two substances are combined using acrylic latex. Under the influence of temperature the matting expands and presses the monolith into an immovable position. As the swell matting is a good insulator, there is no need for additional insulation. However, if the exhaust gases cause excessive heating, the pressure on the monoliths can reach such a level that there is the danger of fracture. If the exhaust-gas temperatures are not high enough, the pressing force exerted on the monolith is insufficient, allowing the monolith to move and possibly be destroyed. Exhaust-gas pulsation may lead to erosion in the swell matting.

In order to restrict linear expansion and to achieve better mixing of the exhaust gases, several monoliths are frequently used in one catalytic converter. The shape of the inflow funnel into the catalytic converter must be designed carefully, so that the exhaust gases flow through the monolith evenly. The external shape of the ceramic body depends on the space available underneath the vehicle, and may be triangular, oval or round.

The metal catalytic converter is an alternative to the ceramic monolith. It is made of finely corrugated, 0.05 mm thick metal foil, wound and hard-soldered in a high-temperature process. As in the case of the ceramic catalytic converter, the surface is coated with catalytically effective material. As a result of its thin walls, more channels can be accommodated in the same area. That means less resistance to the exhaust gas, which is beneficial with regard to performance optimization in high-performance vehicles.

Catalytic converters also have an acoustic effect. As a result of the narrow ceramic pipes in the monolith, numerous small sound sources are formed. The

Dual-bed three-way catalytic converter
1 Lambda sensor for lambda closed-loop control, 2 Monolith, 3 Wire-knit mounting, 4 Heat-insulated double shell.



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sound waves are thereby partially extinguished by interference or are damped by friction.

When designing the exhaust system, careful catalytic-converter matching is imperative, as its high level of flow resistance has a considerable influence on the system's vibration characteristics as well as on the performance of the engine (see also Catalytic Afterburning, page 521).

Mufflers

Mufflers (or silencers) are intended to smooth the exhaust-gas pulsations and make them as inaudible as possible. There are basically two physical principles involved: reflection and absorption. Mufflers also differ according to these principles. However, they mostly comprise a combination of reflection and absorption. As mufflers together with the pipes of the exhaust system form an oscillator with natural resonance, the position of the mufflers is highly significant for the quality of sound-damping. The objective is to tune the exhaust systems as low as possible, so that their natural frequencies do not excite bodywork resonances. To avoid structure-borne noise and to provide heat insulation against the underbody of the vehicle, mufflers often have double walls and an insulating layer. Depending on the space available underneath the vehicle, mufflers are produced either as "winding cups" or from half-shells.

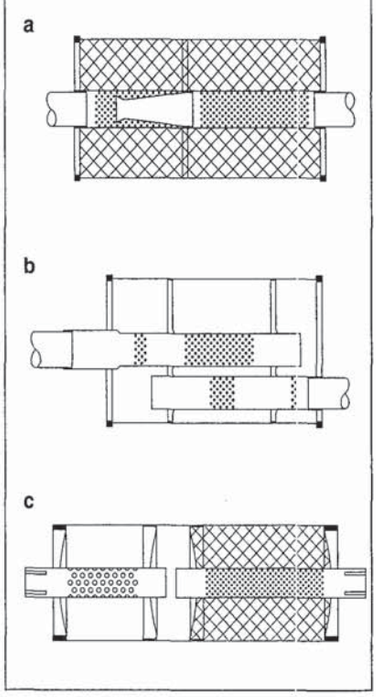
Reflection mufflers consist of chambers of varying lengths which are connected together by pipes. The differences in the cross-sections of the pipes and the chambers, the diversion of the exhaust gases, and the resonators formed by the connecting pipes with the chambers, produce muffling which is particularly effective at low frequencies. The more such chambers are used, the more efficient is the muffler.

Reflection mufflers cause a higher exhaust-gas backpressure, as a rule have greater power loss, and are heavier.

Absorption mufflers are constructed with one chamber, through which a perforated pipe is passed. The chamber is filled with sound-deadening material. The sound enters the absorption material through the perforated pipe and is converted into heat by friction. The absorption material usually consists of long-fiber mineral wool (basalt or rock wool) with a bulk density of 120...150 g/l. The level of muffling depends on the bulk density, the sound-absorption grade of the material, and on the length and coating-thickness of the chamber. Damping takes place across a very broad band, but only begins at higher frequencies. The shape of the perforations, and the fact that the pipe passes through the wool ensures that the material is not blown out by the pulsation of the exhaust gas. Sometimes the mineral wool is protected by a layer of stainless-steel wool around the perforated pipe.

Absorption mufflers are principally used as rear mufflers.

Muffler (silencer) principles
a) Absorption muffler,
b) Reflection muffler,
c) Combination of a) and b).



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Acoustic tuning devices

A number of different components can be used to remove disturbing frequency areas of the noise emitted from the tailpipe.

The Helmholtz resonator damps sound in its natural frequency range and functions as a suction resonator. It is a through-flow resonator and amplifies at its natural frequency, but thereafter it has a broad damping range.

Pipes perforated with holes work in a similar way to a watering-can rose. The one large sound source, the pipe, is converted into many small sound points, formed by the perforations. A broad-band filter effect occurs as a result of interference and swirling of the exhaust gas.

Venturi nozzles damp low-frequency sound. They must be designed so that the flow speed in the nozzle throat is always below the speed of sound. The funnel must be set at a specific angle, as otherwise hissing noises occur.

Soot filters

In order to remove solid particles (particulates) from diesel-engine exhaust gas, soot filters are in development. Various kinds of filter system are used, e.g., steel-wool filters, ceramic-monolith filters, ceramic-coil filters etc.

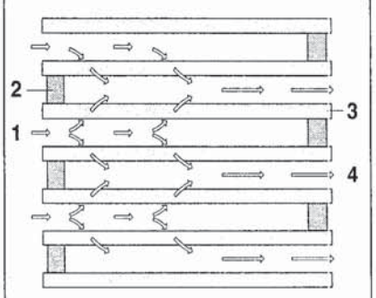
The ceramic-monolith filter currently represents the best compromise with regard to the requirements made of the filter. In contrast to the flow-through catalytic-converter monoliths, the channels for the soot filter are alternatingly opened and closed, so that the particle-laden exhaust gas is forced to flow through the uncoated, porous walls of the honeycomb structure. The particles are deposited in the pores. Depending on the porosity of the ceramic body, the effectiveness of these filters ranges from 70 to 90 %.

In order to guarantee full functioning of the filters they must be regenerated at certain intervals. Two cleaning processes are possible; in both cases the soot particles are burnt away:

In the chemical process, additives in the fuel reduce the flammability of the soot particles to the usual exhaust-gas temperature. The secondary emissions arising as a result of the additives may have a disadvantageous effect.

In the thermal process, a high-power heating element is connected, which raises the exhaust-gas temperature to approx. 700 °C. The regeneration is most simply carried out with the engine switched off. The filter regeneration point is ascertained either via a time control or an aneroid box. If it is necessary to regenerate the filter while the vehicle is running, two filters can be fitted which are alternatingly either filtering or being regenerated. This is, however, very cost-intensive. A further possibility is to divert the exhaust gases via a sound muffler during regeneration, whereby the exhaust gases are emitted unfiltered for approx. 5% of the journey. Heating elements are also being developed which permit simultaneous regeneration and filtering of engine exhaust gases (full-flow regeneration).

Ceramic soot filter (principle)
1 Exhaust input, 2 Ceramic plug, 3 Cell partition, 4 Exhaust-gas exit.



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Engine management for spark-ignition (SI) engines

Requirements

SI-engine torques

The power output P from a spark-ignition (SI) engine is determined by the available clutch torque and the engine speed.

The clutch torque is produced from the torque generated by the combustion process, reduced by the friction torque (friction losses in the engine) and the charge-cycle losses, and the torque required for operating the auxiliary systems.

The combustion torque is generated in the power cycle and determined by the following variables:

- the air mass that is available for combustion once the intake valves have closed,
- the fuel mass available at the same time, and
- the point at which the ignition spark initiates combustion of the air/fuel mixture.

Primary function of engine management

The primary function of engine management is to adjust the torque generated by the engine. For this purpose, all the variables that influence the torque are controlled in the various engine-management subsystems.

Cylinder-charge control

In Bosch engine-management systems with an electronic accelerator pedal (EGAS), the required charging of the engine cylinders with air is determined and the throttle valve opened accordingly in the "cylinder-charge control" subsystem. In conventional injection systems, the driver directly controls the opening of the throttle valve by pressing the accelerator pedal.

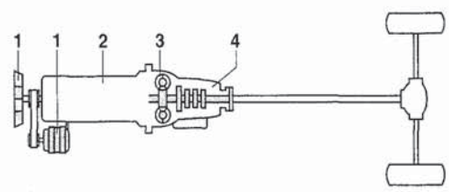
A/F mixture formation

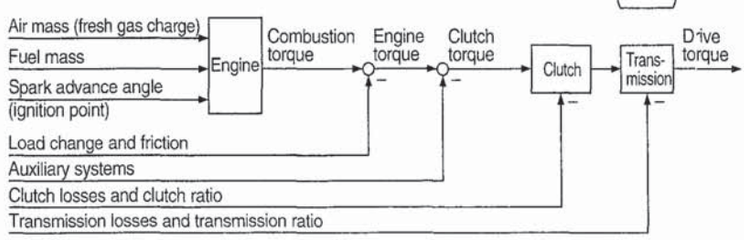
In the "mixture formation" subsystem, the associated fuel mass is calculated and from this the necessary injection time and the optimum injection point are determined.

Ignition

In the "ignition" subsystem, the crankshaft angle is determined at which the ignition spark ignites the mixture at the correct time.

Drivetrain torques
1 Auxiliary systems (alternator, A/C compressor etc.),
2 Engine,
3 Clutch,
4 Transmission.





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The objective of this management system is to make available the torque requested by the driver, and at the same time to satisfy the exacting demands placed on exhaust emissions, fuel consumption, power output, comfort and safety.

Cylinder charge

Constituent elements

The gas mixture in the cylinder after the intake valves have closed is termed the cylinder charge. This charge consists of the supplied fresh A/F mixture and residual exhaust gas.

Fresh A/F mixture

The constituent elements of the fresh mixture drawn in are fresh air and the fuel suspended in it. The majority of the fresh air flows through the throttle valve; additional fresh mixture can be drawn in through the evaporative-emissions control system (if fitted). The air supplied via the throttle valve and present in the cylinder after the intake valves have closed is the decisive factor in the work performed at the piston during combustion, and thus in the torque delivered by the engine.

Measures for increasing maximum torque and maximum engine power therefore almost always necessitate an increase in the maximum possible cylinder charge. The theoretical maximum charge is predetermined by the piston displacement.

Residual exhaust gas

The share of residual exhaust gas in the charge is formed

- by the exhaust-gas mass which remains in the cylinder and is not discharged during the period when the exhaust valve is open, and
- in systems with exhaust-gas recirculation (EGR), by the mass of the recirculated exhaust gas.

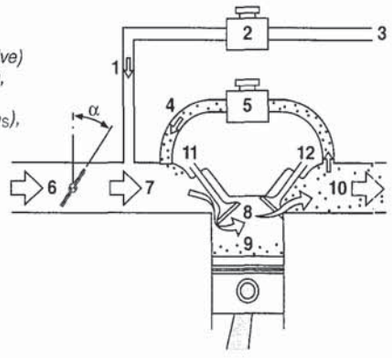
The residual exhaust-gas share is determined by the charge cycle.

The residual exhaust-gas mass does not contribute directly to the combustion process but does influence ignition and the course of the combustion process. This residual exhaust-gas share can therefore be perfectly desirable when the engine is in part-load operation.

In order for a desired torque to be generated, the reduction in the fresh-mixture volume (due to the residual-gas quantity) must be compensated for by increasing the throttle-valve opening. This reduces

Cylinder charge in the SI engine

1 Air and fuel vapor,
2 Purge valve with variable valve-opening cross-section,
3 Connection to evaporative-emissions control system,
4 Exhaust gas,
5 Exhaust-gas recirculation valve (EGR valve) with variable valve-opening cross-section,
6 Air mass flow (ambient pressure p_U),
7 Air mass flow (intake-manifold pressure p_S),
8 Fresh A/F-mixture charge (combustion-chamber pressure p_B),
9 Residual exhaust-gas charge (combustion-chamber pressure p_B),
10 Exhaust gas (exhaust-gas back pressure p_A),
11 Intake valve,
12 Exhaust valve.
α Throttle-valve angle.



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the engine's pumping losses, and fuel consumption drops as a result. A specifically introduced residual exhaust-gas share can likewise influence combustion and thus the emission of nitrogen oxides (NO_x) and unburnt hydrocarbons (HC).

Control

In a spark-ignition engine with external mixture formation, the power output is proportional to the air mass flow drawn in. In future, it will also be possible to directly control the direct-injection SI engine operating with lean A/F mixtures via variation of the injected fuel mass.

Throttle valve

The throttle valve is used when the engine power, and thus (at a specific engine speed) the engine torque, are to be controlled by means of the air mass flow. When the throttle valve is not fully open, the air drawn in by the engine is throttled, thereby reducing the torque generated. This throttling effect is dependent on the position and thus on the opening cross-section of the throttle valve.

The maximum engine torque is achieved when the throttle valve is fully open (see Fig.).

Charge cycle

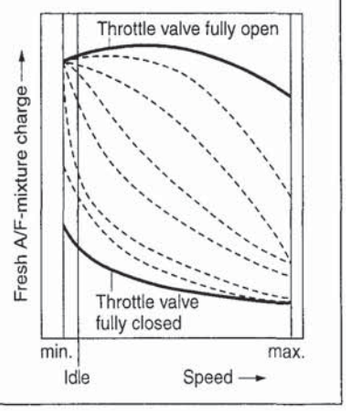
The charge cycle of fresh A/F mixture and residual exhaust gas is controlled by the appropriate opening and closing of the intake and exhaust valves. The cams on the camshaft determine the points at which the valves open and close (valve timing) and the course of the valve lift. This influences the charge-cycle process and thus also the amount of fresh A/F mixture available for combustion.

The valve overlap, i.e. the overlapping of the opening times of the intake and exhaust valves, has a decisive impact on the residual exhaust-gas mass in the cylinder. This situation involves "interior" exhaust-gas recirculation. The residual exhaust-gas mass can also be increased by "exterior" exhaust-gas recirculation. In this case, an additional EGR valve connects the intake manifold and exhaust manifold. When the valve is open, the engine draws in a mix of fresh A/F mixture and exhaust gas.

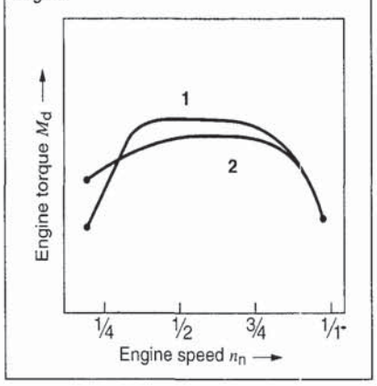
Supercharging

The obtainable torque is proportional to the charge of fresh A/F mixture. It s therefore possible to increase the maximum torque by compressing the air in the cylinder by means of dynamic supercharging, mechanical supercharging, or exhaust-gas turbocharging (P. 392).

Throttle map of an SI engine
– – – Intermediate position of throttle valve.



Torque curve for turbocharged engine compared with naturally aspirated engine at same rated power
1 Turbocharged engine, 2 Naturally aspirated engine.



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Fuel delivery with electric fuel pump

Function

The electric fuel pump must deliver sufficient quantities of fuel to the engine and maintain enough pressure for efficient injection under all operating conditions. Essential requirements include:

- maintaining flow rates between 60 and 200 liters/h at the rated voltage,
- maintaining fuel-system pressures of 300...450 kPa,
- the ability to pressurize the system during operation at 50...60 % of the rated voltage, important for cold-starting response.

In addition, the electric fuel pump is increasingly being used as the presupply pump for modern direct-injection systems, both for gasoline and for diesel engines. For gasoline direct-injection systems, at times pressures of up to 700 kPa must to be provided. This, together with the very high viscosity range when pumping diesel fuel, signifies new challenges facing the hydraulic and electric systems of the electric fuel pump.

Design

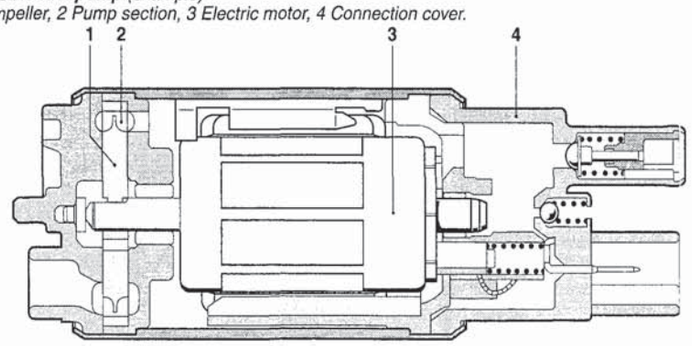
The electric fuel pump consists of:

- the end cover including the electrical connections, non-return valve (to maintain system pressure) and the hydraulic discharge fitting. Most end covers also include the carbon brushes for the drive-motor commutator and interference-suppression elements (inductance coils, with condensers in some applications).
- the electric motor with armature and permanent magnets. Electronically commutated (EC) fuel pumps are being developed for use with special fuels which feature for instance marked electrolytic effects, and for use in other environments which have negative effects on carbon-brush and commutator assemblies.
- a positive-displacement or flow-type pump assembly.

Positive-displacement pump

As the positive-displacement unit's pump element rotates it draws in fluid through the suction side and through a sealed area on its way to the high-pressure side. Electric fuel pumps fall into two categories, the roller cell and the internal-gear unit. Positive-displacement pumps provide good performance in high-pressure (400 kPa and above) systems. They also perform well at low supply voltages, i.e. the flow rate curve remains relatively "flat" and constant throughout a wide range of operating voltages. Efficiency ratings can be as high as 25 %. The unavoidable pressure pulses may cause noise; the extent of this problem varies according to the pump's design configuration and mounting location. Yet another disadvantage may be encountered with hot fuel, when the unit tends to pump gas instead of fuel, leading to reduced flow rates (problem potential varies according to in-

Electric fuel pump *(example)*
1 Impeller, 2 Pump section, 3 Electric motor, 4 Connection cover.



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stallation location). Standard positive-displacement pumps usually incorporate peripheral primary circuits to deal with this problem by discharging the gas.

While the flow-type pump has to a large extent replaced the positive-displacement pump in electronic gasoline injection systems for performing the classical function of the electric fuel pump, a new field of application has opened up for the positive-displacement pump in terms of the above-mentioned presupply for direct-injection systems with their significantly increased pressure requirements and viscosity range. This is especially true for the presupply of diesel and biodiesel.

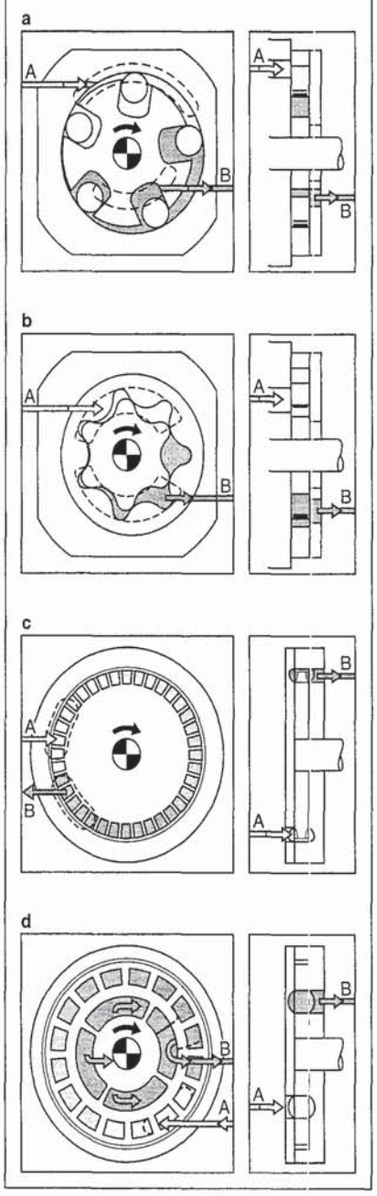
Flow-type pumps

Designs based on the principles used for the peripheral pump and the side-channel pump have become the standard for electric fuel pumps, with a slight preference for the side-channel pump as this tends to provide higher pressures and improved efficiency. An impeller equipped with numerous peripheral vanes rotates within a chamber consisting of two fixed housing sections. Each of these sections features a passage along the path of the impeller's vanes, with the openings on one end of the passage on a plane with the suction openings. From here they extend to the point where the fuel exits the pump at system pressure. Within the passage is a baffle element designed to prevent internal leakage. A small gas-discharge orifice (not necessary in diesel applications) located at a specified angular distance from the suction opening, improves performance when pumping hot fuel; this orifice facilitates the discharge of any gas bubbles which may have formed (with minimal leakage).

The pulses reflected between the impeller vanes and the fluid molecules result in pressurization along the length of the passage, inducing a spiral rotation of the fluid volume in the impellers and in the passages.

Because pressurization is continuous and virtually pulse-free, flow-type pumps are quiet in operation. Pump design is also substantially less complex than that of the positive-displacement unit. Single-stage pumps generate system pressures

Electric fuel-pump designs
a) Roller-cell pump,
b) Internal-gear pump,
c) Peripheral pump,
d) Side-channel pump.



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extending up to 450 kPa. Still higher system pressures, as will become necessary for brief periods in future for highly supercharged engines, and for engines with gasoline direct injection (see above), are possible, but under continuous-duty conditions such pressures would overload today's conventional electrical systems (permanent-magnet DC motors with conventional electromechanical commutation) and would result in a significantly reduced service life. The following remedial measures are being considered:

- High-pressure operation only when required → demand control of the electric fuel pump, e.g. with the aid of a timing module or another upstream device.
- Equipping of the fuel-pump motor with a carbon commutator in place of the conventional copper commutator so as to safeguard the service life also at high current and additionally with corrosive and/or high-viscosity fuels.
- For applications where the wide range of operating conditions and fuels place particularly high demands on the pump's versatility, work is proceeding on electronically commutated (EC) fuel-pump drives. Such an electrical system features unlimited service life.

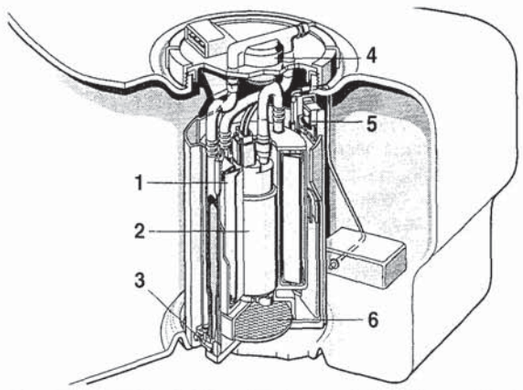
The efficiency ranges between 10 and approx. 20 %. The fuel systems of newly designed vehicles with spark-ignition engines rely almost exclusively on flow-type pumps for fuel delivery.

Electric fuel pumps: integration in injection system and in fuel tank

Whereas the first electronic fuel-injection systems almost always featured electric fuel pumps designed for in-line installation outside the tank, current and more recent applications tend to have in-tank installation as a standard feature. The electric fuel pump is one of the elements within the in-tank units now being designed to include an increasingly wide array of components such as: the suction filter, a fuel-baffle chamber to maintain delivery during cornering (usually with its own "active" supply based on a suction-jet pump or a separate primary circuit in the main electric pump), the fuel gauge sensor, and a variety of electrical and hydraulic connections.

Another advance is the returnless fuel system (RLFS), usually in the form of an in-tank unit with an integral fuel-pressure regulator designed to maintain a continuous return circuit within the in-tank assembly. A pressure-side fine-mesh fuel filter can also be incorporated in this unit. Further functions will in future be integrated in the delivery module, e.g. diagnostic devices for tank leakage, timing module for fuel-pump control.

In-tank unit: complete integrated assembly for returnless fuel systems
1 Fuel filter, 2 Electric fuel pump, 3 Suction-jet pump (regulated), 4 Fuel-pressure regulator, 5 Fuel-gauge sensor, 6 Suction strainer.



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A/F-mixture formation

Influencing variables

Air-fuel (A/F) mixture

To be able to operate, a spark-ignition engine requires a specific air-fuel mixture ratio. Ideal theoretical complete combustion is available at a mass ratio of 14.7:1. This is also termed the stoichiometric ratio. I.e.: an air mass of 14.7 kg is needed to burn a fuel mass of 1 kg. Or expressed as a volume: 1 *l* fuel burns completely in roughly 9500 *l* air.

The specific fuel consumption of a spark-ignition engine is essentially dependent on the mixture ratio of the A/F mixture. It is necessary to have an excess of air in order to ensure genuine complete combustion, and thus as low a fuel consumption as possible. Limits are imposed though by the flammability of the mixture and the available combustion time.

The A/F mixture also has a decisive impact on the efficiency of the exhaust-gas treatment systems. State-of-the-art technology is represented by the three-way catalytic converter. This, though, needs a stoichiometric A/F ratio in order to operate with maximum efficiency. Such a catalytic converter helps to reduce harmful exhaust-gas constituents by more than 98%.

The engines available today are therefore operated with a stoichiometric mixture as soon as their operating status permits this.

Certain engine operating states require mixture corrections. Specific corrections of the mixture composition are necessary e.g. when the engine is cold. The mixture-formation (carburation) system must therefore be in a position to satisfy these variable requirements.

Excess-air factor

The excess-air factor λ (lambda) has been chosen to designate the extent to which the actual air-fuel mixture differs from the theoretically necessary mass ratio (14.7:1):

λ = Ratio of supplied air mass to air requirement with stoichiometric combustion.

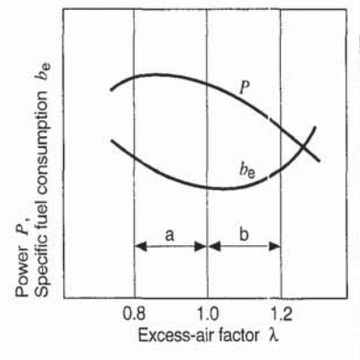
$\lambda = 1$: The supplied air mass corresponds to the theoretically necessary air mass.

$\lambda < 1$: There is an air deficiency and thus a rich mixture. Maximum power output at $\lambda = 0.85...0.95$.

$\lambda > 1$: There is an excess of air or a lean mixture in this range. This excess-air factor is characterized by reduced fuel consumption and reduced power output. The maximum value for λ that can be achieved – the so-called "lean-burn limit" – is very heavily dependent on the engine design and on the mixture-formation system used. The mixture is no longer ignitable at the lean-burn limit. Combustion misses occur and this is accompanied by a marked increase in uneven running.

Spark-ignition engines with manifold injection achieve their peak power output at an air deficiency of 5...15% ($\lambda = 0.95...0.85$), and their lowest fuel consumption at an air excess of 10...20% ($\lambda = 1.1...1.2$).

Effect of excess-air factor λ on power P and specific fuel consumption b_e
a Rich mixture (air deficiency),
b Lean mixture (air excess).



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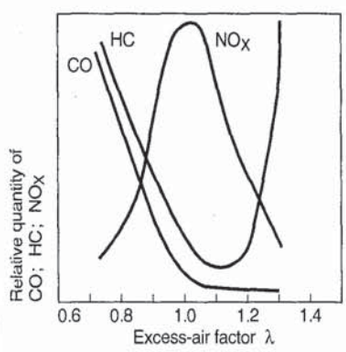
The graphs show the dependence of power output, specific fuel consumption and pollutant buildup on the excess-air factor for a typical engine with manifold injection. It can be deduced from these graphs that there is no ideal excess-air factor at which all the factors assume the most favorable value. For engines with manifold injection, excess-air factors of $\lambda = 0.9...1.1$ have proven effective in realizing "optimal" consumption at "optimal" power output.

Engines with direct injection and charge stratification involve different combustion conditions such that the lean-burn limit is significantly higher. These engines can therefore be operated in the part-load range with significantly higher excess-air factors (up to $\lambda = 4$).

For the treatment of exhaust gas by a three-way catalytic converter, it is absolutely essential to adhere exactly to $\lambda = 1$ with the engine at normal operating temperature. In order to do so, the air mass drawn in must be precisely determined and an exactly metered fuel mass added to it.

For optimum combustion in today's common manifold-injection engines, not only is a precise injected fuel quantity necessary, but also a homogeneous A/F mixture. This necessitates efficient fuel atomization. If this precondition is not satisfied, large fuel droplets will precipitate on the intake manifold or the combustion-chamber walls. These large droplets cannot fully combust and will result in increased hydrocarbon emissions.

Effect of excess-air factor λ on pollutant composition in untreated exhaust gas



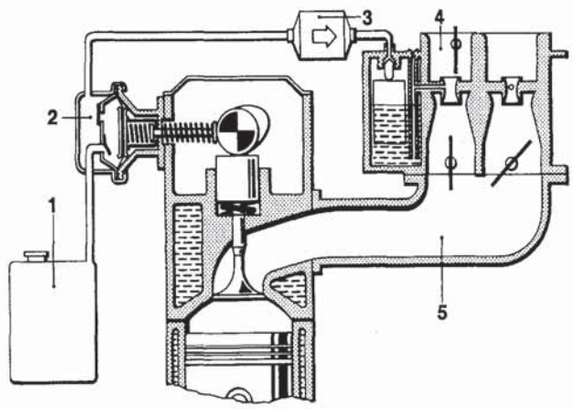
Mixture-formation systems

It is the job of fuel-injection systems, or carburetors, to furnish an A/F mixture which is adapted as well as possible to the relevant engine operating state. Injection systems, especially electronic systems, are better suited to maintaining narrowly defined limits for the mixture composition. This is advantageous with regard to fuel consumption, driving performance and power output. The result of increasingly stringent exhaust-emissions legislation in the automotive sector is that today, injection systems have completely superseded carburetors.

Today, the automotive industry almost exclusively uses systems in which the mixture formation takes place outside the combustion chamber. However, systems with interior mixture formation, i.e. where the fuel is injected directly into the combustion chamber, already formed the basis of the first gasoline injection systems. These systems are increasing in importance as they are very well suited to reducing fuel consumption even further.

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Schematic of a carburetor system
1 Fuel tank, 2 Fuel supply pump, 3 Fuel filter, 4 Carburetor, 5 Intake manifold.



Carburetors

Carburetor systems

The fuel is transported from the fuel tank to the carburetor by a fuel pump (generally a diaphragm unit) powered by the camshaft or distributor shaft. The system is designed to limit the maximum supply pressure. A fine-mesh fuel filter can be installed upstream or downstream from the pump as required.

Carburetor types

Downdraft carburetors

Downdraft carburetors are the most common type. Designs featuring optimized float chamber and metering-jet configurations result in efficient units. These designs work in conjunction with the corresponding intake-manifold layouts for optimum mixture formation and distribution.

Horizontal-draft carburetors

Horizontal-draft carburetors (familiar as fixed-venturi and constant-depression units) are useful for minimizing engine height.

Constant-depression carburetors feature venturi cross sections which vary in size during operation to maintain essentially constant vacuum levels at the fuel outlet. The variation in intake cross section is provided by a pneumatically-actuated plunger; attached to the plunger is a needle which regulates the fuel quantity.

Venturi configurations

The single-throat carburetor with one venturi is the least expensive design.

The two-stage carburetor featuring two venturis provides convenient tuning for individual applications and has become the standard in 4-cylinder applications. The first barrel controls part-throttle operation, while the second venturi opens for maximum power.

The double-barrel carburetor features two carburetor sections sharing a single float chamber and operating in parallel, making it ideal for use on 6-cylinder engines. The two-stage four-barrel carburetor has four venturi fed from a single float chamber.

Design and operating principles

The driver uses the accelerator pedal to vary the throttle valve's aperture so that the airflow into the engine is varied and with it the engine's power output. The carburetor varies the amount of fuel metered to the engine to reflect the current intake air flow. Together with the needle valve, the float regulates the fuel flow to the carburetor while maintaining a constant fuel level in the float chamber.

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Airflow is monitored by an air funnel designed to induce a venturi effect. The progressively narrower diameter increases the velocity of the air, producing a corresponding vacuum at the narrowest point. The resulting pressure differential relative to the float chamber – which can be further augmented with a boost venturi – is exploited to extract fuel from the float chamber. The jets and metering systems adapt fuel delivery to airflow.

Fuel-metering systems

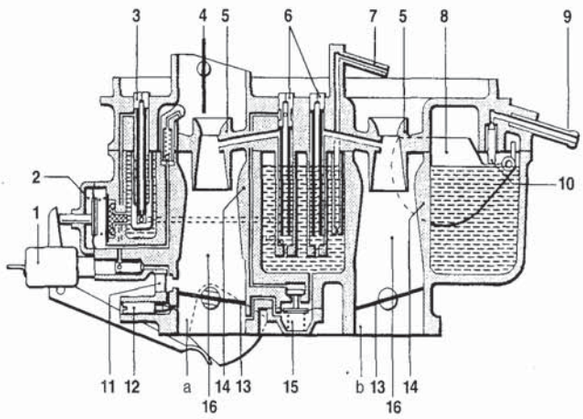
Main system
The fuel is metered by the main jet. Correction air is added as a delivery aid to the fuel through side orifices in the venturi tube.

Idle and progression system
At idle, the vacuum which the air stream produces at the fuel outlet is not sufficient to withdraw fuel from the main system. For this reason, there is a separate idle system with an outlet located downstream from the throttle valve at the point of maximum vacuum. The emulsion required for idling emerges from the idle circuit after initial processing by the idle fuel and air-correction jets.

During transitions to the main metering system the throttle valve controls a series of orifices, or a slit, drawing fuel from the idle circuit.

Other systems
These basic devices are supplemented by a range of additional systems. These are designed to adapt carburetor performance for warm operation (part-throttle control, full-throttle enrichment), to compensate for fuel accumulation with n the intake-manifold during acceleration (accelerator pump) and to meet the special engine requirements encountered during starting and in the warm-up phase. Other supplementary systems include lambda closed-loop mixture control and devices to deactivate the fuel supply during trailing-throttle operation.

Schematic of a two-stage carburetor
a) Primary stage, b) Secondary stage.
1 Idle cutoff valve, 2 Accelerator pump, 3 Idle circuit, 4 Choke, 5 Boost venturi, 6 Main systems with venturi tubes, 7 Full-throttle enrichment, 8 Float, 9 Fuel supply, 10 Needle valve, 11 Bypass plug, 12 Idle mixture screw, 13 Throttle valves, 14 Venturi wall, 15 Part-throttle control valve, 16 Venturi chamber.



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Electronically-controlled carburetor system (ECOTRONIC)

Basic carburetor

The basic carburetor is restricted to the throttle valve, float system, idle and transition systems, main system and choke. An idle-air control system with a choke-activated needle jet is also provided.

Additional components and actuators

The throttle-valve actuator is an electropneumatic servo device for controlling the cylinder charge. The actuator's plunger moves the throttle valve via a lever attached to the carburetor's throttle shaft.

The choke valve actuator is a final-control element designed to adapt the mixture in response to variations in engine operating conditions. This unit closes the choke valve to enrich the mixture by raising the pressure differential (vacuum) at the main jets while simultaneously increasing flow rates from the idle circuit.

Sensors

The throttle-valve potentiometer monitors the throttle valve's position and travel. One temperature sensor monitors the engine's operating temperature while a second sensor can be installed if necessary to monitor the temperature within the intake manifold.

The idle switch serves to identify trailing-throttle operation; it can be replaced by appropriate software in the electronic control unit (ECU).

Electronic control unit (ECU)

The ECU's input circuit converts incoming analog signals into digital form. The processor performs further operations with the input data in order to calculate output values with reference to the programmed data map. The output signals control several functions, including regulation of the servo elements that operate the choke valve and main throttle valve.

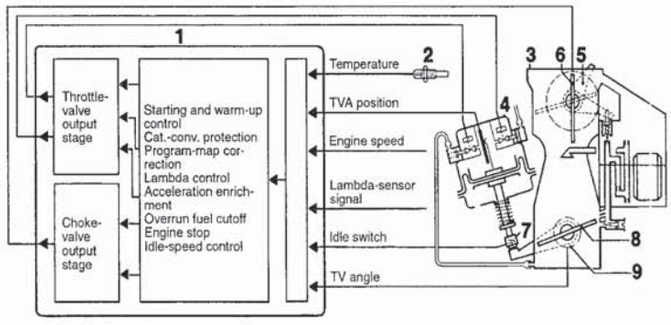
Basic functions

The basic carburetor determines the primary functions of the system. The idle, transition and full-throttle systems all contribute to matching performance to the programmed curves. The base calibrations can be intentionally "lean", as the choke-valve control can provide a corrective enrichment.

Electronic functions

Electronic open and closed-loop control circuits regulate a number of secondary operations within the ECU. Several of these are illustrated below. Further functions may include: ignition control, transmission-shift control, fuel consumption displays and diagnosis capabilities.

Schematic of an electronically controlled carburetor (ECOTRONIC)
1 ECU, 2 Temperature sensor, 3 Carburetor, 4 Throttle actuator, 5 Choke actuator, 6 Choke valve, 7 Idle switch, 8 Throttle valve, 9 Throttle potentiometer.



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Gasoline injection systems

Systems for external A/F mixture formation

Gasoline injection systems for external mixture formation are identified by the fact that the air-fuel mixture is created outside the combustion chamber (in the intake manifold).

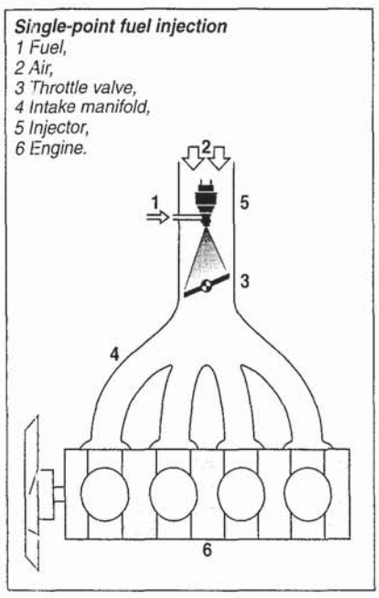
Single-point injection (SPI)
Single-point injection is an electronically controlled injection system in which an electromagnetic fuel injector injects the fuel intermittently into the intake manifold at a central point ahead of the throttle valve. The Bosch single-point injection systems are called Mono-Jetronic and Mono-Motronic.

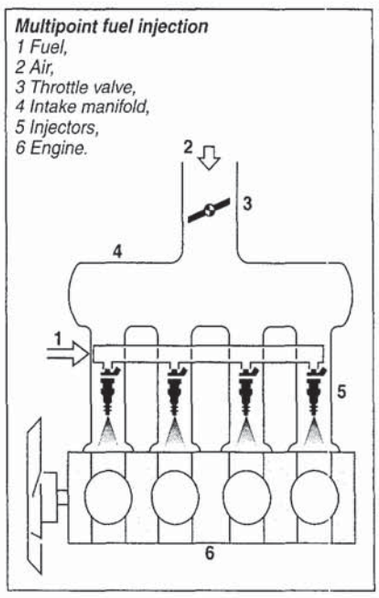
Multipoint injection (MPI)
Multipoint injection creates the ideal preconditions for satisfying the demands placed on a mixture-formation system. In multipoint injection systems, each cylinder is assigned a fuel injector, which injects the fuel directly ahead of that cylinder's intake valve. Examples of such systems are KE- and L-Jetronic with their respective variants.

Mechanical injection system:
The K-Jetronic system operates without a drive and injects the fuel continuously. The injected fuel mass is not determined by the fuel injector but rather prespecified by the fuel distributor.

Combined mechanical-electronic injection system:
KE-Jetronic is based on the mechanical basic system of K-Jetronic. Thanks to the extended acquisition of operating data, this system facilitates electronically controlled supplementary functions in order to adapt the injected fuel quantity more exactly to the different engine operating states.

Electronic injection systems:
Electronically controlled injection systems inject the fuel intermittently with electromagnetically actuated fuel injectors. The injected fuel mass is determined by the injector opening time (for a given pressure drop across the injector).





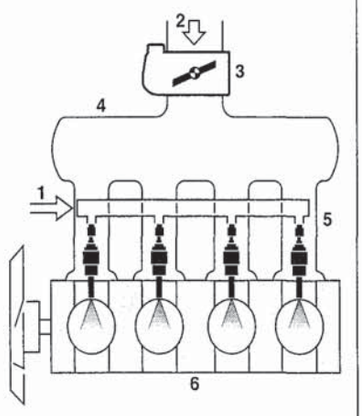
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Examples: L-Jetronic, LH-Jetronic, and Motronic as an integrated engine management system.

The high standards required of a vehicle's smooth running and exhaust-emissions necessitate high demands being made on the A/F mixture composition of each working cycle. Precisely timed injection is significant as well as precise metering of the injected fuel mass in accordance with the air drawn in by the engine.

In modern multipoint injection systems, therefore, not only is each engine cylinder assigned an electromagnetic fuel injector but also this fuel injector is activated individually for each cylinder. In this way, both the fuel mass appropriate to each cylinder and the correct start of injection are calculated by the control unit (ECU). Injecting the precisely metered fuel mass directly ahead of the cylinder intake valve(s) at the correct moment in time improves mixture formation. This, in turn, helps to a large extent in preventing wetting of the intake-manifold walls with fuel, which can result in temporary deviations from the desired lambda value during transient engine operation. The advantages of multipoint injection can thus be fully exploited.

Direct injection (DI)
1 Fuel,
2 Air,
3 Throttle-valve (EGAS),
4 Intake manifold,
5 Injectors,
6 Engine.



The engine intake manifolds thus carry only the combustion air and can therefore be optimally adapted to the gas-dynamic requirements of the engine.

Systems for internal A/F mixture formation

In direct-injection systems for internal mixture formation, the fuel is injected directly into the combustion chambers by electromagnetically actuated fuel injectors. Each cylinder is assigned a fuel injector. Mixture formation takes place inside the cylinder. To ensure efficient combustion, it is essential that the fuel be finely atomized when leaving the injectors.

In normal operation, a direct-injection engine draws in only air and not an air-fuel mixture, as is the case in conventional injection systems. Herein lies an advantage of this new system: no fuel can precipitate on the intake-manifold walls. With external mixture formation, the air-fuel mixture is generally present throughout the entire combustion in a homogeneous state and in a stoichiometric ratio. On the other hand, formation of the mixture in the combustion chamber permits two completely different operating modes:

Stratified-charge operation

In stratified-charge operation, the mixture only has to be combustible in the area around the spark plug. The remaining section of the combustion chamber thus only contains fresh mixture and residual exhaust gas without unburnt fuel. In the idle and part-load ranges, this creates an altogether highly lean mixture and thus a reduction in the fuel consumption.

Homogeneous operation

In homogeneous operation, as with external mixture formation, there is a homogeneous mixture throughout the entire combustion chamber; and the entire fresh air available in the combustion chamber takes part in the combustion procedure. For this reason, this operating mode is used in the full-load range.

MED-Motronic is the management system for direct-injection gasoline engines.

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Single-point injection systems

Single-point fuel injection has advanced beyond the compact fuel-injection system stage to become part of a comprehensive engine-management system.

The various single-point injection systems differ in the design of the central-injection unit. All systems feature an injector located above the throttle plate; they differ from multipoint injection units in that they frequently operate at low pressure (0.7...1 bar). This means that an inexpensive, hydrodynamic electric fuel pump can be used which is generally in the form of an in-tank unit. The injector is flushed continuously by the fuel flowing through it in order to inhibit the formation of air bubbles. This arrangement is an absolute necessity in such a low-pressure system. The designation "Single-Point Injection (SPI)" corresponds to the terms Central Fuel Injection (CFI), Throttle Body Injection (TBI) and Mono-Jetronic (Bosch).

Mono-Jetronic

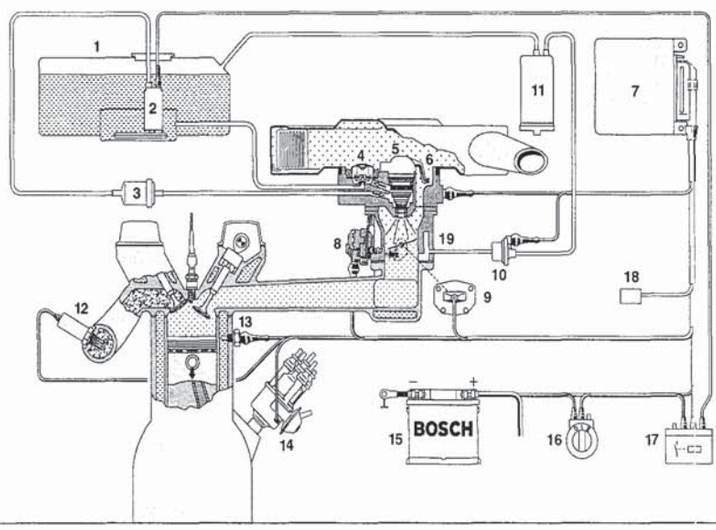
Mono-Jetronic is an electronically controlled, low-pressure single-point injection system for 4-cylinder engines, and features a centrally located solenoid-controlled fuel injector. At the heart of the system is the central injection unit, which uses the throttle valve to meter the intake air while injecting the fuel intermittently above the throttle valve. The intake manifold then distributes the fuel to the individual cylinders. Various sensors monitor all important engine operating data, which are then are used to calculate the triggering signals for the injectors and other system actuators.

Central injection unit

The injector is located above the throttle, in the intake-air path, in order to ensure homogeneous mixtures and consistent cylinder-to-cylinder distribution. The fuel spray is directed into the sickle-shaped orifice between the housing and throttle plate, whereby fuel wetting of the intake-tract walls is inhibited to a great extent, and the high pressure differential pro-

Schematic of a Mono-Jetronic system

1 Fuel tank, 2 Electric fuel pump, 3 Fuel filter, 4 Pressure regulator, 5 Injector, 6 Air-temperature sensor, 7 ECU, 8 Throttle actuator, 9 Throttle potentiometer, 10 Canister-purge valve, 11 Carbon canister, 12 Lambda sensor, 13 Coolant-temperature sensor, 14 Ignition distributor, 15 Battery, 16 Ignition switch, 17 Relay, 18 Diagnostic connector, 19 Central injection unit.



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motes optimum mixture formation. The injector operates at a system pressure of 1 bar (referred to atmospheric pressure). Efficient fuel atomization ensures consistently good mixture distribution, even in the critical full-load range. Injector triggering is synchronized with the ignition pulses.

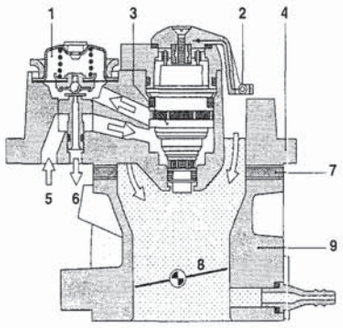
System control

In addition to the engine speed n, the main actuating variables for the injection system can include the air volume/air mass flow, the absolute manifold pressure, and the throttle position α. The (α/n) system applied with Mono-Jetronic can meet stringent emission requirements when used in conjunction with lambda closed-loop control and a 3-way catalytic converter. A self-adaptive system employs the signal from the lambda sensor as a reference to compensate for component tolerances and engine changes, thus maintaining high precision throughout the service life of the system.

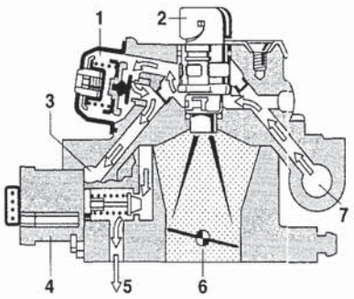
Adaptation functions

The injection time is extended to provide additional fuel for cold starts and during the post-start and warm-up phases. When the engine is cold, the throttle actuator adjusts the throttle position to supply more air to the engine, thus maintaining idle speed and exhaust emissions at a constant level. The throttle potentiometer recognizes the change in throttle position and initiates an increase in the fuel quantity via the ECU. The system regulates the enrichment for acceleration and full-throttle operation in the same way. The overrun fuel cutoff provides reductions in fuel consumption and in exhaust emissions during trailing-throttle operation. Adaptive idle-speed control lowers the idle speed and stabilizes it. For this purpose, the ECU issues a signal to the servomotor to adapt the throttle-valve position as a function of engine speed and temperature.

Mono-Jetronic central injection unit
1 Pressure regulator, 2 Air-temperature sensor, 3 Injector, 4 Upper part (hydraulics), 5 Fuel supply, 6 Fuel return, 7 Insulator plate, 8 Throttle valve, 9 Lower part.



Multec central injection unit (Opel)
1 Pressure regulator, 2 Injector, 3 Fuel return, 4 Stepper motor for idle-speed control, 5 To intake manifold, 6 Throttle valve, 7 Fuel inlet.



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Multipoint injection systems

K-Jetronic

Operating principle
– Continuous injection,
– Direct air-flow measurement.

K-Jetronic is a mechanical system which does not require an engine-driven injection pump. It meters a continuous supply of fuel proportional to the quantity of air being drawn into the engine.

Because of direct air-flow measurement, K-Jetronic also takes into account changes caused by the engine and permits the use of emission-control equipment, for which precise intake-air monitoring is an essential requirement.

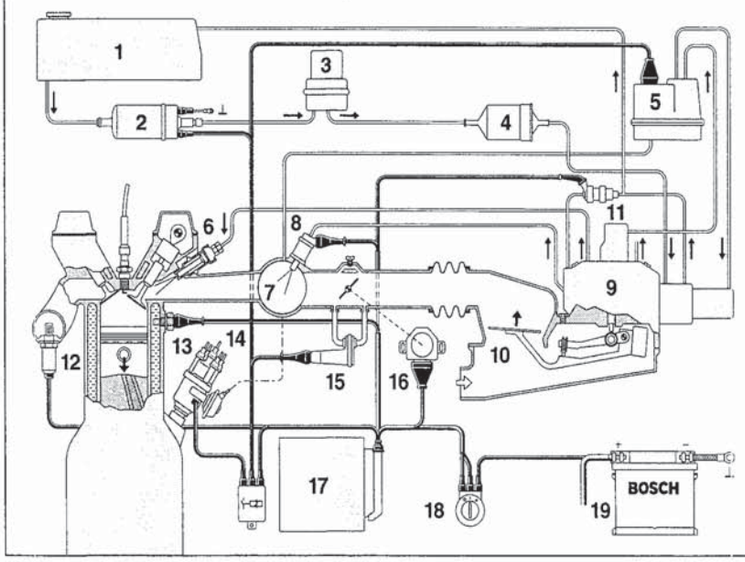
Operation
The intake air flows through the air filter, the air-flow sensor, and the throttle valve, before entering the intake manifold and continuing to the individual cylinders.

The fuel is delivered from the fuel tank by an electric (roller-cell) fuel pump. It then flows through the fuel accumulator and fuel filter to the fuel distributor. A pressure regulator in the fuel distributor maintains the fuel at a constant system pressure. The fuel flows from the fuel distributor to the injectors. Excess fuel not required by the engine is returned to the tank.

Mixture-control unit
The mixture-control unit consists of the air-flow sensor and the fuel distributor.

Air-flow sensor
The air-flow sensor consists of an air funnel and a pivoting air-flow sensor plate . A counterweight compensates for the weight of the sensor plate and pivot as-

Schematic of a K-Jetronic system
1 Fuel tank, 2 Electric fuel pump, 3 Fuel accumulator, 4 Fuel filter, 5 Warm-up regulator, 6 Injector, 7 Intake manifold, 8 Electric start valve, 9 Fuel distributor, 10 Air-flow sensor, 11 Frequency valve, 12 Lambda sensor, 13 Thermo-time switch, 14 Ignition distributor, 15 Auxiliary-air valve, 16 Throttle switch, 17 ECU, 18 Ignition switch, 19 Battery.



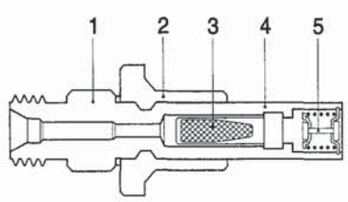
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sembly. The sensor plate is displaced by the air flow, while the control plunger in the fuel distributor exerts hydraulic counterpressure to maintain the system in a balanced state. The position of the airflow sensor plate provides an index of intake air flow, and is transmitted to the fuel distributor's control plunger by a lever.

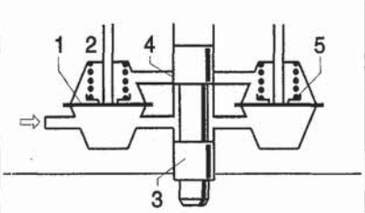
Fuel distributor

The amount of fuel supplied to the individual cylinders is regulated by varying the aperture of the metering slots in the fuel-distributor barrel. The number of rectangular-shaped metering slots in the barrel corresponds to the number of engine cylinders. The specific size of the metering-slot aperture depends on the control-plunger's position. In order to ensure constant pressure drop at the slots for various flow rates, a differential pressure regulator is located downstream of each metering slot.

Fuel injector
1 Hexagon, 2 Rubber molding, 3 Fine-mesh strainer, 4 Valve body, 5 Valve needle.



Fuel distributor in mixture-control unit
1 Diaphragm, 2 To injector, 3 Control plunger, 4 Metering slot, 5 Differential-pressure regulator.



Injector

The injector opens automatically at a pressure of approximately 3.8 bar, and has no metering function. It provides efficient mixture formation by opening and closing at a frequency of approx. 1500 Hz ("chatter").

It is held in place by a rubber molding. It is pressed, not screwed, into position. The hexagon serves to brace the injector when the fuel-supply line is screwed on.

Warm-up regulator

The warm-up regulator is controlled by an electrically-heated bimetallic element; it enriches the mixture in the warm-up phase by reducing the counterpressure (control pressure) exerted against the control plunger. A reduction in this control pressure means that the stroke of the airflow sensor plate for a given air flow increases (reflected by a correspondingly larger metering-slot aperture). The result is a richer mixture during warm-up.

Where desired, the warm-up regulator can be expanded to incorporate the following functions:
- full-throttle enrichment,
- acceleration enrichment,
- altitude compensation.

Auxiliary-air valve

The auxiliary-air valve, controlled by either a bimetallic spring or an expansion element, supplies the engine with additional air (which is monitored by the airflow sensor, but bypasses the throttle valve) during the warm-up phase. This supplementary air compensates for the cold engine's higher friction losses; it either maintains the normal idle speed or increases it in order to heat the engine and exhaust more quickly.

Electric start valve, thermo-time switch

The thermo-time switch activates the electric start valve as a function of engine temperature and elapsed time. During low-temperature starts, the start valve injects supplementary fuel into the intake manifold (cold-start enrichment).

Lambda closed-loop control

Open-loop control systems do not regulate the A/F ratio with enough accuracy to

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allow compliance with stringent emissions limits.

Lambda closed-loop control is required for operation of the 3-way catalytic converter. When it is installed, the K-Jetronic system must include an electronic control unit which uses the Lambda sensor's signal as its main input variable.

A solenoid frequency valve regulates the A/F mixture ratio by controlling the pressure differential at the metering slots. However, this principle cannot be applied to meet the more stringent emissions requirements scheduled for the future.

KE-Jetronic

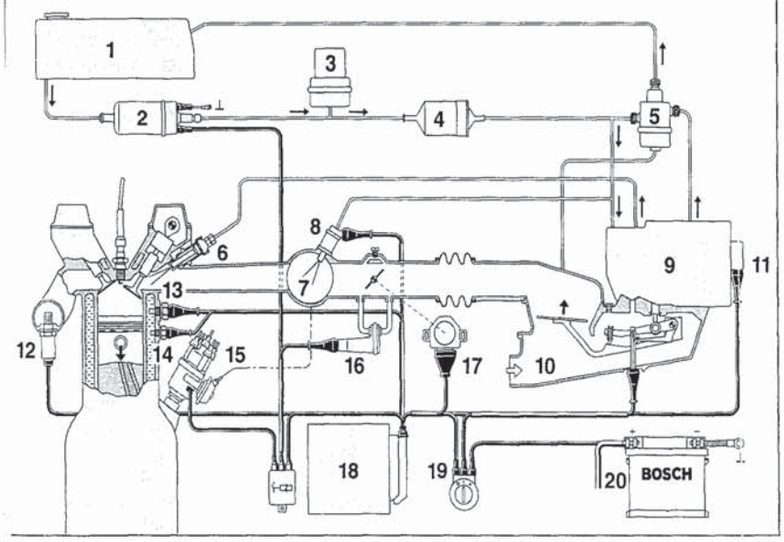
KE-Jetronic is an advanced version of the K-Jetronic system. KE-Jetronic includes an ECU for increased flexibility and supplementary functions. Additional components include:

- a sensor for the intake air flow,
- a pressure actuator for mixture ratio adjustment, and
- a pressure regulator which maintains system pressure at a constant level as well as providing a fuel-cutoff function when the engine is switched off.

Operation

An electric fuel pump generates the system pressure. The fuel flows through the fuel distributor, while a diaphragm regulator maintains the system pressure at a constant level. With K-Jetronic, the control circuit performs mixture corrections via the warm-up regulator. In contrast, with KE-Jetronic the primary pressure and the pressure exerted upon the control plunger are equal. The ratio is corrected by adjusting the pressure differential in all the fuel distributor's chambers simultaneously.

Schematic of a KE-Jetronic system
1 Fuel tank, 2 Electric fuel pump, 3 Fuel accumulator, 4 Fuel filter, 5 Fuel-pressure regulator, 6 Injector, 7 Intake manifold, 8 Electric start valve, 9 Fuel distributor, 10 Air-flow sensor, 11 Electrohydraulic pressure actuator, 12 Lambda sensor, 13 Thermo-time switch, 14 Coolant-temperature sensor, 15 Ignition distributor, 16 Auxiliary-air valve, 17 Throttle switch, 18 ECU, 19 Ignition switch, 20 Battery.

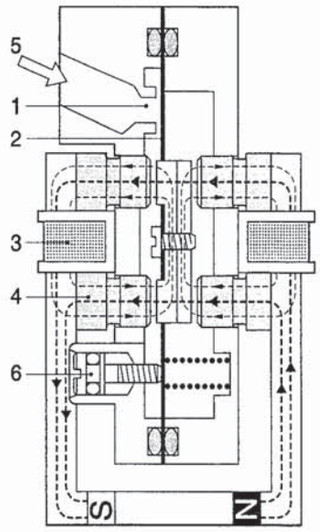


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The system pressure is present upstream from the metering slots, and applies a counterpressure to the control plunger. As with K-Jetronic, the control plunger is moved by an air-flow sensor flap. A damper unit prevents the oscillations that could be induced by the forces generated at the sensor flap. From the control plunger the fuel flows through the pressure actuator, the lower chambers of the differential-pressure valve, a fixed flow restrictor, and the pressure regulator, before returning to the fuel tank. Together with the flow restrictor, the actuator forms a pressure divider in which the pressure can be adjusted electrodynamically. This pressure is present in the lower chambers of the differential-pressure valves.

A pressure drop corresponding to the actuator current occurs between the actuator's two connections. This causes variations in the pressure differential at the metering slots, and alters the amount of fuel injected. The current can also be reversed to shut down the fuel supply completely. This feature can be employed for such functions as overrun fuel cutoff and engine-speed limitation.

Electrohydraulic pressure actuator
1 Nozzle, 2 Valve plate, 3 Coil, 4 Magnetic pole, 5 Fuel inlet, 6 Adjustment screw.



Electrohydraulic pressure actuator
This electrohydraulic actuator is flange-mounted on the fuel distributor. It is an electrically-controlled pressure regulator which operates using the nozzle/flapper-plate system. The mixture enrichment is directly proportional to the current flow.

Electronic control unit (ECU)
The ECU processes signals from the ignition (engine speed), temperature sensor (coolant temperature), throttle potentiometer (intake air flow), throttle switch (idle and overrun, WOT), starter switch, Lambda (O_2) sensor, pressure sensor and other sensors. Its most important functions are the control of:
- starting and post-start enrichment,
- warm-up enrichment,
- acceleration enrichment,
- full-throttle enrichment,
- overrun fuel cutoff,
- engine-speed limitation,
- idle-speed control,
- altitude compensation,
- closed-loop lambda control.

A coding switch (trim plug) makes it possible to select between operation with lambda control (with catalytic converter) and without it. This permits a choice between leaded and unleaded gasoline.

Lambda closed-loop control
The signal from the Lambda sensor is processed in the KE-Jetronic's ECU. The pressure actuator carries out the necessary adjustments.

L-Jetronic

Operating principle
- Air-flow measurement,
- Main controlled variables: air flow and engine speed,
- Intermittent injection.

L-Jetronic combines the advantages of direct air-flow measurement with the unique

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possibilities afforded by electronics. It is similar to K-Jetronic in that it recognizes all changes in engine condition (due to wear, combustion-chamber deposits, changes in valve setting). This ensures consistently good exhaust-gas composition.

Operation

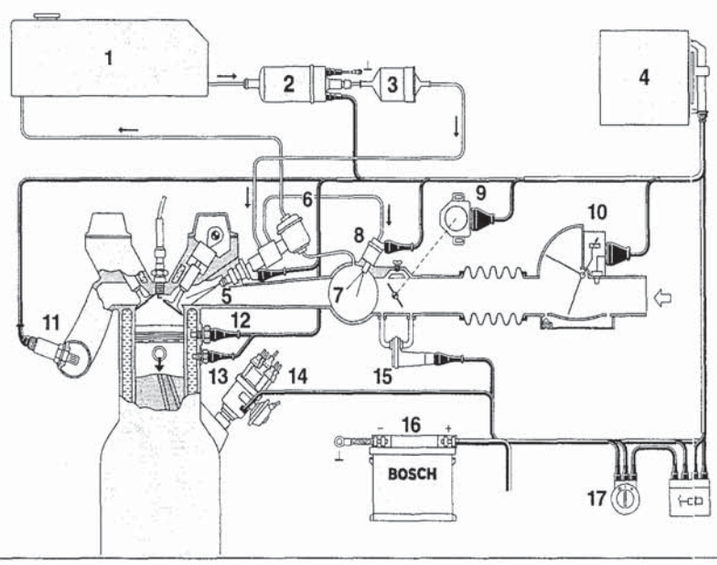
The fuel is injected through the engine's solenoid-operated injectors. A solenoid valve assigned to each cylinder is triggered once per crankshaft revolution. All of the injectors are wired in parallel to reduce the complexity of the electrical circuit. The pressure differential between fuel and intake-manifold pressures is maintained at a constant level of 2.5 or 3 bar such that the injected fuel quantity is only dependent on the opening period of the valves. For this purpose, the ECU delivers control pulses whose duration is dependent on the intake air flow, the engine speed, and other influencing variables. These are monitored by sensors and processed in the ECU.

Fuel supply

An electric fuel pump supplies the fuel and generates the injection pressure. The fuel is pumped from the fuel tank, through a paper filter, and into a high-pressure supply line at the other end of which there is a pressure regulator (spring-loaded diaphragm). The pressure regulator maintains a constant pressure at the metering orifice, regardless of the injected fuel quantity.

Schematic of an L-Jetronic system

1 Fuel tank, 2 Electric fuel pump, 3 Fuel filter, 4 ECU, 5 Injector, 6 Fuel-pressure regulator, 7 Intake manifold, 8 Electric start valve, 9 Throttle switch, 10 Air-flow sensor, 11 Lambda sensor, 12 Thermo-time switch, 13 Coolant-temperature sensor, 14 Ignition distributor, 15 Auxiliary-air valve, 16 Battery, 17 Ignition switch.



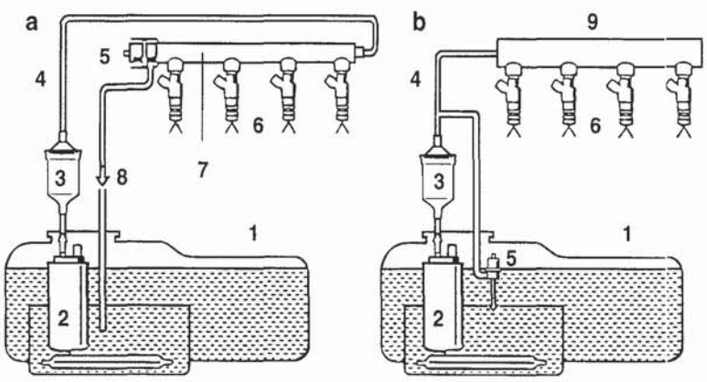
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Standard system:
The fuel flows through the high-pressure line on its way to the engine-mounted fuel rail with the injectors. The pressure regulator is installed on the fuel rail. After flowing through the rail the portion of the fuel not required by the engine flows through the return line attached to the regulator and back to the tank. Because the returning fuel has been warmed on the way back from the engine, fuel temperatures within the tank rise.

Fuel vapor is generated in the tank as a function of fuel temperature. For environmental purposes these vapors are routed through the tank ventilation system for storage in an activated charcoal canister until they can be returned through the intake manifold for combustion within the engine.

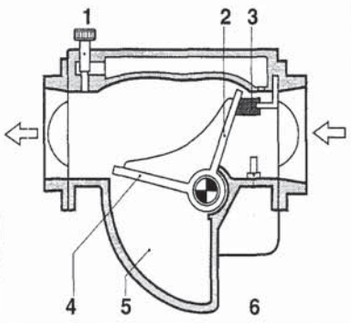
Returnless system:
A returnless fuel-supply system reduces the tendency of the fuel in the fuel tank to heat up, thus making it easier to comply with the legal requirements governing vehicular evaporative emissions. The pressure regulator is located either in the fuel tank or in its immediate vicinity, which means that the return line from the engine to the fuel tank can be dropped. The amount of fuel pumped to the fuel rail is limited to the quantity being used by the injectors. The excess flow volume emerging from the pump returns directly to the tank without the round trip to the engine and back. Assuming equivalent operating conditions, and depending upon the specific vehicular application, this system can reduce in-tank fuel temperatures by up to 10 K, cutting vaporization by roughly one third.

Fuel supply
a) Standard system, b) Returnless system.
1 Fuel tank, 2 Electric fuel pump, 3 Fuel filter, 4 Pressure line, 5 Fuel-pressure regulator, 6 Injectors, 7 Fuel rail (continuous flow), 8 Return line, 9 Fuel rail (no return flow).

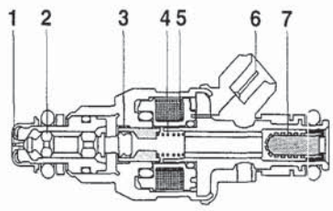


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Air-flow sensor
1 Idle-mixture adjustment screw, 2 Air-flow sensor flap, 3 Stop, 4 Compensation flap, 5 Damping chamber, 6 Air-temperature sensor.



Injector
1 Pintle, 2 Needle, 3 Armature, 4 Spring, 5 Solenoid winding, 6 Electrical terminals 7 Fuel strainer.



Air-flow sensor
The intake air flow deflects a sensor flap against the constant return force of a spring to a defined angular position, which is converted by a potentiometer into a voltage ratio. This voltage ratio determines the pulse length of a timing element in the ECU. A temperature sensor in the air-flow sensor indicates changes in air density caused by temperature variations.

Fuel injectors
Fuel injectors serve to meter and atomize the fuel. When the solenoid winding is energized, the nozzle needle is lifted a mere 0.05 mm off its seat.

Throttle-valve switch
This transmits a control signal to the ECU when the throttle valve is either completely closed (idle) or fully opened (full-throttle [WOT]).

Engine-temperature sensor
The engine-temperature sensor is designed as a temperature-sensitive resistor (thermistor) and controls the warm-up enrichment.

Auxiliary-air valve, electric start-valve, thermo-time switch
Design and function are similar to those of the corresponding K-Jetronic components.

Electronic control unit (ECU)
This ECU converts the engine variables into electrical pulses. Transmission intervals for these pulses are correlated with ignition timing, while their duration is basically a function of speed and intake air flow. Since all injectors are activated simultaneously, only a single driver stage is required. The temperature sensors respond to lower engine and air temperatures by increasing the injection duration. The throttle-switch signals allow mixture adaptation for idle and full-throttle operation.

Lambda closed-loop control
The ECU compares the signal from the Lambda sensor with a setpoint value before activating a two-state controller. The control adjustment is then performed, as are all corrections, by modifying the injection duration.

L3-Jetronic
L3-Jetronic incorporates functions extending beyond those provided by the L-Jetronic's analog technology. The L3 system's ECU employs digital technology to adjust the mixture ratios based on a load/engine-speed map. In order to save space, the ECU is installed in the engine compartment, directly on the air-flow sensor, where the two components form a single monitoring and control unit.

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LH-Jetronic

LH-Jetronic is closely related to L-Jetronic. The difference lies in the method of intake air-flow measurement, with LH-Jetronic using a hot-wire air-mass meter to measure the mass of the intake air. Thus, the results no longer depend on the air density, which varies with temperature and pressure.

The other LH-Jetronic components and the basic system concept are to a large extent the same as those in L-Jetronic.

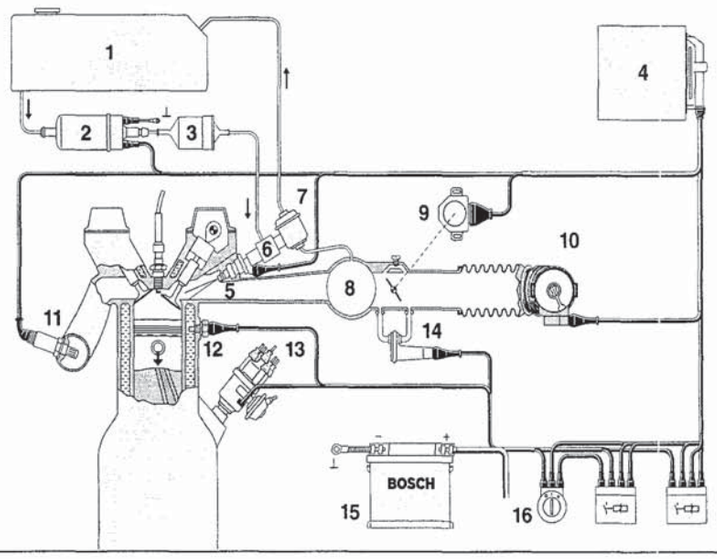
Operating-data processing in the ECU

LH-Jetronic is equipped with a digital ECU. Arrangements for adjusting the mixture ratio vary from those used with L-Jetronic in using a load/engine-speed map programmed for minimum fuel consumption and exhaust emissions. The ECU processes the sensor signals when calculating the injection duration that determines the injected fuel quantity. The ECU includes a microprocessor, a program and data memory, and an A/D converter. The microprocessor is provided with a suitable voltage supply and with a stable clock rate for data processing. The clock rate is defined by a quartz oscillator.

Hot-wire air-mass flow meter

The stream of intake air is conducted past a heated wire (hot wire). This wire forms part of an electrical bridge circuit. The flow of current through the wire serves to maintain it at a constant temperature above that of the intake air. This principle makes it possible to employ the current requirement as an index of the air mass being drawn into the engine. A resistor converts the heating current into a voltage signal, which the ECU then processes along with engine speed as a main input variable. A temperature sensor is mounted in the hot-wire air-mass flow meter to ensure that its output signal is not influenced by the temperature of the intake air. The A/F ratio at idle can be adjusted with a potentiometer. As contamination on

Schematic of an LH-Jetronic system
1 Fuel tank, 2 Electric fuel pump, 3 Fuel filter, 4 ECU, 5 Injector, 6 Fuel distributor, 7 Fuel-pressure regulator, 8 Intake manifold, 9 Throttle switch, 10 Hot-wire air-mass flow meter, 11 Lambda sensor, 12 Coolant-temperature sensor, 13 Ignition distributor, 14 Idle-speed actuator, 15 Battery, 16 Ignition switch.



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the surface of the hot wire could affect the output signal, each time the engine is shut down the wire is electrically heated for one second to burn-off any contamination. The hot-wire air-mass flow meter has no moving parts, and its aerodynamic resistance within the intake tract is negligible.

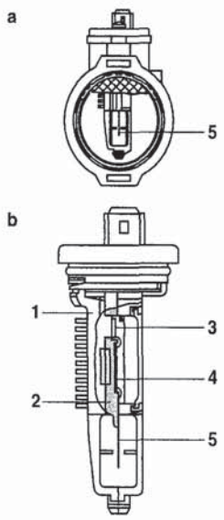
Hot-film air-mass flow meter

The operating principle of the hot-film air-mass flow meter is the same as that of the hot-wire sensor. However, in the interests of simplified design, a substantial portion of the electrical bridge circuit is installed on a ceramic substrate, in the form of thin-film resistors. In addition, there is no need to burn contaminants off the film. The contamination problem is solved by placing the areas on the sensor element which are decisive for thermal transmission at a downstream location. This prevents them from being affected by deposits on the sensor element's leading edge.

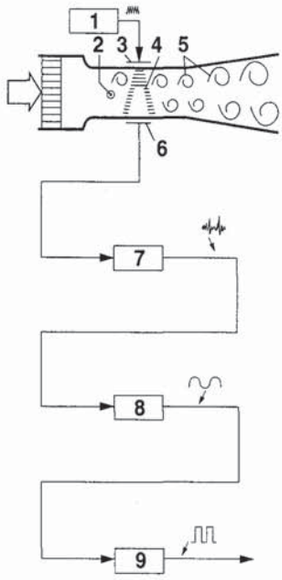
Kármán vortex volumetric flow meter

Yet another option for measuring intake air is provided by a sensor which uses the Kármán vortex principle to measure the volumetric flow rate. This meter monitors vortices generated as the intake air flows past vortex generators. The frequency of these vortices is a measure of the volumetric flow rate. This frequency is measured by emitting ultrasonic waves perpendicular to the direction of the intake-air flow. The propagation velocity of these waves as modified by the vortices is detected by an ultrasonic receiver and the resulting signals are evaluated in the ECU.

Hot-film air-mass flow meter
a Housing, b Hot-film sensor (installed in center of housing). 1 Heat sink, 2 Spacer, 3 Driver stage, 4 Hybrid, 5 Sensor element (metallic film).



Kármán vortex volumetric flow meter
1 Oscillator, 2 Vortex generator, 3 Transmitter, 4 Ultrasonic waves, 5 Eddy currents, 6 Receiver, 7 Amplifier, 8 Filter, 9 Pulse shaper.



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Electromagnetic fuel injectors

Design and operation

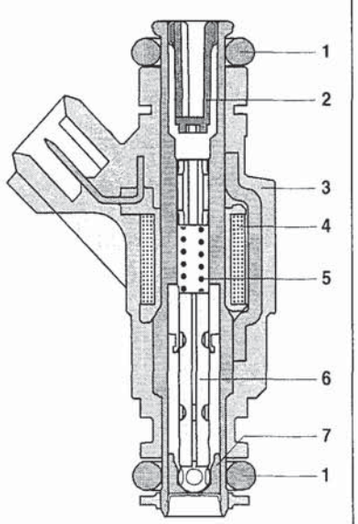
Fuel injectors essentially consist of a valve housing with current coil and electrical connection, a valve seat with spray-orifice disk and a moving valve needle with solenoid armature.

A filter strainer in the fuel feed protects the injector against contamination. Two O-rings seal the injector against the fuel-distribution pipe and the intake manifold. When the coil is de-energized, the spring and the force resulting from the fuel pressure press the valve needle against the valve seat to seal the fuel-supply system against the intake manifold (see Fig.).

When the injector is energized, the coil generates a magnetic field which attracts the armature and lifts the valve needle off of its seat to allow fuel to flow through the injector.

The injected fuel quantity per unit of time is essentially determined by the system pressure and the free cross-section of the spray orifices in the spray-orifice disk. The valve needle closes again when the field current is switched off.

Fuel injector EV6 *(example)*
1 O-rings, 2 Filter strainer, 3 Valve housing with electrical connection, 4 Current coil, 5 Spring, 6 Valve needle with solenoid armature, 7 Valve seat with spray-orifice disk.



Spray formation

The fuel injectors' spray formation, i.e. spray shape, spray angle and droplet size, influences the formation of the A/F mixture. Individual geometries of intake manifold and cylinder head make it necessary to have different types of spray formation.

Tapered spray:

Individual fuel sprays emerge through the openings in the spray-orifice disk. These fuel sprays combine to form a tapered spray. Tapered sprays can also be obtained by means of a pintle projecting through the injector needle tip. Tapered-spray injectors are typically used in engines with one intake valve per cylinder. The tapered spray is directed into the opening between the intake-valve disk and the intake-manifold wall.

Dual spray:

Dual-spray formation is used in engines with two intake valves per cylinder. The openings in the spray-orifice disk are arranged in such a way that two fuel sprays emerge from the injector. Each of these sprays supplies an intake valve.

Air-shrouding:

In the case of an air-shrouded injector, the pressure drop between intake-manifold and ambient pressures is used to improve mixture formation. Air is routed through an air-shrouding attachment into the outlet area of the spray-orifice disk. In the narrow air gap, the air is accelerated to a very high speed and the fuel is finely atomized when it mixes with it.

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Ignition

The ignition system's function is to initiate combustion in the flammable air-fuel mixture by igniting it at precisely the right moment. In the spark-ignition (Otto) engine, this is achieved with an electrical spark, i.e. an arc generated between the spark plug's electrodes. Consistently reliable ignition under all circumstances is essential for ensuring fault-free catalytic-converter operation.

Misfiring results in damage to or destruction of the catalytic converter due to overheating during afterburning of the uncombusted mixture.

Mixture ignition

Provided the composition of the mixture is stoichiometric, an energy of approximately 0.2 mJ is required for each individual ignition of the A/F mixture via electric spark. Over 3 mJ are required for a rich or lean mixture. This energy represents only a fraction of the total energy in the ignition spark, the actual ignition energy. If sufficient ignition energy is not available, there will be no ignition, the mixture cannot ignite, and misfiring will result.

The system must therefore deliver enough ignition energy to ensure consistently reliable ignition of the mixture, even under unfavorable conditions. Igniting a small flammable mixture cloud flowing past the spark can be enough to initiate the process.

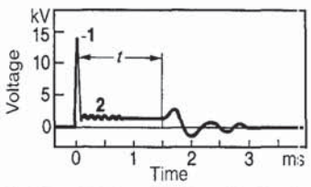
This mixture cloud ignites, the flame spreads to the remaining mixture in the cylinder, and the fuel starts to combust. Ignitability is enhanced by efficient fuel atomization and good access of the mixture to the electrodes, as well as through extended spark duration and spark length (large electrode gap).

The spark plug determines the location and length of the spark; spark duration depends upon the type and design of the ignition system, as well as on the momentary ignition conditions.

Spark generation

Adequate voltage must be present before a spark will arc from one electrode to another. At the moment of ignition, the voltage across the electrodes abruptly rises from zero up to the arcing (ignition) voltage and the plug fires. Once the spark has ignited, the spark-plug voltage drops to the sparking voltage. The A/F mixture may ignite at any point during the firing period of the ignition spark (spark duration). Once the spark has broken away, the voltage is damped and drops to zero.

Spark-plug voltage characteristic with stationary or semi-stationary A/F mixture
1 Ignition voltage, 2 Spark voltage, t Spark duration.



Although intense mixture turbulence is basically desirable, it can extinguish the spark, thus leading to incomplete combustion. The energy stored in the ignition coil should therefore suffice for one or more consecutive sparks, depending on individual requirements.

High-voltage generation and energy storage

Battery-ignition systems generally employ an ignition coil to generate the high-tension voltage needed to generate the spark. The ignition coil operates as an autotransformer but within coil-ignition systems it also assumes the further important function of storing the ignition energy. When the contact breaker closes, energy from the vehicle's electrical system flows into the coil's primary winding. This energy is then stored in a magnetic field until the firing point, when the secondary winding discharges it to one of the engine's spark plugs. The ignition coil is designed to ensure that the available high-tension voltage in the coil is always well in excess of the spark plug's maximum-possible ignition-voltage requirement. Energy levels of 60...120 mJ within the coil correspond to an available voltage of 25...30 kV.

The operational reserves of high volt-

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age and ignition energy are sufficient to compensate for all electrical losses. Inadequate maintenance reduces these high-voltage reserves, and leads to ignition and combustion miss. Engine power drops and fuel consumption increases. In addition, this phenomenon can result in damage to or destruction of the catalytic converter, should one be installed. In extreme cases, the engine either fails to start – especially when cold – or stalls.

Ignition systems are also available with capacitive energy storage (CDI or Capacitor-Discharge Ignition) for use on high-performance and racing engines. These systems store the ignition energy in the electrical field of a capacitor before a special transformer transmits it to the spark plug in the form of a high-voltage ignition pulse.

Ignition timing and adjustment

Approximately two milliseconds elapse between the mixture's initial ignition and its complete combustion. The ignition spark must therefore arc early enough to ensure that main combustion, and thus the combustion-pressure peak in the cylinder, occur shortly after piston TDC. The ignition angle should therefore move further in the advance direction along with increasing engine speed. The chosen firing point should ensure that the following requirements are met:

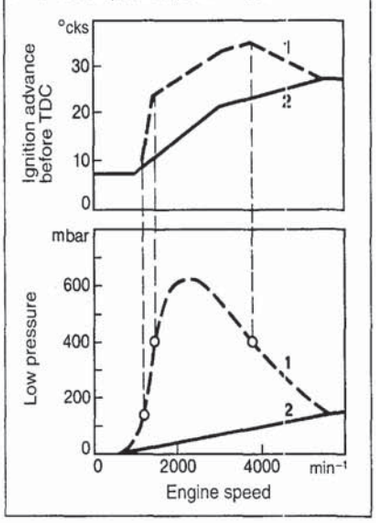
– maximum engine performance
– low fuel consumption
– no engine knock
– clean exhaust gas.

Since it is impossible to obtain optimal compliance with all of these requirements simultaneously; compromises must be found on a case-to-case basis. Optimal ignition timing is defined according to a variety of parameters. The most important are engine speed, engine load, engine design, fuel quality and momentary operating conditions (starting, idle and trailing throttle, etc.). In the simplest case, spark-advance mechanisms sensitive to variations in engine speed and intake-manifold vacuum adapt the ignition timing to suit the engine's current operating conditions.

In modern engine-management systems with extended functions, additional adjustments can be used e.g. for rapid torque adaptation or for swift heating of the catalytic converter.

Example of cumulative ignition timing consisting of centrifugal and vacuum advance

1 Part-load operation, 2 Full load.

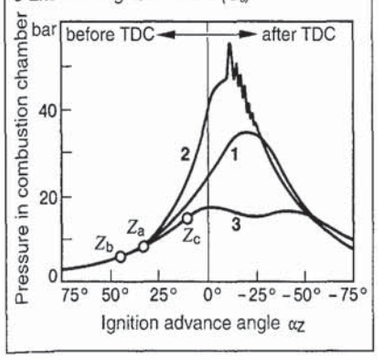


All the adjustment strategies can operate either individually or simultaneously. The degree to which the ignition timing is advanced or retarded is determined by the ignition-advance curves calibrated specifically for each individual engine configuration.

At full load, the accelerator pedal is depressed fully and the throttle is wide open (WOT). Along with increasing engine speeds, ignition takes place earlier in order to maintain the combustion pressure at the levels required for optimal engine performance. The leaner A/F mixtures encountered during part-throttle operation are more difficult to ignite. Because this means that more time is required for ignition, it must be triggered earlier, with the timing being shifted further in the "advance" direction. The manifold vacuum employed to determine the necessary degree of spark advance is monitored downstream from the throttle valve. If the vacuum bore is located near the throttle valve (P. 489), the vacuum initially increases as the throttle is opened wider and begins to

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Combustion-pressure curve for various ignition firing points
1 Correct ignition advance (Z_a),
2 Excessive ignition advance (Z_b),
3 Excessive ignition retard (Z_c).



fall in the proximity of the full-throttle (WOT) position. The progressively wider throttle openings required to increase engine speed on the operating curve for part-throttle road operation are reflected in the relationship between vacuum and min^{-1} shown in the diagram.

Yet another diagram shows the curves for combustion-chamber pressure in a 4-stroke engine with correct and incorrect ignition timing. Even if the timing is initially correct, neglected maintenance can allow it to drift over the course of time. If the timing shifts towards a later firing point ("retard"), the result is a gradual drop in engine power and increased fuel consumption. Excessive "advance" may result in extreme cases in serious damage to spark plugs or to the engine if the engine knocks. The level of exhaust emissions also increases.

Ignition and emissions

Owing to the fact that it directly affects the various exhaust-gas components, the ignition has a significant effect upon exhaust emissions. Because various – and in this context sometimes mutually antagonistic – factors such as fuel economy, driveability, etc., are also potential optimization criteria, it is not always possible to specify the ideal ignition timing for minimum emissions.

Shifts in ignition timing induce mutually inverse response patterns in fuel consumption and exhaust emissions: While more spark advance increases power and reduces fuel consumption, it also raises HC and, in particular, NO_x emissions. Excessive spark advance can cause engine knock and lead to engine damage. Retarded ignition results in higher exhaust-gas temperatures, which can also harm the engine. Electronic engine-management systems featuring programmed ignition curves are designed to adapt ignition timing in response to variations in factors such as min^{-1}, load, temperature, etc. They can thus be employed to achieve the optimum compromise between these mutually antagonistic objectives.

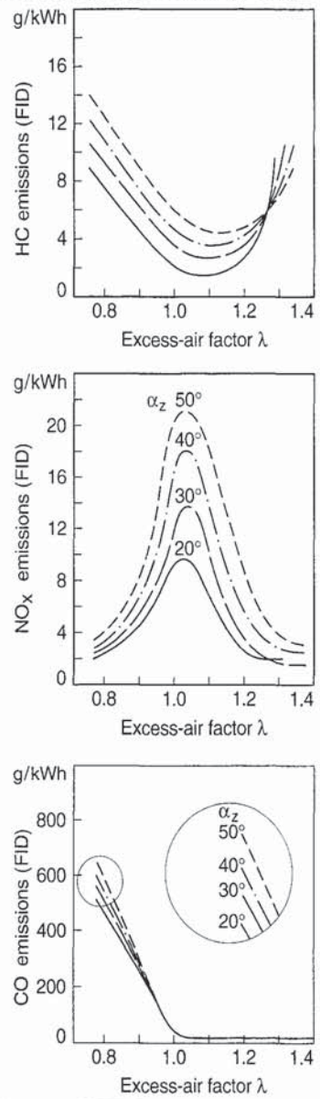
Overview of various ignition systems

Function	Ignition system			
	CI	TI	EI[1]	DLI
Designation	Coil ignition system	Transistorized ignition system	Engine management: Electronic ignition system	Engine management: Distributorless semiconductor ignition system
High-voltage generation	Inductive			
Ignition triggering	Mechanical	Electronic		
Ignition angle determined from engine speed and load	Mechanical		Electronic	
Spark distribution to appropriate cylinder	Mechanical			Electronic

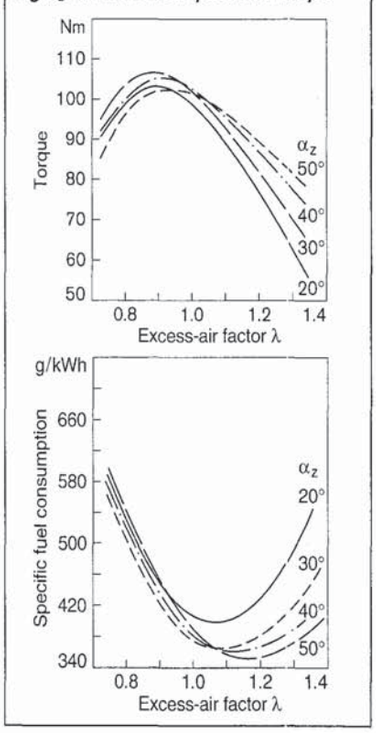
[1]) Any desired ignition timing is not possible with EI. DLI therefore dominates with integrated engine-management systems.

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Influence of excess-air factor λ and ignition point α_z on pollutant emissions



Influence of air factor λ and ignition timing α_z on fuel consumption and torque



Ignition energy

The ignition system generates a high-voltage spark at the spark plug to initiate combustion. An ignition-spark energy of approx. 0.2 mJ is adequate to ignite a stoichiometric air-fuel mixture, while richer or leaner mixtures require substantially higher levels of spark energy. Excess energy, i.e., from an ignition system designed to generate a high-energy spark of extended duration (transistorized or electronic ignition) stabilizes flame propagation and reduces the fluctuations from cycle to cycle. The reduction in fluctuations results in smoother engine operation and lower HC emissions. Increased spark projection, larger electrode gaps and thin electrodes also have a positive influence on the engine's smoothness and HC emissions.

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Ignition coil

The ignition coil functions as both an energy-storage device and a transformer. The coil, which is powered by DC voltage from the vehicle's electrical system, supplies the ignition pulses for the spark plugs at the required high voltage and discharge energy. The ignition driver stage with its defined deactivation current combines with a primary winding featuring specific resistance and inductance characteristics to determine the amount of energy stored within the ignition coil's magnetic field. The secondary winding can be designed to provide peak voltage, spark current and discharge duration in accordance with individual requirements.

The contact-breaker points used with coil ignition (CI) can only handle interrupt currents of up to approx. 5 A. TI, EI and DLI ignition systems and Motronic ECUs can handle much higher interrupt currents. The series resistors generally employed with coil ignition (they can be bypassed to increase energy during cold starts) can be omitted in electronic ignition systems. Here the electronic circuitry activates the ignition coil depending on battery voltage, engine speed and other influencing variables in such good time that full energy is available at the ignition point.

Each ignition coil is designed to meet the requirements of a particular application. It must charge quickly in order to furnish the voltages and ignition energies required at high engine speeds. Important priorities thus include low primary inductance and, in some cases, higher primary interrupt currents (for adequate energy storage).

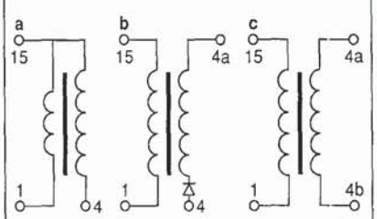
Design and operation

Traditional ignition coils with asphalt or oil insulation enclosed in metal casings are being increasingly replaced by units featuring an epoxy-resin filler. These not only allow more latitude in the selection of geometry, type and number of electrical terminals, but also provide more compact dimensions, better vibration resistance and lower weight. The ignition coil is generally attached by way of the iron core to the engine or vehicle body. Rod-type ignition coils are installed in the cylinder-head recess above the spark plug.

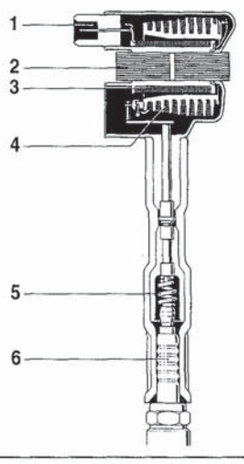
The coil's synthetic materials provide good adhesion between all of the high-voltage components and the molded epoxy resin, which penetrates into all the capillary spaces. Supplementary iron cores are sometimes embedded on the inside of the synthetic molding.

The secondary winding is mostly designed as a disk or sandwich coil, with the

Ignition coils (schematic)
Rotating distribution: a) Single-spark ignition coil. Distributorless ignition: b) Single-spark ignition coil, c) Dual-spark ignition coil.



Single-spark ignition coil
1 External low-voltage terminal, 2 Laminated iron core, 3 Primary winding 4 Secondary winding, 5 Internal high-voltage connection via spring contact, 6 Spark plug.



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windings distributed among a series of segments. Even distribution of stresses among the insulating elements in all chambers combines with high dielectric strength to permit compact dimensions while at the same time making foil and paper between wire layers redundant. The winding's self-capacitance is also reduced.

Because lower breakdown voltages are required for the negative (relative to engine ground) ignition spark, the positive terminals for the primary and secondary windings are generally combined on those ignition coils used with rotating high-voltage distribution.

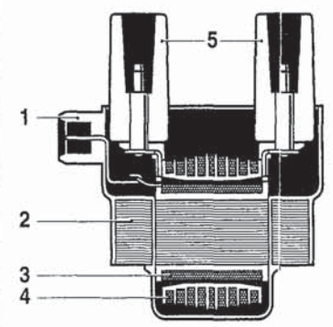
Single and dual-spark ignition coils are an alternative for use in ignition systems with distributorless ignition (DLI).

When a single-spark coil per spark plug is used, the primary current is controlled to furnish the relevant spark plug with an ignition pulse at precisely the right moment in time. High-voltage diodes are used to prevent the positive 1...2 kV high-voltage pulse generated when the primary current is activated from causing the spark plug to fire prematurely.

On the dual-spark coil, the secondary winding is galvanically insulated from the primary winding. Each of the two high-voltage outputs are connected to a spark plug. Ignition sparks are created at the two spark plugs when the primary current is deactivated. As with rotating high-voltage distribution, this system does not usually require any special precautions to prevent activation sparks.

Connection and installation are facilitated by combining several ignition coils in a common casing to form a single assembly. However, the individual coils continue to operate as independent units. The integration of output stages in the ignition coils means that short primary leads can be used (lower voltage drop). This arrangement also prevents power loss in the driver circuits from overheating the ECU.

Dual-spark ignition coil (distributorless ignition)
1 Low-voltage terminal, 2 Laminated iron core, 3 Primary winding, 4 Secondary winding, 5 High-voltage terminals.



Spark plug

Function

The spark plug introduces the ignition energy generated by the ignition coil into the combustion chamber. The high voltage creates an electric spark between the spark-plug electrodes which ignites the compressed A/F mixture. As this function must also be guaranteed under extreme conditions (cold starting, full load), the spark plug plays a decisive role in the optimum performance and reliable operation of a spark-ignition engine. These requirements remain the same over the entire service life of the spark plug.

Requirements

The spark plug must satisfy a variety of extreme performance demands: It is exposed to the varying periodic processes within the combustion chamber as well as external climatic conditions. However, the combustion chamber must remain sealed.

During spark-plug operation with electronic ignition systems, ignition voltages of up to 30,000 V may occur and must not damage the insulator. This insulation capability must also be guaranteed at temperatures in the region of 1000 °C.

Because the spark plug is subjected to mechanical stresses in the form of exposure to periodic pressure peaks (up to

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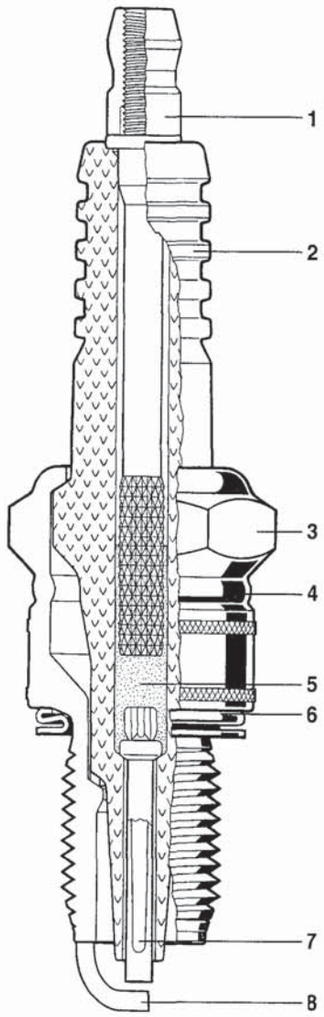
80 bar) within the combustion chamber, its materials must exhibit extreme resistance to thermal loads and continuous vibratory stress. At the same time, that section of the spark plug that protrudes into the combustion chamber is exposed to high-temperature chemical processes, making resistance to aggressive combustion deposits essential. Because it is subjected to rapid variations between the heat of the combustion gases and the cool A/F mixture, the spark-plug insulator must feature high resistance to thermal stresses (thermal shock). Effective heat dissipation at the electrodes and the insulator is also essential for reliable spark-plug performance.

Design

In a special high-grade ceramic insulator, an electrically conductive glass seal forms the connection between the center electrode and terminal stud. This glass element acts as a mechanical support for the components while providing a gas seal against the high-pressure combustion gases. It can also incorporate resistor elements for interference suppression and burn-off.

The connection end of the insulator is glazed for improved protection against contamination. The connection between it and the nickel-plated steel shell is gas-tight. The ground electrode, like the center electrode, is primarily manufactured using nickel-based alloys to cope with the high thermal stresses. It is welded to the shell. The thermal conduction properties of both the center and the ground electrodes are improved by using a nickel-alloy jacket material and a copper core. Silver and platinum, or platinum alloys, are employed as electrode material for special applications. The spark plugs have either an M4 or a standard SAE thread, depending upon the type of high-voltage connection. Spark plugs with metal shields are available for watertight systems and for maximum interference suppression.

Spark plug
1 High-voltage connector (terminal nut),
2 Al_2O_3 Ceramic insulator,
3 Shell,
4 Heat-shrinkage zone,
5 Conductive glass,
6 Captive gasket,
7 Composite center electrode Ni/Cu,
8 Ground electrode.



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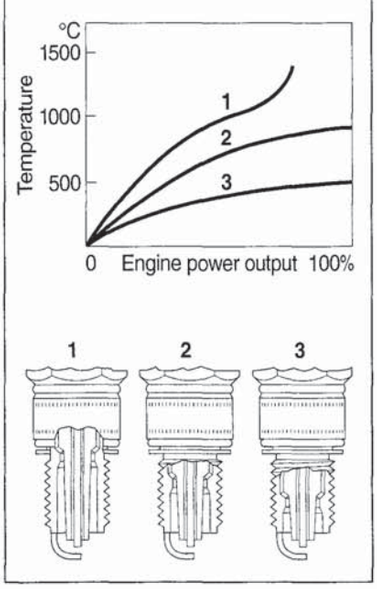
Heat range

The operating temperature of the spark plug represents a balance between heat absorption and dissipation. The aim is to achieve a self-cleaning temperature of approx. 500 °C even at low engine power outputs.

If the temperature drops below this level, there is the danger that unburnt hydrocarbons and oil residue from incomplete combustion sequences will settle on the cold areas of the spark plugs (particularly when the engine is not at normal operating temperature, at low outside temperatures, and during repeated starts). This can create a conductive connection (shunt) between the center electrode and the spark-plug shell by way of which the ignition energy leaks away in the form of short-circuit current (risk of misfires). At higher temperatures, the residues containing carbons burn on the insulator nose; the spark plug thus "cleans" itself.

An upper temperature limit of approx. 900 °C should be observed because in this range wear of the spark-plug electrodes increases markedly (due to hot-gas corrosion) and if this limit is significantly exceeded this increases the risk of auto-ignition (ignition of the air-fuel mixture on hot surfaces). Such auto-ignition subjects the engine to extreme loads and can result in the engine's destruction within a short period of time. The spark plug must therefore be adapted accordingly in terms of its heat-absorption capability to the engine type. The identifying feature of a spark plug's thermal loading capacity is its heat range, which is defined by a code number and determined in comparison measurements with a reference standard source.

Spark-plug temperature response
1 high, 2 medium, and 3 low heat range.



The Bosch ionic-current process presents possibilities for adapting the heat range to each engine. Characteristic changes in the combustion procedure due to increased thermal loading of the spark plugs can be detected using ionic current and used in the assessment of the auto-ignition process. The spark plug must be adapted in such a way as to preclude any possibility of premature ignition.

The use of center-electrode materials with high thermal conductivity (silver or nickel alloys with copper core) makes it possible to substantially extend the insulator nose without changing the plug's heat range, thus extending the plug's operating range downward into a lower thermal-load range and reducing the probability of fouling. These advantages are inherent in all Bosch Super (thermo-elastic) spark plugs.

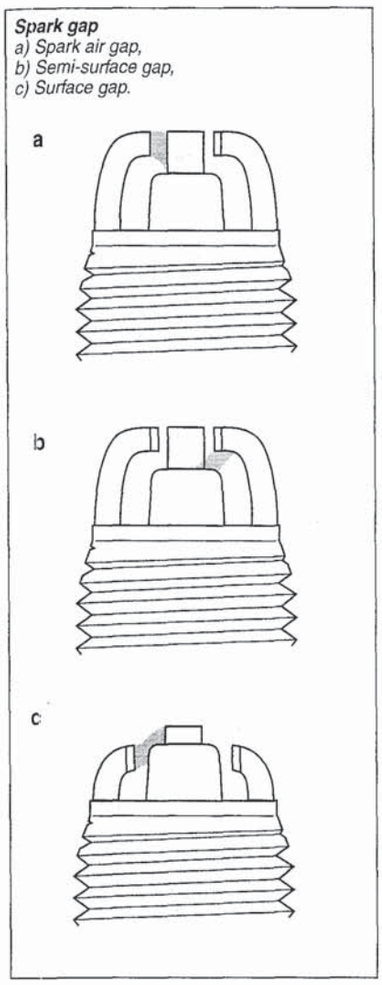
Reducing the likelihood of combustion miss and ignition miss – with their attendant massive increases in hydrocarbon emissions – provides benefits in exhaust emissions and fuel consumption in part-throttle operation at low load factors.

Electrode gap and ignition voltage

The electrode gap should on the one hand be as large as possible so that the ignition spark activates a large volume element and thus results in reliable ignition of the air-fuel mixture due to the development of a stable flame-core.

It is frequently impossible to achieve a smooth idle when the electrode gap is too

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Spark gap
a) Spark air gap,
b) Semi-surface gap,
c) Surface gap.

narrow. On the other hand, the electrode gap must be narrow enough to guarantee that the ignition voltage will continue to produce a reliable arc, even at the end of the spark plug's service life and under unfavorable circumstances.

The required ignition voltage is influenced not only by the size of the electrode gap, but also by the electrodes' shape and temperature and the materials used in their manufacture. Parameters specific to the combustion chamber such as mixture composition (lambda value), flow velocity, turbulence and density of the gas to be ignited, also play an important role.

On today's high-compression engines, which frequently feature high charge turbulence, electrode gaps must be carefully defined in order to guarantee reliable ignition and thus misfire-free operation throughout the required service life.

Spark gap
The configuration of the arcing path s defined by the mutual arrangement of the electrodes.

Spark air gap (a):
A linear spark arcing directly between the center and ground electrodes ignites the air/fuel mixture between the electrodes.

Semi-surface gap (b):
The semi-surface gap is created by positioning the ground electrodes (at a defined distance to the insulator end face) to the sides of the center electrode. Under certain conditions, the ignition spark forms between the center electrode and the surface of the insulator tip before arcing across a gas-filled gap to the ground electrode.

Shifting a portion of the spark's propagation travel to this shunt path extends the gap across which any given voltage can produce an arc.

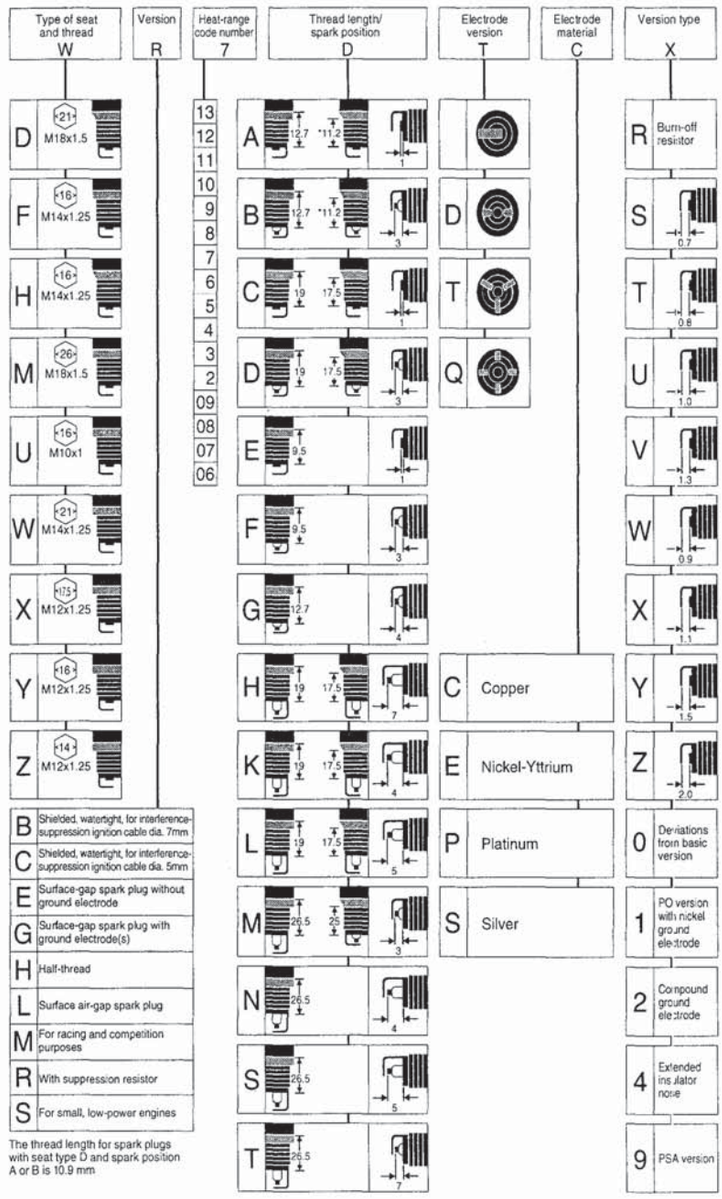
The increased electrode gap improves the ignition properties.

Surface gap (c):
On surface-gap spark plugs, the ground electrodes are positioned to the sides of the ceramic body. The sparks thus form over the surface of the insulator tip before arcing across a small gas-filled gap to the ground electrode. The arrangement of the ground electrode to the side of the ceramic body helps to reduce the quenching losses, i.e. the flame core can extend more effectively into the combustion chamber, thus enhancing the ignition properties of the plug.

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Designation codes for Bosch spark plugs

Individual spark-plug specifications are contained in the designation code. This code includes all vital spark-plug characteristics except the electrode gap, which is indicated on the package. Spark-plug specifications for individual engine applications are defined by Bosch and the engine manufacturers.



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Conventional coil ignition (CI)

Many vehicles are still equipped with conventional coil ignition. When the contact breaker closes with the ignition switched on, current from the battery or alternator flows through the ignition coil's primary winding, generating a powerful magnetic field in which the energy is stored. At the ignition point, the contact breaker interrupts the current, the magnetic field collapses and the high voltage necessary for ignition is induced in the secondary winding. This voltage is fed from terminal 4 to the ignition distributor via a high-tension cable and from there to the individual spark plugs.

The following is a basic definition of the relationship between the speed of a four-stroke SI engine and the number of sparks generated per minute:

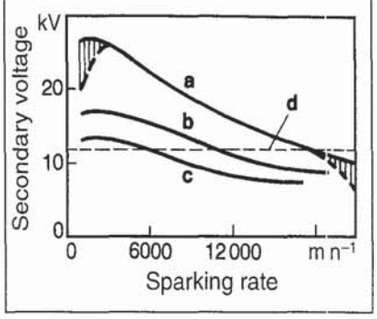
$$f = z \cdot n/2$$

f Spark-generation rate, z Number of cylinders, n Engine speed.

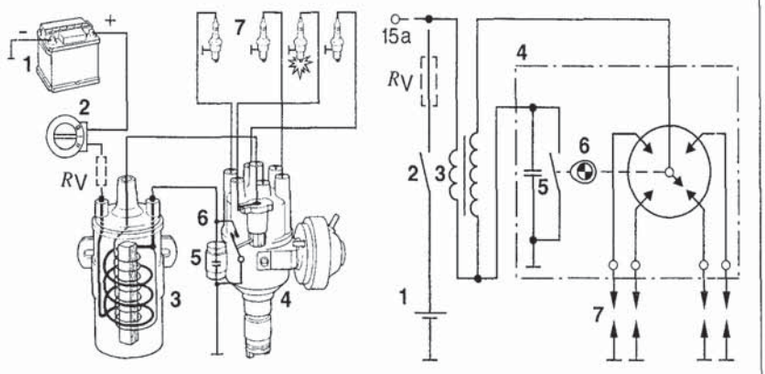
At low engine speeds, the contact-breaker points remain closed long enough to exploit the coil's full energy-storage potential. At higher engine speeds, this contact period – the dwell angle – is shorter, and the primary current is interrupted before maximum energy can be transferred to the coil. The resulting reduction in stored energy means that less high-tension current is then available from the coil.

In response, ignition coils are designed to provide high-tension voltage well in excess of the spark plugs' requirements, even at maximum engine speeds. Contamination on the insulating components acts as a capacitive and ohmic shunt, increasing the ignition loads placed upon the system, with combustion and ignition misfiring as the ultimate consequences.

Secondary voltage as a function of sparking rate
a *Without ohmic shunts (R > 10 MΩ),*
b *Shunt resistance 1 MΩ,*
c *Shunt resistance 0.5 MΩ,*
d *Required ignition voltage.*



Conventional coil-ignition system (CI), *components at left, circuit diagram at right*
1 Battery, 2 Ignition switch, 3 Coil, 4 Distributor, 5 Ignition condenser, 6 Contact breaker, 7 Spark plugs, R_V Ballast resistor for increased start voltage (optional).



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Ignition coil
Description: see P. 481.

Ignition distributor
The distributor is a separate, self-contained component within the ignition system. It has the following functions:
- it distributes the ignition pulses to the engine's spark plugs in the defined sequence (CI, TI, and electronic ignition),
- triggers the ignition pulse, either when the contact breaker interrupts the primary current, or, with breakerless systems (CI, TI, EI in some cases), using a pulse generator,
- adjusts the ignition timing with a spark-advance mechanism on conventional ignition systems (CI, TI).

In modern electronic ignition systems, operating either alone or in combination with the fuel-injection system (Motronic), the distributor generally comprises only a rotor arm connected to the camshaft and the distributor cap with high-voltage cables.

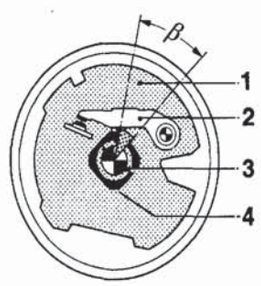
The contact-breaker points and the spark-advance mechanism perform separate functions from those of the distributor proper. They are combined with it in a single unit because they require a synchronized drive.

The ignition pulse passes through the center connection and the carbon brush or the center-tower spark gap to the distributor's rotor arm which then distributes this ignition energy by arcing it to fixed electrodes pressed into the periphery of the distributor cap. From here, the ignition pulses travel through the ignition cables to the spark plugs. A dust cover is sometimes installed to separate this high-voltage section from the rest of the unit.

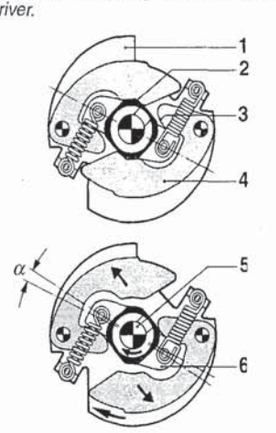
Contact breaker
A cam opens the contact-breaker points to interrupt the flow of primary current to the coil for ignition. The number of cam lobes corresponds to the number of engine cylinders. The portion of the distributor shaft's rotation during which the points remain closed is the dwell angle.

The contact-breaker points are subject to three types of wear:

Contact breaker
1 Moving breaker-plate assembly, 2 Breaker lever, 3 Distributor shaft, 4 Distributor cam.



Centrifugal advance mechanism, at rest (above), in operation (below)
1 Support plate, 2 Distributor cam, 3 Contact path, 4 Advance flyweight, 5 Distributor shaft, 6 Driver.



- contact pitting,
- contact arm (rubbing-block) wear,
- plastic deformation and local compression of the contact metal.

Contact pitting stems from the breaking sparks (residual arcing) induced by induction voltage during interruption of the primary current. The ignition condenser is designed to suppress this type of arcing, but residual sparks continue to occur. Although contact wear and rubbing-block wear are mutually counteractive, the effects of the latter are generally more pronounced, resulting in a tendency for the

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ignition to drift in the "retard" direction, toward a later ignition point.

Spark-advance mechanism
Ignition distributors are generally equipped with two spark-advance mechanisms: a speed-sensitive centrifugal advance mechanism and a load-dependent vacuum-controlled device.

Centrifugal advance mechanism
The centrifugal advance mechanism adjusts the ignition timing in response to changes in engine speed. The support plate upon which the flyweights are mounted rotates with the distributor shaft. The flyweights move outward as engine speed increases, thereby turning the driver over the contact path to the distributor shaft in the direction of rotation. In this way, the distributor cam also turns towards the distributor shaft by the ignition advance angle α. The point of ignition is advanced by this angle.

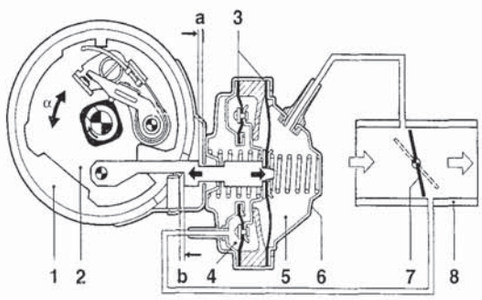
Vacuum adjustment mechanism
The vacuum mechanism adapts the ignition timing to changes in engine output and load factor. Intake manifold vacuum is monitored or tapped off in the vicinity of the throttle valve. The vacuum acts upon two aneroid capsules.

Operation of advance mechanism
Because the air/fuel mixture combusts more slowly during operation at low load factors, it must be ignited earlier to compensate. Meanwhile, the proportion of those residual gases which have been burned but not discharged from the combustion chamber increases, and the mixture leans out. Vacuum for the advance mechanism is tapped off immediately downstream from the open throttle valve. As the engine load decreases, the vacuum in the advance unit rises, causing the diaphragm and its control arm to move to the right. The control arm turns the breaker-plate assembly against the distributor shaft's direction of rotation; the point of ignition is advanced still further.

Operation of retard mechanism
Here the connection with the intake manifold's internal vacuum is downstream from the closed throttle. The ring-shaped vacuum retard unit reduces exhaust emissions by reducing ignition advance under specific operating conditions (e.g. idle, trailing throttle). The ring diaphragm and its control arm move to the left when vacuum is applied. The control arm rotates the breaker-plate assembly together with the contact breaker in the distributor shaft's direction of rotation.

This spark-retard system operates independently of the advance mechanism. The advance mechanism has priority: simultaneous vacuum in both units during part-throttle operation shifts the unit to its "advance" position.

Vacuum advance mechanism with ignition advance and retard units
a Advance adjustment up to stop, b Retard adjustment up to stop
1 Ignition distributor, 2 Breaker-plate assembly, 3 Diaphragm, 4 Vacuum retard unit, 5 Vacuum advance unit, 6 Vacuum unit, 7 Throttle valve, 8 Intake manifold.



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Transistorized ignition (TI)

With conventional coil-ignition systems, ignition energy and maximum voltage are restricted by various electrical and mechanical factors limiting the breaker points' switching capacity. The demands placed upon battery-ignition systems are often more than the contact-breaker assembly can satisfy in its role as a power switch. In electronic ignition systems, the points are assisted or replaced entirely by wear-free control devices. Transistorized (coil) ignition is available in both breaker-triggered and breakerless versions.

Transistorized coil ignition with contact control is especially suitable for upgrading existing coil-ignition systems (CI). Breaker-triggered transistorized coil ignition systems are no longer installed as original equipment.

Breakerless transistorized ignition

On breakerless transistorized ignition systems, the cam-actuated contact breaker is replaced by a magnetic "pulse generator". This generates current and voltage pulses magnetically (without contacts) to trigger the high-voltage ignition pulse through the system electronics. The pulse generator is installed in the ignition distributor.

These triggering devices operate according to various principles.

Induction-type pulse generators (TI-I)

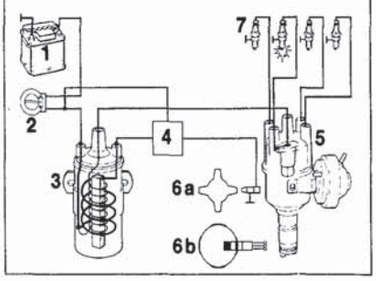
The induction-type pulse generator is a permanently-excited AC generator consisting of stator and rotor. The number of teeth or arms corresponds to the number of cylinders in the engine. The frequency and amplitude of the alternating current generated by the unit vary according to engine speed. The ECU processes this AC voltage and uses it for ignition control.

Hall-effect pulse generators (TI-H)

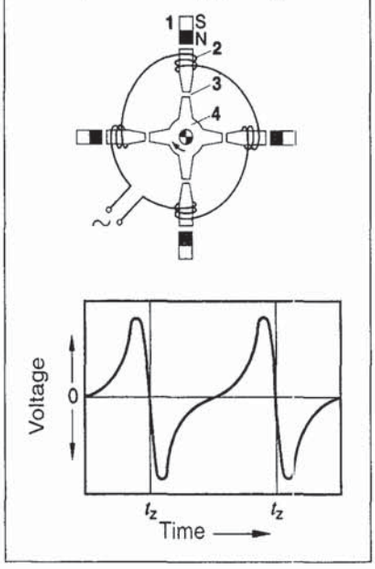
This type of ignition-pulse generator utilizes the Hall effect. A speed-sensitive magnetic field produces voltage pulses in an electrically charged semiconductor layer to control activation of the ECU's primary current.

Ignition pulse generators (impulsers) display clear benefits over mechanical contact breakers: They do not wear, and are thus maintenance-free. They allow precise control of ignition timing with attendant benefits in engine performance.

Breakerless transistorized ignition system
1 Battery, 2 Ignition switch, 3 Coil, 4 Electronic trigger box, 5 Ignition distributor with centrifugal and vacuum advance mechanism, 6a Induction-type pulse generator, 6b Hall-type pulse generator (alternative), 7 Spark plugs.



Ignition distributor with induction-type pulse generator
1 Permanent magnet, 2 Induction winding with core, 3 Variable air gap, 4 Trigger wheel.



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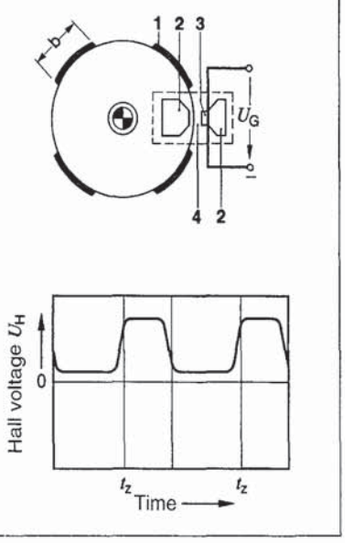
Electronic control units

Virtually all of the electronic control units (trigger boxes) in use today are equipped with primary-current regulators and closed-loop dwell-angle control.

The primary-current regulator limits the current in order to protect the ignition coil and the driver stage. When used in conjunction with a coil featuring low primary resistance, it provides high starting current at low battery voltages. This makes it possible to dispense with series resistors upstream of the coil as well as with the bridging function for starting.

Closed-loop dwell-angle control ensures that the desired primary current is obtained in the control range as far as possible at the point of ignition. This reduces the power losses in the ECU. It also compensates for battery-voltage fluctuations and ignition-coil temperature effects. Depending on system design, this dwell-angle control is effective up to medium engine speeds. At high engine speeds, the dwell angle is determined by the break time required to achieve adequate arcing durations. The residual energy remaining in the coil after the break time promotes optimal coil charging with reduced dwell times.

Ignition distributor with Hall sensor
1 Vane with width b, 2 Soft-magnetic conductive elements, 3 Hall IC, 4 Air gap, U_G Sensor voltage (transformed Hall voltage).



The sparkless closed-circuit current deactivation switches off the primary current with the ignition on and the engine off to ensure that no sparks occur at the spark plug. However, there are also TC-I systems (with induction-type pulse-generator) with intrinsic closed-circuit current deactivation.

Transistorized ignition is sometimes employed together with auxiliary devices to adjust the spark advance. An example would be the idle-speed control which is installed between the Hall generator and the ECU; below idle speed it reacts to further decreases in engine min^{-1} by advancing the ignition, thus increasing torque and preventing engine speed from dropping any further. The electronic retard device reduces ignition advance at high engine speeds to prevent knocking. It is connected in parallel with the ECU. Today, both of these functions are integrated in the electronically adjusted ignition systems within the engine-management system.

Hybrid units have become the ECU standard for transistorized ignition systems owing to their ability to combine high packaging density with low weight and excellent reliability. Hybrid technology replaces the printed-circuit board, with an Al_2O_3 substrate bearing conductor paths and resistors applied in a silk-screening process. Semiconductor devices and capacitors in chip form complete the circuit. As the Darlington power-transistor chip is mounted insulated on the metallic base plate, cooling is excellent, permitting operation at high temperatures.

Ignition coils (description on P. 481)

The performance specifications of ignition coils for conventional ignition differ from those of ignition coils with electronic circuit-breakers (transistorized ignition, see P. 490). A coil designed for one application should never be employed in the other.

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Note: Unlike with breaker-triggered ignition, terminal 1 of the systems mentioned must not be shorted to ground (e.g. during compression testing) as this would overload the low-resistance primary winding of the ignition coil.

Capacitor-discharge ignition (CDI)

The operating concept behind CDI, or "thyristor ignition," as it is also called, differs from that of the ignition systems described above. CDI was developed for use with high-speed, high-output multicylinder reciprocating IC engines in high-performance and competition applications and for rotary-piston engines.

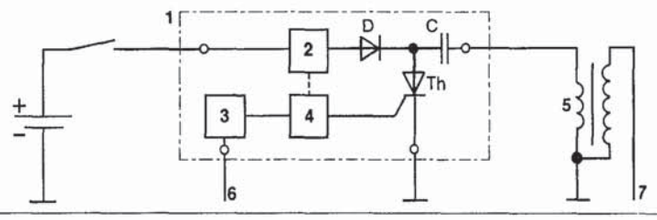
The salient characteristic of the CDI system is that it stores ignition energy in the electrical field of a capacitor. Capacitance and charge voltage of the capacitor determine the amount of energy which is stored. The ignition transformer converts the primary voltage discharged from the capacitor to the required high voltage. Capacitor-discharge ignition is available in both breaker-triggered and breakerless versions.

The major advantage of the CDI is that it generally remains impervious to electrical shunts in the high-voltage ignition circuit, especially those stemming from spark-plug contamination. For many applications, the spark duration of 0.1...0.3 ms is too brief to ensure that the air-fuel mixture will ignite reliably. Thus CDI is only designed for specific types of engine, and today its use is restricted to a limited application range, as transistorized ignition systems now afford virtually the same performance. CDI is not suited for aftermarket installations.

CDI can also be employed for distributorless ignition (DLI) with the installation of one ignition coil per cylinder, with energy distribution taking place at the medium-voltage level.

Capacitor-discharge ignition system with induction-type pulse generator, schematic

1 Control unit, 2 Charger, 3 Pulse shaper, 4 Control stage, 5 Ignition transformer, 6 To induction-type pulse generator, 7 To ignition distributor.



Accident hazard

All electronic ignition systems (including capacitor discharge, transistorized ignition and all systems with partial or comprehensive electronic control) are potentially dangerous. Always switch off the ignition or disconnect the battery before performing any service or maintenance including:

– Replacing components such as spark-plugs, coils, transformers, distributors, high-tension cables, etc.

– Connecting engine test devices such as stroboscopic lamps, dwell-tach testers, ignition oscillographs etc. Dangerously high voltage levels are present throughout the ignition system whenever it is turned on. All service operations should be performed exclusively by qualified technicians.

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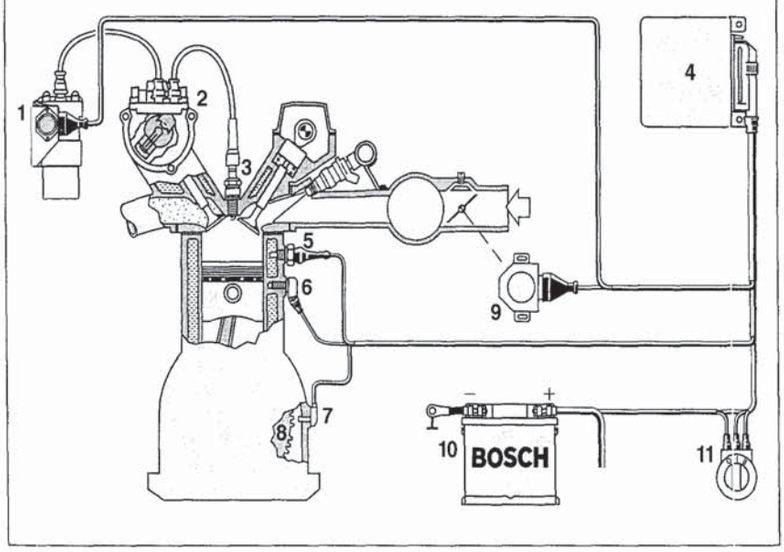
Electronic ignition (EI and DLI)

Electronic ignition derives its name from the fact that it calculates the ignition point electronically. The characteristic curves provided by the conventional distributor's centrifugal and vacuum-advance units are replaced by an optimized electronic ignition map. Mechanical high-tension distribution is retained with EI ignition. Fully electronic distributorless semiconductor ignition (DLI) uses stationary electronically controlled components to replace the mechanical, rotating high-tension distributor.

Electronic ignition systems operate more precisely than mechanical systems, with major benefits originating in the fact that the ignition process can be triggered from the crankshaft instead of from a distributor (distributor drive tolerances are no longer a factor). The limitations which mechanical adjustment mechanisms place upon the performance curve (summation of curves for load and engine speed in a single progression) are also avoided. The number of input variables is also theoretically unlimited, usually allowing extensions in the ignition angle's adjustment range. The fixed-drive ignition distributor's limitations regarding the engine's ignition-voltage requirements and ignition-angle adjustment range are such that it has difficulty coping with larger numbers of cylinders; efficient spark distribution cannot always be guaranteed. Corrective measures include dividing the ignition into two circuits (e.g., for 8- and 12-cylinder engines) and static voltage distribution.

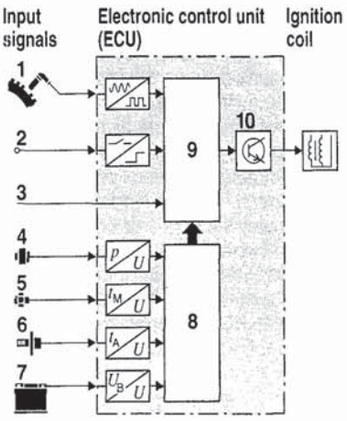
Electronic ignition can be combined with electronic fuel-injection (Motronic), knock control, ASR, etc., making it possible to employ sensors and/or signals from other units in more than one system. A serial bus (CAN, P. 884) further reduces the number of inputs and processing circuits on the ECU's input-side.

Schematic of an electronic ignition system (EI)
1 Ignition coil with ignition driver stage, 2 High-voltage distributor, 3 Spark plug, 4 ECU, 5 Engine-temperature sensor, 6 Knock sensor, 7 Engine-speed and reference-mark sensor, 8 Ring gear for sensor, 9 Throttle switch, 10 Battery, 11 Ignition switch.

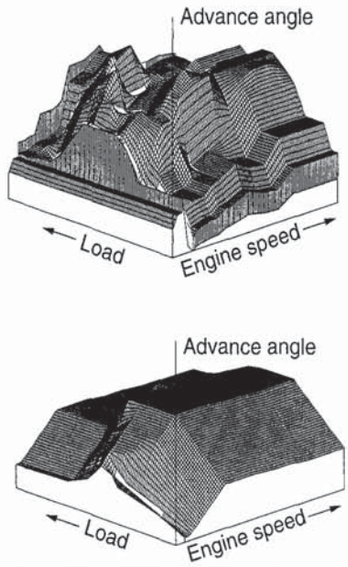


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Electronic ignition, signal processing
1 Engine speed, 2 Switch signals, 3 CAN (serial bus), 4 Intake-manifold pressure, 5 Engine temperature, 6 Intake-air temperature, 7 Battery voltage, 8 Microprocessor, 9 Analog/digital converter, 10 Driver stage.



Ignition maps
Above: electronically optimized, Below: mechanical advance system.



Operation

The engine's speed and crankshaft position are monitored directly at the ring gear, using either a separate rotor or a specific pin sequence employing an inductive, rod-type sensor, with two sensors being employed on older units. Triggering is either incremental or segmentary, according to whether the information is taken from teeth distributed evenly around the crankshaft or a crankshaft segment per cylinder pair:

- Beginning of segment = maximum spark advance angle,
- End of segment = starting angle.

In the incremental system illustrated here, the reference mark (shown as a tooth gap) represents a defined crankshaft position, providing a reference for electrically monitoring crankshaft angle using the ring-gear teeth. A distributor without an advance mechanism can also be employed for triggering; here the control signal is provided by a Hall generator. An absolute-pressure sensor in the intake manifold is the best way to monitor load. This provides a better gauge of cylinder charge than the spark-advance or retard bores on the throttle valve. It is also possible to incorporate load switches, throttle-valve potentiometers or electronic load signals from the mixture-preparation system. The microcomputer in the ECU processes the engine-speed and load signals prior to using them to calculate the precise ignition angle within the ignition map. The computer can also process other input variables, such as e.g. engine temperature, or information on trailing throttle or full-load operation from the throttle-valve switch, to derive correction values and regulate other vehicle-specific functions as required. The dwell angle for charging the ignition coil is also specified by the computer. The battery voltage is monitored here to enable voltage correction. The system responds to deviations in battery voltage from the specified baseline by extending or reducing coil-charging times accordingly. This ensures consistent availability of maximum voltage while at the same time limiting the accumulation of heat in ECU and coil to a minimum. At speeds below the starting

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speed, the computer interrupts the current flow through the ignition coil in order to prevent overheating. The driver stage can be either integrated inside the ECU or installed externally, e.g. on the ignition coil.

Signal processing in the ECU

After initial processing the digital signals go directly to the processor. Analog signals are first converted into digital form. There are also ECU's which can transmit supplementary digital or analog signals e.g., for overrun fuel cutoff and exhaust-gas recirculation). There exists a range of EI ignition-system versions of varying complexity. A comparison of the ignition map with the response curve of a distributor shows that it is possible to program each point in the ignition map independently from every other point. Thus the optimum ignition timing (e.g., for maximum fuel economy) can be selected for every operating condition, according to the limits imposed by factors such as exhaust emissions, pre-ignition limit and driveability. The entire system is completely maintenance-free and does not recuire any adjustments during the engine's service life.

Rotating voltage distribution

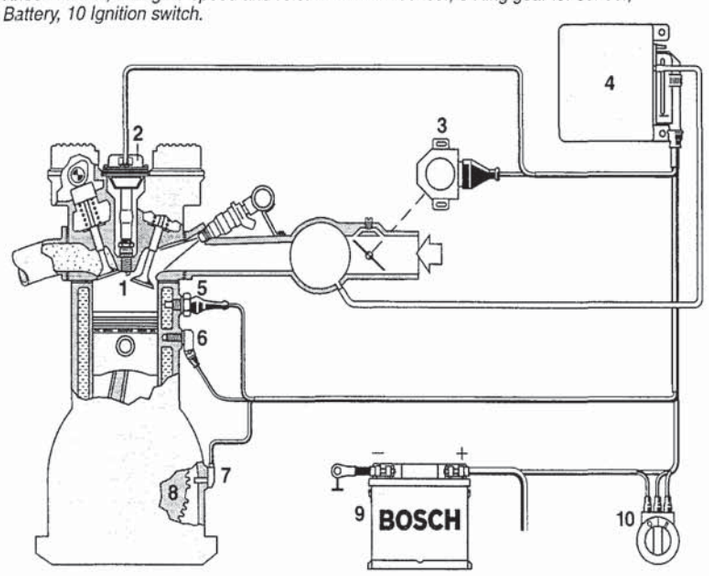
A high-voltage distributor distributes the ignition pulses to the spark plugs of the individual cylinders (as described earlier). If the distributor's adjustment range is insufficient for handling a larger number of cylinders, then two ignition circuits are employed, i.e., two 4-cylinder distributors can be used for one 8-cylinder engine. Synchronization through the crankshaft can be used for "two times 4 cylinders", whereas "two times 3 cylinders" with constant ignition-angle spacing (not employed up to now) must be controlled by the camshaft.

Distributorless (stationary) voltage distribution

Systems with single-spark ignition coil

Each cylinder has its own ignition coil with driver output stage, installed either di-

Schematic of a fully-electronic distributorless ignition system (DLI)
1 Spark plug, 2 Single-spark coil, 3 Throttle switch, 4 ECU, 5 Engine-temperature sensor, 6 Knock sensor, 7 Engine-speed and reference-mark sensor, 8 Ring gear for sensor, 9 Battery, 10 Ignition switch.



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rectly above the spark plug or separately. Either synchronization with the camshaft sensor or a method for detecting the compression cylinder is required. On engines with an even number of cylinders, the system reverts to crankshaft triggering in the event of camshaft-sensor failure, although two coils are then always activated simultaneously (one of the sparks is discharged during an exhaust stroke). This system, suitable for engines with any number of cylinders, provides the greatest latitude for adjustment, as there is only one spark per cycle. All these advantages mean that the single-spark ignition coil is being increasingly used and is taking over from the dual-spark ignition coil in spite of costing more.

Systems with dual-spark ignition coil

One ignition coil is required for every two cylinders. The crankshaft can be used for synchronization. The high-voltage end of each ignition coil is connected to the spark plugs for two cylinders whose operating cycles are 360° out of phase with each other.

As there is an additional spark during the exhaust stroke, it is important to ensure that residual mixture or fresh mixture is not ignited. Furthermore, the dual-spark system is only suited for use with even numbers of cylinders. Owing to its cost advantage relative to the single-spark unit, the dual-spark ignition system is the most common distributorless ignition in use today.

Knock control

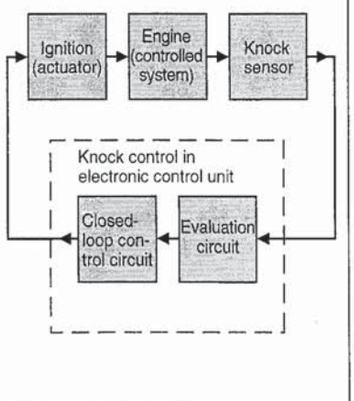
Application

Electronic control of the ignition point offers the possibility of controlling very precisely the ignition angle as a function of engine speed, load and temperature.

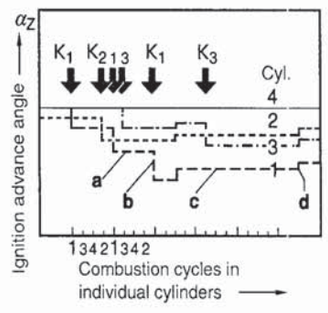
Nevertheless, without knock control a clear safety distance to the knock limit is required. This distance is necessary to ensure that even in the most knock-sensitive case with regard to engine tolerances, engine aging, environmental conditions and fuel quality, no cylinder reaches or exceeds the knock limit. The resulting engine design leads to lower compressions with retarded ignition points and thus worsening of fuel consumption and torque figures.

These disadvantages can be avoided through the use of knock control. Experience shows that knock control increases engine compression and significantly improves fuel consumption and torque. However, the pre-control ignition angle now no longer needs to be deter-

Schematic of knock-control system



Knock control *Control algorithm for ignition adjustments with a 4-cylinder engine*
$K_{1...3}$ knock in cylinders 1...3, Cylinder 4, no knock.
a Delay prior to ignition retard, b Retardation, c Delay before return to original ignition point, d Spark advance.



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mined for the most knock-sensitive but rather for the most insensitive conditions (e.g. engine compression at lower tolerance limit, best possible fuel quality, most knock-insensitive cylinder). Each individual engine cylinder can now be operated over its entire service life in virtually all operating ranges at its knock limit and thus at optimal efficiency.

The essential precondition for this ignition-angle configuration is reliable knock detection for each individual cylinder as from a particular knock intensity, and over the entire engine operating range.

Combustion knock

The characteristic vibrations of combustion knock are detected by knock sensors, converted into electrical signals and forwarded to the Motronic.

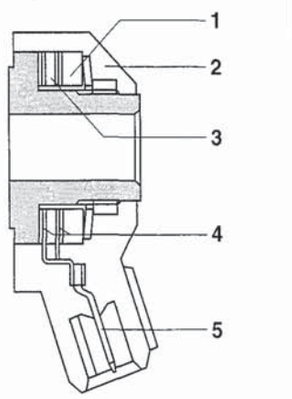
It is essential to work out carefully the number and installation positions of the required knock sensors. Reliable knock detection must be guaranteed for all cylinders and for all engine operating points, particularly at high engine speeds and loads. As a rule, 4-cylinder in-line engines are equipped with one knock sensor, 5- and 6-cylinder engines with two sensors and 8- and 12-cylinder engines with two or more sensors.

Knock detection, evaluation

For the purpose of knock detection, the vibrations characteristic to knocking are converted into electrical signals by one or more knock sensors mounted at appropriate positions on the engine, and then forwarded to the Motronic for evaluation. It is here that the corresponding evaluation algorithm is applied to detect knock for each cylinder and each combustion. Detected combustion knocks result in a retardation of the ignition point by a programmable amount at the cylinder in question. If knock stops, the ignition point is advanced again in stages up to the pre-control value.

The knock-detection and knock-control algorithms are matched in such a way as to eliminate any knocking that is audible and damaging to the engine.

Knock sensor
1 Seismic mass, 2 Potting compound, 3 Piezoceramic element, 4 Contacts, 5 Terminals.



Adaptation

Real engine operation produces different knock limits and thus also different ignition points for the individual cylinders. In order to adapt the ignition-point pre-control values to the particular knock limit, the ignition-point retardation values individual to each cylinder and dependent on the operating point are stored.

They are stored in non-volatile maps of the permanently powered RAM covering load and engine speed. In this way, the engine can also be operated in the event of rapid load and speed changes in each operating point at optimal efficiency and without audible combustion knocks.

The engine can even be approved for fuels with lower anti-knock properties.

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Combined ignition and injection system (Motronic)

The Motronic engine-management system has undergone substantial development since its introduction in 1979. Intially, integration was based on the basic gasoline-injection systems, in combination with distributorless semiconductor ignition, as were available in the following systems:
- KE-Motronic, based on KE continuous gasoline injection (P. 469),
- Mono-Motronic, based on intermittent single-point injection (P. 465),
- M-Motronic, based on intermittent multipoint injection (P. 471).

On the basis of M-Motronic, and by applying further integration steps, the Motronic system assumed control of all the manipulated variables of a spark-ignition engine that influence torque:
- ME-Motronic with electronic accelerator pedal (ETC) for controlling gasoline injection, ignition and fresh-air charge for manifold injection (P. 498).
- MED-Motronic with further integrated open- and closed-loop control functions for the high-pressure fuel circuit. With direct injection and realization of the various operating modes of this engine type (P. 507).

System overview

The Motronic system comprises all the sensors for recording the current engine and vehicle operating data (P. 501, Operating-data acquisition) and all the actuators for the adjustments to be carried out on the SI engine (P. 503, Operating-data processing).

Electronic control unit

The ECU employs sensors to monitor the relevant status of engine and vehicle at extremely short intervals (milliseconds).

Input circuits suppress sensor-signal interference and convert the signals to a single unified voltage scale. An analog-digital converter then transforms the conditioned signals into digital values. Further signals are received by way of a digital interface. Using this information, the microprocessor identifies the operating state desired by the driver and from it calculates for instance:
- the required torque,
- the resulting cylinder charge with the associated injected fuel quantity, and
- the correct ignition timing.

The low-level signal data from the microprocessor outputs are adapted by the driver stages to the levels required by the various actuators. A semiconductor memory chip stores all programs and performance maps, ensuring system consistency which remains completely impervious to fluctuations resulting from signal-level and component tolerances.

Digital accuracy is a function of word length, as well as of the consistency of the quartz's basic clock frequency and the types of algorithms used for the calculations. The consistency and precision of the reference voltages and the components installed in the analog input circuits influence the analog accuracy.

Program design must satisfy the engine's severe real-time demands: in an 8-cylinder engine at maximum speeds, less than 2.5 ms are available between two ignitions. All essential calculations must be completed within this period. In addition to these crankshaft-synchronous processes, there are also time-synchronous operations. Both types can be suspended by interrupts.

ME-Motronic

Design

Basic functions

In order to set the operating state desired by the driver (main function), the position of the gas pedal is converted by the microprocessor into a setpoint value for the engine torque. Then, while taking into consideration the highly varying current operating data from the ME-Motronic, these are converted into the variables for determining the engine torque:
- the charging of the cylinders with air,
- the mass of the injected fuel and
- the ignition angle.

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ME-Motronic system diagram
1 Carbon canister, 2 Shut-off valve, 3 Canister-purge valve, 4 Intake-manifold pressure sensor, 5 Fuel rail/fuel injector, 6 Ignition coil/spark plug, 7 Phase sensor, 8 Secondary-air pump, 9 Secondary-air valve, 10 Air-mass meter, 11 Throttle device (ETC), 12 EGR valve, 13 Knock sensor, 14 Engine-speed sensor, 15 Temperature sensor, 16 Lambda sensor, 17 ECU, 18 Diagnosis interface, 19 Diagnosis lamp, 20 Vehicle immobilizer, 21 Tank pressure sensor, 22 In-tank unit, 23 Accelerator-pedal module, 24 Battery.

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Additional function
As well as the basic functions, the ME-Motronic has a large number of additional open- and closed-loop control functions. Examples of such functions are:
- idle-speed control,
- closed-loop lambda control,
- control of evaporative-emissions control system,
- exhaust-gas recirculation for reducing NO_X emissions,
- control of secondary-air system for reducing HC emissions, and
- cruise control.

These functions have been made necessary by the legislation covering the reduction of exhaust emissions, the calls for a decrease in fuel consumption but also increased demands placed on driving comfort and safety.

In addition, the ME-Motronic system can be further extended with the following functions:
- Control of turbocharger control and variable-tract intake manifold for increasing engine power output,
- Camshaft control for reducing exhaust emissions and fuel consumption and for increasing power,
- Knock control, engine-speed limitation and vehicle-speed limitation for protecting components in the engine and vehicle.

Torque-guided control concept
The torque-guidance principle is aimed at simplifying the many and sometimes very different assignments placed on the engine and the vehicle. Only then is it possible, depending on the engine or vehicle type, to determine the required functions in each case and to integrate them in the relevant Motronic version.

Most of the additional open- and closed-loop control functions likewise influence the engine torque. In a torque-guided system, all these functions behave like the driver: they request an engine torque irrespective of each other. The torque-guided ME-Motronic uses torque coordination to sort the often contradictory torque requirements and to then put the most important requirement into effect.

Vehicle management
The ME-Motronic can communicate via the CAN bus system (P. 884) with the ECUs of other vehicle systems. Among other things, in conjunction with the automatic-transmission ECU, it reduces torque at the gear-shift point so that the transmission is subjected to less loading and wear. When wheel spin occurs, the ECU belonging to the Traction Control System (TCS) signals to the ME-Motronic to reduce the generated torque.

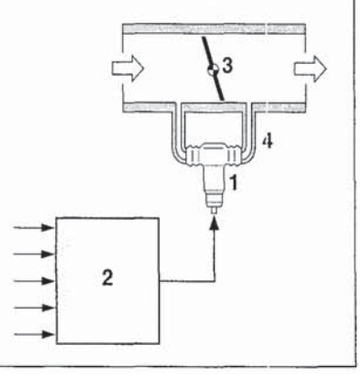
Diagnosis
The ME-Motronic incorporates components for on-board monitoring (OBD). It therefore complies with the most stringent emission limits and the demands placed on the integrated diagnosis functions.

Cylinder-charge control systems

Throttle-valve control
In an SI engine with external A/F mixture formation, the cylinder charge is the factor that determines torque output and thus power output. The throttle valve controls the air-flow intake into the engine and thus the cylinder charge.

In conventional systems, a cable or a linkage transfers the movement of the accelerator pedal to the throttle valve. Requirements for additional air are covered by a bypass air actuator which directs an additional air flow around the

Air control with bypass air actuator *(principle)*
1 Idle actuator (bypass air actuator), 2 ECU, 3 Throttle valve, 4 Bypass.



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throttle valve (see Fig.), or a throttle-valve actuator alters the stop for minimum air quantity. However, the air flow required by the engine can only be electronically influenced to a limited extent (idle-speed control).

Systems with ETC (electronic throttle control, integrated in the engine control unit for ignition, injection and other auxiliary functions) detect the position of the accelerator pedal with a pedal-travel sensor (potentiometer).

Taking into account the current engine operating status, the ECU calculates how far the throttle valve needs to open and actuates the throttle-valve drive accordingly (see Fig.). A throttle-valve angle sensor (potentiometer) monitors exact compliance with the desired throttle-valve position. Two potentiometers (for redundancy reasons) at the accelerator pedal and throttle device are part of the ETC monitoring system which, while the engine is running, continually checks all the sensors and calculations which affect throttle-valve opening.

Charge-cycle control

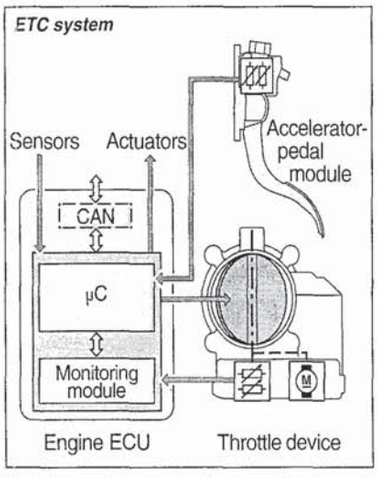
The behavior of the gas columns flowing into and out of the cylinders changes considerably as a function, for instance, of engine speed or throttle-valve opening. With fixed valve timing, the charge cycle can only be optimized for a specific operating range.

Camshaft control or camshaft-lobe control (P. 387) permits not only variable valve timing, and thus adaptation to various engine speeds, but also enables the residual exhaust-gas mass in the cylinder to be influenced ("internal EGR").

("External") EGR effectively reduces the temperature-dependent NO_x emissions. Exhaust gases which have already been combusted are added to the A/F mixture in order to reduce peak combustion temperatures. For this purpose, the Motronic triggers the exhaust-gas recirculation valve as a function of the engine operating point. The EGR valve then adds a partial flow of exhaust gas to the fresh mixture, thereby increasing the total charge while the fresh-air charge remains constant. In this way, the engine must be throttled less heavily in order for a specific torque to be reached (lower fuel consumption).

The achievable torque, which is proportional to the fresh-mixture charge, can be increased still further by compressing the air in the cylinder. This is the purpose of dynamic supercharging, exhaust-gas turbocharging, or mechanical supercharging (P. 390).

ETC system



Operating-data acquisition

Vehicle-operator command

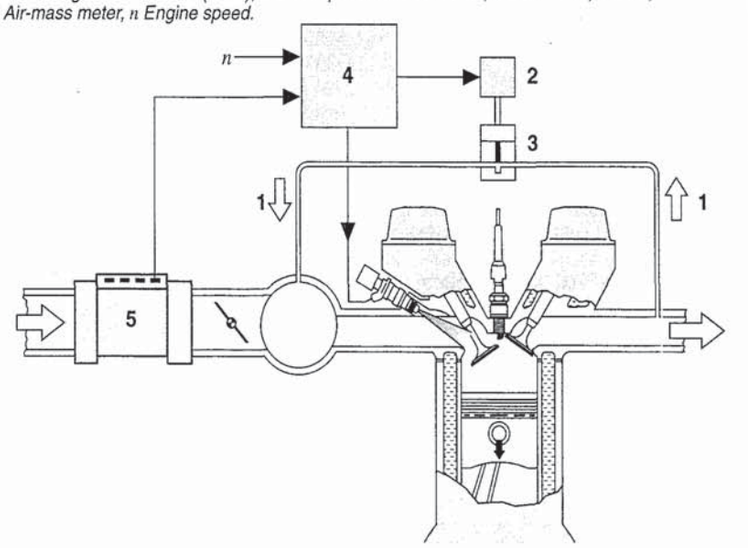
In the case of engine management with an electronic throttle valve (ETC), there is no longer a mechanical connection between the accelerator pedal and the throttle-valve actuator. Instead, the position of the accelerator pedal is sensed by a pedal-travel sensor or accelerator-pedal module and converted into an electrical signal. The engine-management system interprets this signal as the vehicle-operator command.

Air charge

In engine concepts with manifold injection, there is a linear relationship between the air charge and the torque generated by combustion. In the torque-guided ME-Motronic, the air charge is therefore not just one of the main factors involved in calculating the injected fuel quantity and

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Exhaust-gas recirculation *(example)*
1 Exhaust-gas recirculation (EGR), 2 Electropneumatic converter, 3 EGR valve, 4 ECU, 5 Air-mass meter, n Engine speed.



ignition angle, but also serves to calculate the torque currently being delivered by the engine. The following sensors can be used for this purpose:
– Hot-film air-mass meter,
– Intake-manifold pressure sensor,
– Ambient pressure sensor,
– Boost-pressure sensor (BPS) and
– Throttle-valve sensor.

Engine speed, crankshaft position and camshaft position
A speed sensor on the crankshaft monitors the crankshaft revolutions per minute (rotational speed). It then forwards this information on the crankshaft position, that is, the piston position of all cylinders, as an important input variable to the ME-Motronic. The cylinder's piston position is used, for instance, to determine the injection and ignition points.

To trigger the single-spark ignition coils, the ME-Motronic with static voltage distribution also requires additional information on the camshaft position. This information is mostly monitored with a Hall-effect sensor and forwarded to the ECU as a switch signal.

A/F mixture composition
The lambda factor (λ) is the figure which quantifies the A/F mass ratio of the mixture. The catalytic converter only performs optimally at $\lambda = 1$. The lambda sensor measures the oxygen concentration in the exhaust gas and provides information on the current value of the excess-air factor (P. 458).

Combustion knocks
The characteristic vibrations of combustion knocks are detected by knock sensors, converted into electrical signals and forwarded to the ME-Motronic (P. 498).

Engine and intake-air temperatures
The voltage at the NTC resistor of the engine-temperature sensor is a function of the coolant temperature. It is input via the analog-digital converter and is a measure of the temperature. In the same way, a sensor in the intake port measures the temperature of the intake air.

Operating-data processing

Torque guidance
Torque guidance is a feature of the ME-

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Motronic. Numerous subsystems within the Motronic (e.g. idle-speed control, engine-speed limitation), and the systems for management of the drivetrain (e.g. TCS or transmission-shift control) or of the complete vehicle (e.g. air-conditioner control), direct their torque requirements to the basic Motronic system with the aim of modifying the engine torque which has just been generated. The ME-Motronic evaluates and coordinates these requirements and implements the resulting setpoint torque using the available manipulated variables (optimum engine operation in terms of exhaust gas and fuel consumption at each operating point).

An essential precondition for torque guidance is the electronic accelerator pedal ETC with the throttle valve controlled independently of the accelerator pedal.

Calculation of setpoint torque

The basic variable for the ME-Motronic torque structure is the internal torque arising from combustion. The function of torque guidance is to set the internal torque, through appropriate selection of the manipulated engine variables, in such a way that the vehicle-operator command is satisfied, and all the losses and additional requirements are covered. Because the Motronic "knows" the optimum values for charge, injection time and ignition angle for each and every desired setpoint torque, it can ensure optimal engine operation in terms of exhaust gas and fuel consumption.

Setting of setpoint torque

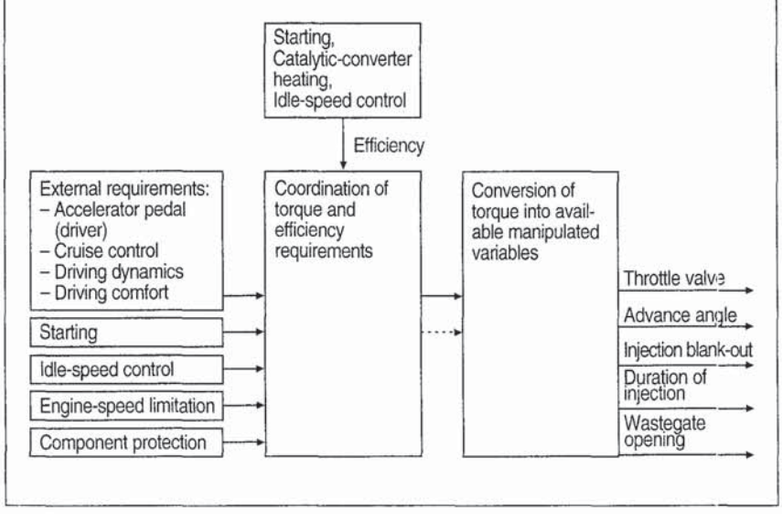
For setting the internal torque, the ME-Motronic's torque converter has a slow path (cylinder-charge path, steady-state operation) for the throttle valve (ETC), and a high-speed path (ignition-angle path, dynamic operation) for variation of the ignition angle and/or injection blank-out of individual cylinders.

Calculation of cylinder charge

The air mass in the cylinder after the intake valves have closed is termed the air charge. Referring the actual air charge to the maximum obtainable charge produces a variable that is independent of the engine piston displacement: the "relative (air) charge". The associated injected fuel quantity can be calculated from the relative air charge.

The relative charge cannot be measured directly but it can be determined from the available measurement signals.

Influencing torque in a gasoline engine



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For this purpose, an intake-manifold model calculation is used in which all the air-mass flows into the intake manifold and from the intake manifold to the combustion chamber are calculated. The most important measured variable in this model is either the air flow through the throttle valve (measured with a hot-film air-mass meter) or the absolute intake-manifold pressure (measured with a pressure sensor).

Starting out from this intake-manifold model, it is then also possible to determine the actual relative air charge.

Cylinder-charge control

In today's spark-ignition engines, the relative cylinder charge is also the main factor in influencing the engine torque, and is therefore used in the torque structure as a manipulated variable. The intake-manifold model is also used to control the cylinder charge by means of the throttle valve: for this purpose, a required cylinder charge is first of all calculated from the desired torque (which was calculated by ME-Motronic torque guidance). This cylinder charge is then converted into an associated throttle-valve angle and finally passed on as the setpoint value to the position controller of the throttle-valve actuator.

Calculation of injection period

The fuel mass for a stoichiometric air-fuel ratio can be calculated from the air charge in the cylinder. When the fuel-injector constant (this is a function of the fuel-injector properties) is taken into consideration, this furnishes the duration of injection.

The duration of injection is also dependent on the pressure differential between fuel-supply pressure (approx. 300 kPa) and injection back pressure. Fuel-supply systems with fuel recirculation keep the supply pressure constant with respect to the intake manifold. This ensures that, in spite of a changing intake-manifold pressure, the same pressure differential is applied at the fuel injectors. On the other hand, fuel-supply systems without fuel recirculation keep the supply pressure constant with respect to the environment. As the intake-manifold pressure changes, the pressure differential varies between fuel supply and intake manifold. A compensation function corrects this error.

A further adaptation factor individual to each cylinder serves to accommodate fuel-pressure pulsations caused by the opening and closing of the fuel injectors.

The effective opening duration calculated in this way is valid under the precondition that the fuel injector is already open. It must therefore be extended by the opening and closing time of the injector. An additional, battery-voltage-dependent duration of injection, which is added to the valve-opening period, compensates for this effect.

If the effective injection duration is too short, the influences of the valve-opening and closing time become excessive. In order to ensure exact fuel metering, the injection duration is restricted to a minimum value. This value is below the injection duration associated with the minimum possible cylinder charge.

To ensure optimum combustion, it is necessary to precisely define the injected fuel quantity and the exact point for the start of injection. The fuel is generally injected into the intake manifold while the intake valves are still closed. The end of injection is determined by the "fuel-hold angle", which is specified in "crankshaft degrees". The point at which the intake valve closes serves as the reference point. It is then possible to use the duration of injection for calculating, as a function of engine speed, the start of injection as an angle. The fuel-hold angle is determined taking into account the current operating conditions.

The ME-Motronic triggers a fuel injector for each cylinder. The fuel can thus be held (stored) separately for each cylinder to optimum effect (sequential injection).

Calculation of ignition and dwell angles

When calculating the ignition angle, the first step is to determine the "basic ignition angle" while taking into account the current engine operating conditions. This basic ignition angle can be adapted in specific operating states.

When calculating the dwell angle, the first step is to determine the dwell period from a speed- and battery-voltage-dependent map. Once a temperature-depen-

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dent correction has been taken into account, the associated dwell angle is determined by means of a time/angle conversion.

The difference between the end of dwell (determined from the resulting ignition angle) and the dwell angle determines the start of dwell.

Operating states

In some operating states, the fuel requirement differs markedly from the steady-state requirements of an engine at normal operating temperature, with the result that corrective intervention in the mixture formation becomes necessary.

Starting

Intake-air adjustment, fuel injection and ignition are specially calculated during the starting procedure. An increased injected fuel quantity adapted to the engine temperature serves to build up a fuel film on the intake-manifold and cylinder walls and covers the increased fuel demand while the engine is running up. The ignition angle is likewise adapted to the starting procedure. The air charge while the engine is at a standstill is not affected by the throttle valve, which however is opened slightly in anticipation of the post-start phase.

Post-start

In this phase, the still increased charge and injected fuel quantity are reduced as a function of engine temperature and time at the end of the starting procedure. The ignition angle is adapted accordingly.

Warming-up

After starting at low engine temperatures, the engine's increased torque demand is covered up to a specific temperature threshold by adaptation of the charge, fuel injection and ignition.

Catalytic-converter heating

Highly retarded ignition angles generate very hot engine exhaust gas, thereby quickly bringing the catalytic converter up to operating temperature.

Idle

At idle, the engine-generated torque is just enough to permit sustained engine operation and running of the auxiliary systems. With idle-speed control, the desired idle speed remains stable under all conditions.

Full load

At full load, the throttle valve is wide open (WOT) and there are no throttling losses. The engine delivers the maximum torque relative to the current engine speed.

Acceleration and deceleration

Heavy acceleration and deceleration result in rapid changes in the intake-manifold pressure and thus in the conditions for the fuel film on the intake-manifold wall. In order to prevent leaning during acceleration, additional fuel is injected to build up the wall film. During deceleration, the injected fuel quantity is reduced accordingly.

Overrun fuel cutoff and restart

During the transition to no-combustion overrun conditions, the ME-Motronic cuts off the engine torque without jolting, and also provides for a smooth torque build-up when fuel feed starts again.

Additional closed- and open-loop control functions

Idle-speed control

Idle-speed control specifies a torque at which the desired engine speed is maintained under the given operating conditions. This torque increases as engine speed drops and decreases as engine speed rises.

Lambda closed-loop control

It is only possible to convert the noxious exhaust-gas constituents in the three-way catalytic converter in a very narrow range ("lambda window" where $\lambda = 0.99...1$). Lambda closed-loop control is essential in order to remain inside this window (P. 522).

Evaporative-emissions control system

The function of evaporative-emissions control systems is to limit the HC emissions. Such systems are equipped with a carbon canister, which accommodates

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the end of the vent line from the fuel tank. Degassing fuel is retained in the canister's activated carbon. When the engine is running, a vacuum is created in the intake manifold which causes air to be drawn in from the environment through the activated carbon and into the intake manifold. The fuel vapors trapped in the canister are entrained with this air and carried to the engine for combustion. A canister-purge valve meters this "scavenging flow".

Knock control

Knock sensors convert the vibrations characteristic to knocking into electrical signals. These are used by the Motronic to adjust the ignition point in order to prevent audible knocking and knocking which could damage the engine (P. 496).

Boost-pressure control

On engines with exhaust-gas turbocharging, the setpoint value of the desired boost pressure is adjusted by electronic boost-pressure control. It is converted into a setpoint value for the desired maximum charge and then by means of torque management into a setpoint value for the throttle-valve angle and a control duty factor for the "wastegate". In the "wastegate", by means of a change in the control pressure and stroke this signal leads to a change in the cross-section at the bypass valve.

Safety and security, comfort and convenience functions

The safety and security functions include:
- Engine-speed/vehicle-speed limitation,
- Torque/power limitation,
- Limitation of exhaust-gas temperature,
- Vehicle immobilizer.

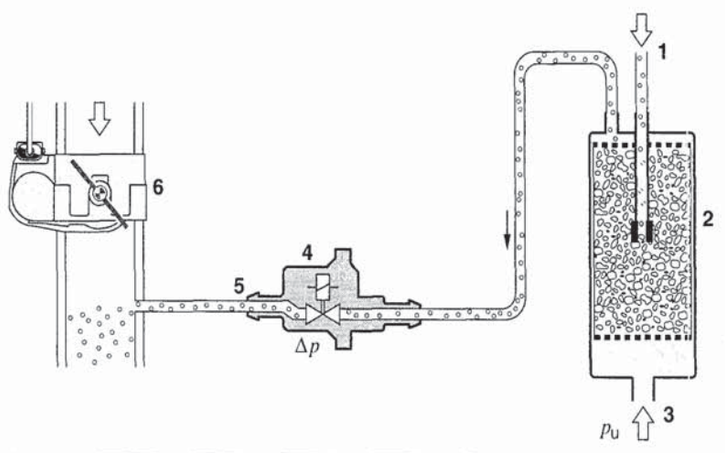
The comfort and convenience functions include:
- Load-change damping,
- Surge-damping function,
- Cruise control.

Integrated diagnosis (OBD)

"On-board diagnosis" (OBD) is a basic feature of the ME-Motronic. During normal operation, it continually compares the system's responses with the ECU's commands, as well as comparing the various sensor signals with each other for plausibility.

During vehicle inspection, a tester connected to a standardized interface reads out the stored faults and displays them. Due to the demands of the Californian Environmental Protection Agency, an ex-

Evaporative-emissions control system
1 Line from fuel tank to carbon canister, 2 Carbon canister, 3 Fresh air, 4 Canister-purge valve, 5 Line to intake manifold, 6 Throttle device with throttle valve.
Δ_p Difference between intake-manifold pressure p_S and ambient pressure p_U.



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panded diagnosis system (OBD II; adapted for Europe: EOBD) monitors all those components which if they fail could result in a significant increase in noxious emissions (fault indication with diagnosis lamp).
Examples of systems subject to diagnosis:
- Air-mass meter,
- ETC throttle-valve actuator,
- Combustion misses,
- Catalytic converter,
- Lambda sensor,
- Fuel supply,
- Tank system,
- Secondary-air injection,
- Exhaust-gas recirculation (EGR).

Fault memory
If exhaust-gas-related faults are detected, an entry is made in the non-volatile fault memory. Apart from the officially stipulated fault codes, each entry includes a "freeze frame", which contains additional information on the general conditions under which the fault occurred (e.g. engine speed, engine temperature). Service-related faults can also be stored.

Limp-home operation
If a fault occurs during driving (e.g. failure of a component), this leads not only to an entry in the fault memory. Substitute values and emergency functions now come into force for calculating cylinder charge, fuel injection and ignition in such a way that the vehicle can continue to be driven, albeit with restricted comfort.

For this purpose, the ETC throttle-valve actuator has a limp-home position, in which the throttle valve is held in place by spring force. The engine speed is then restricted to low values so that the vehicle can continue to be driven (with impaired functions) in spite of this important actuating device having failed.

MED-Motronic

Requirements

With gasoline direct injection in SI engines, fuel consumption can be reduced by up to 20% when compared with conventional manifold injection, and the CO_2 emissions caused by road traffic can be reduced with lasting effect.

In order for direct injection to be at all feasible, it must be possible during engine operation for coordinated alternation to take place between "charge stratification" at part load and operation with a homogeneous mixture at full load.

Essentially, the features required of the MED-Motronic engine-management system are:
- precise metering of the required injected fuel quantity,
- generation of the necessary injection pressure,
- definition of the start of injection, and
- introduction of the fuel directly and accurately into the engine's combustion chambers.

It must also coordinate the various torque demands on the engine in order then to be able to perform the required adjustments at the engine.

An important system interface delivers the internal torque from combustion. The torque-control structure is divided into the following areas of operation:
- torque demand,
- torque coordination and
- torque conversion.

The most important torque demand derives from the vehicle-operator command generated by the accelerator pedal, from the position of which the engine-management system interprets the demand to be made on the IC engine for a particular torque.

Further torque demands can also originate, for instance, from the transmission-shift control, the traction control system, or the electronic stability program.

Torque coordination is performed centrally in the engine-management system.

Design

As with manifold injection, the high-pressure direct-injection system is designed as a rail, or accumulator-type, injection system. In such systems, the fuel can be directly injected into the cylinders at any

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stipulated moment in time using electromagnetic high-pressure fuel injectors.

In comparison to the ME basic ECU, the ECU for the MED gasoline direct-injection system also incorporates a driver stage for triggering the pressure-control valves.

The intake air mass can be freely adjusted by the electronically controlled throttle valve (ETC). Precision measurement of air mass is made by means of a hot-film air-mass meter. The A/F mixture is monitored by universal lambda sensors LSF and LSU in the exhaust-gas flow upstream and downstream of the catalytic converter. They serve to control $\lambda = 1$ operation and lean-burn operation, and are also responsible for the precise control of catalytic-converter regeneration. It is important to set the exhaust-gas recirculation rate exactly, particularly in dynamic operation.

Operation

Fuel delivery and injection

The MED-Motronic accumulator injection system injects the fuel stored under pressure in the rail. Injection can take place at any moment in time, whereby electromagnetically controlled fuel injectors inject the fuel directly into the combustion chamber. It thus offers the following functions:

- free selection of the injection point,
- variable system pressure.

Low-pressure circuit:

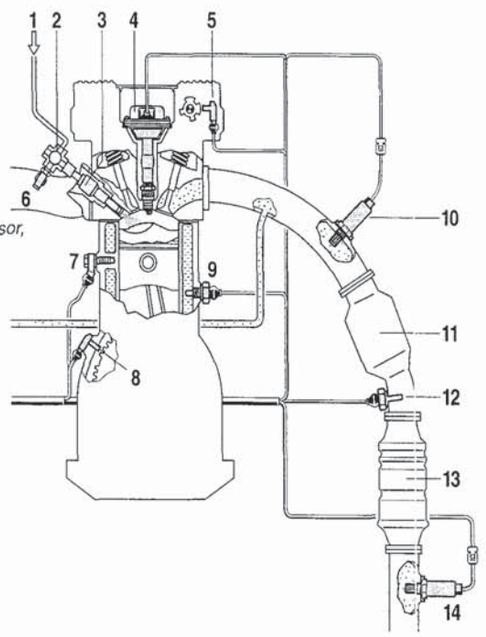
A primary pressure of 0.35 MPa (3.5 bar) is initially generated in a low-pressure circuit on the tank side which comprises an electric fuel pump with a mechanical pressure regulator connected in parallel. This feeds the high-pressure pump driven by the IC engine.

High-pressure circuit:

High-pressure pump: The function of the H.P. pump is to increase the fuel pressure

Direct-injection engine with MED-Motronic components

1 Fuel supply (high pressure),
2 Accumulator (rail),
3 Fuel injector,
4 Ignition coil with spark plug,
5 Phase sensor,
6 Pressure sensor,
7 Knock sensor,
8 Engine-speed sensor,
9 Engine-temperature sensor,
10 Lambda sensor (LSU),
11 Three-way catalytic converter,
12 Exhaust-gas temperature sensor,
13 NO_x catalytic converter,
14 Lambda sensor (LSF).



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from 0.35 MPa (3.5 bar) primary pressure to 12 MPa (120 bar), to ensure that pressure fluctuations in the rail remain at a minimum, and to ensure operation exclusively with fuel (to prevent any mixing with engine oil).

Rail: The rail must on the one hand feature sufficient elasticity to damp the pressure pulsations arising from the periodic fuel-extraction processes when injection takes place, and from the fuel-delivery pulsation generated by the H.P. fuel pump. On the other hand, it must be rigid enough for the rail pressure to adapt quickly to the requirements of engine operation.

Pressure sensor: The pressure sensor (welded diaphragm of high-grade steel as sensor element with measuring resistors) serves to register the pressure level in the rail.

Pressure-control valve: The pressure-control valve adjusts the system pressure across the entire engine operating range according to the map specifications, and irrespective of the injected fuel quantity and pump delivery rate. Excess fuel is returned to the intake side of the H.P. pump.

Fuel injectors: The fuel injectors are the central components of the gasoline direct-injection system and are directly connected to the rail. Start of injection and injected fuel quantity are defined by the triggering signal for the fuel injectors.

Mixture formation and combustion

A complex engine-management system is necessary for the gasoline direct-injection system to be exploited to the full in terms of low fuel consumption and high engine power output. A distinction is made between two basic operating modes in this respect:

Lower load range

In this range, the engine is run with a heavily stratified cylinder charge and a high level of excess air in order to achieve the lowest-possible fuel consumption. Retarded injection just before the ignition point is used with the aim of achieving the ideal state of having two zones in the combustion chamber: In one zone the combustible air-fuel mixture cloud at the spark plug, embedded in the second zone in the form of an insulating layer of air and residual exhaust gas. In this way, the engine can be run to a large extent unthrottled while gas-exchange losses are avoided. In addition, thermodynamic efficiency increases due to the avoidance of heat losses at the combustion-chamber walls.

Upper load range

As engine load and thus the injected fuel quantity increases, the stratified-charge cloud becomes increasingly richer. This would result in a deterioration in exhaust emissions, particularly with regard to soot. The engine is therefore operated with a homogeneous cylinder charge in this higher torque range.

Lambda coordination assumes the control function between $\lambda = 1$ and lean-burn operation. The fuel is injected during the intake stroke in order to ensure that fuel and air are thoroughly mixed.

Alternation between torque ranges

In the event of a switch between homogeneous and stratified-charge operation, it is crucial to control injected fuel quantity, air charge and ignition angle in such a way that the torque delivered by the engine to the transmission remains constant. The torque structure means that here also the important functions for controlling the electronic throttle valve have been transferred directly from the ME-Motronic. The throttle valve must be closed prior to the actual changeover from stratified to homogeneous operation.

Exhaust-gas treatment, catalytic-converter control

An important consideration with gasoline direct injection is that in stratified-charge operation the NO_X content in the very lean exhaust gas cannot be reduced by a three-way catalytic converter. Exhaust-gas recirculation with a high EGR rate helps to reduce the NO_X content of the exhaust gas by roughly 70 %. To comply with emissions-control legislation, it is also absolutely essential to treat the NO_X emissions. To reduce this exhaust-gas content, the NO_X accumulator-type catalytic converter offers the greatest potential.

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Engine-test technology

Service and maintenance requirements for modern vehicles are declining steadily; electronic systems are essentially maintenance-free. But malfunctions can still occur. Factors such as wear, contamination and corrosion can impair the operation of engine and electronic systems, and settings can drift over time. Rapid and reliable diagnosis of malfunctions is the most important function of any service facility. It is important to distinguish between testing and diagnosis. Testing entails the determination of certain measured values for comparison with the prescribed specifications. Diagnostic procedures (e.g. engine diagnostics) attempt to correlate deviations from specified values with system functions, malfunction patterns and experience in order to determine the failure mode or to locate the defective component.

Engine diagnosis

Engine, ignition and fuel-metering systems are becoming more complex and less accessible. Universal, automated procedures, free from the effects of subjective influences, are thus essential elements of the computer-controlled testing program in the automotive service facility. These include:

- comparison of cylinder output by selective shorting of the ignition or smooth running analysis via current engine speed,
- comparison of compression via the shape of the starter current curve,
- determination of mixture distribution performed by selective measurement of the exhaust-gas HC content,
- analysis of the primary and secondary ignition-voltage patterns.

Electronic-system testing

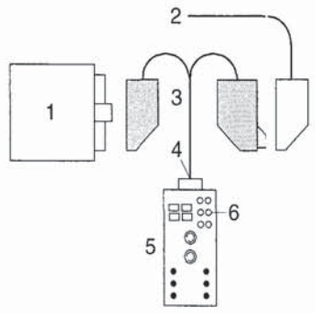
Testing of engine-related electronic systems is carried out using test equipment specifically designed to utilize on-board engine monitoring technology.

The test connection is established by inserting a universal test adapter between the plug and socket at the junction linking the peripheral device with the ECU. If only the peripheral device (sensor, actuator, wiring and power supply) is to be tested, then it is sufficient to connect the tester to the peripheral's plug, while the ECU is also connected for tests embracing dynamic operation. The individual electronic system requires only a single interchangeable, system-specific adapter cable. The program switches on the universal test adapter are then used to establish a logical test sequence specifically tailored to the requirements of the system.

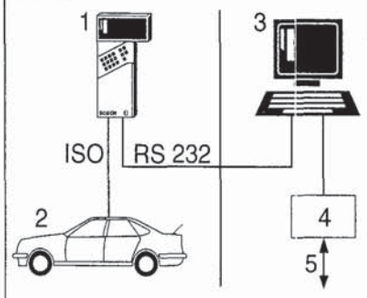
Once connected, the test unit provides a display of both measured values and of signals such as ignition and fuel-injection pulses.

When the ECU remains connected for operational testing, keys can be used to enter simulations of various operating conditions for test-unit evaluation.

Peripheral and functional testing
1 ECU, 2 System wiring harness, 3 System adapter cable, 4 Plug connection, 5 Universal test adapter, 6 Sockets for test unit



System test with ECU diagnostics
1 Test unit, 2 Test vehicle, 3 Information system, 4 Modem, 5 Long-distance data transmission.



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ECU diagnostics

The dominant role being assumed by electronic systems in the vehicle makes it necessary to devote increased attention to the problems associated with service. In addition, because essential vehicle functions are becoming increasingly dependent upon electronics, these systems must satisfy stringent reliability requirements, while emergency default programs are required to deal with system errors.

The solution is to incorporate electronic system-diagnosis functions in the ECU. These rely on the electronic "intelligence" already in place in the vehicle to continuously monitor the system, detect faults, store the fault data and perform diagnostics.

For instance, the ECU carries out its own self-check as follows: Programmed memory chips are provided with test patterns which can be retrieved and used for comparisons. For program memories, a comparison with test sums is employed to ensure that data and programs are correctly stored. The data and address buses are included in the test program.

Sensors are tested for plausibility within specified limits, while open and short circuits are also recognized. Final-control elements can be tested during activation using current-draw limits.

The "off-board test units" used for the evaluation of "on-board diagnostics" employ a communications interface as defined in ISO 9141. The serial port can maintain communications at rates ranging from 10 baud to 10 kbaud. It is designed as a single- or two-wire port, allowing connection of several control units to a central diagnosis plug.

A stimulation address is transmitted to all of the connected ECUs, whereupon each unit recognizes its address and responds by transmitting back a baud-rate recognition word.

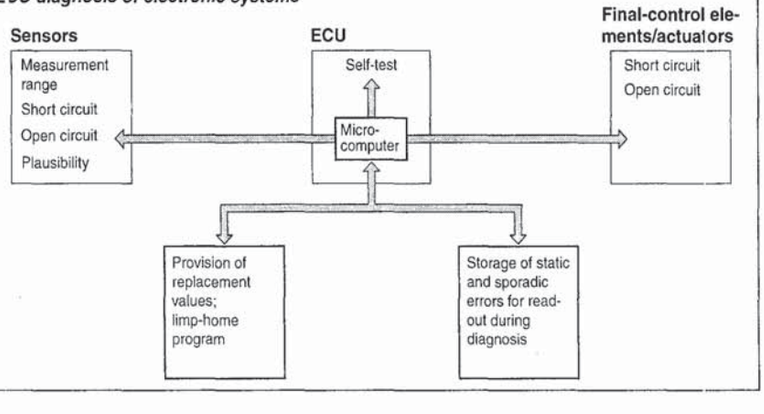
The test unit monitors the period between pulse flanks to determine the sender's baud rate, which it then adopts automatically. The key bytes that follow (assigned by the DIN Motor-Vehicle Committee) specify the protocol for subsequent data communications.

The programmable test unit converts the received data into diagnosis sequences and plain-text information designed specifically for the respective system.

The ECU diagnostic capabilities include:
- identification of system and ECU,
- recognition, storage and readout of static and sporadic malfunctions together with the error path, failure mode and associated parameters,
- readout of current actual values, switching conditions, specifications,
- stimulation of system functions, and
- programming of system variants.

Individual programs for the test unit are stored in plug-in modules, while updates and communications with data systems are supported by a data interface.

ECU diagnosis of electronic systems



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LPG systems

Liquefied petroleum gas

Liquefied petroleum gas assumes a liquid state at pressures of 2...20 bar, depending on the propane/butane ratio and the temperature (see chapter on "Fuels", P. 238).

At the end of the '90s, LPG was being consumed by internal-combustion (IC) engines at a rate of approximately 10.3 million metric tons annually (of which 2.6 million tons were consumed in Europe). Should efforts to utilize the gas contained in petroleum succeed, then these figures could increase exponentially. The mineral-oil tax is a decisive factor in determining the profitability of liquefied petroleum gas.

Natural gas as engine fuel

Reserves of natural gas are extensive. Together with it being used nowhere near as much as crude oil, this fact makes natural gas a very interesting alternative fuel for automotive applications. Both the LPG equipment configurations and the emissions produced would be similar to those for the combination of propane and butane known as LPG (see illustration). Natural gas can be transported in the vehicle either as a high-pressure gas (160...200 bar) or in liquefied form (at –160 °C) in an insulated tank; the disadvantage of the former mode lies in its limited operating range.

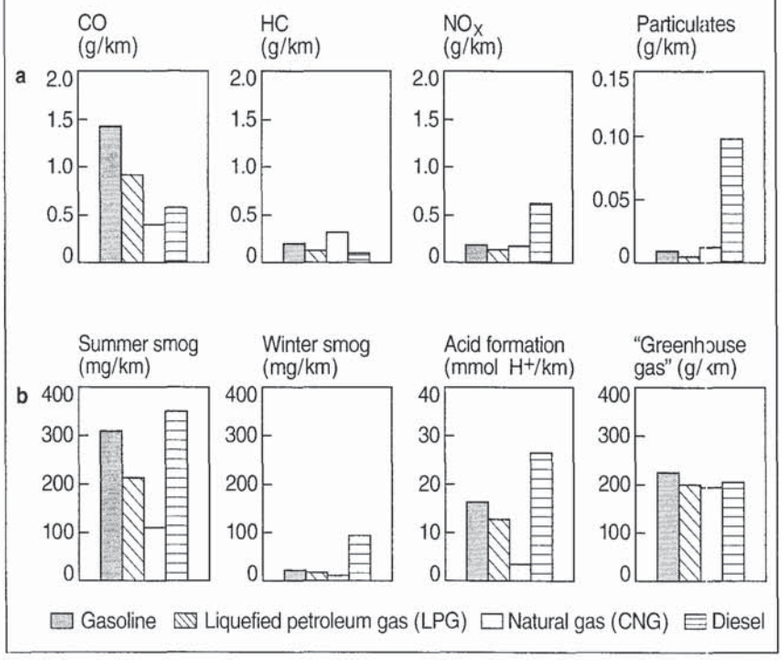
The only real difference between concepts based on natural gas and LPG is the way the gas is transported in the vehicle; actual differences in operating principles are minimal.

Operation on LPG

Any vehicle equipped with an IC engine can be converted for operation on LPG. For the most part, spark-ignition (SI) engines are re-equipped for dual-fuel operation (system can be switched between gasoline and LPG). LPG-powered taxis and buses are

Exhaust emissions and effects with gasoline, LPG, CNG and diesel

Empirical values taken from five spark-ignition and five diesel-engine vehicles, status 1993 in Europe, including indirect emissions (manufacture, transport, etc.).
a) Pollutant emissions, b) Pollutant effect ("greenhouse gas" assessment, status 1998).



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generally set up for single-fuel (LPG only) operation, while regulations require this configuration on gas-powered industrial trucks intended for indoor use. When engines are converted, it should be remembered that they operate as naturally aspirated engines while running on LPG (consumption in liters increases by approx. 25% as compared to gasoline).

Exhaust emissions

Because LPG mixes well with air, emissions (of CO_2 and other components such as polycyclic aromatic hydrocarbons) are substantially lower than those produced by gasoline-burning engines, and are lower even than those produced by fuel-injection engines equipped with three-way closed-loop catalytic converters. LPG contains no lead or sulphur compounds. It's very good combustion characteristics are complemented by excellent mixture formation and distribution properties. These characteristics are even more significant at low temperatures.

Advantages

- Extremely economical for drivers who cover large distances.
- Assuming the same level of technology (electronic control systems, etc.), emissions from an LPG engine are substantially lower than those achieved with gasoline or diesel fuel.

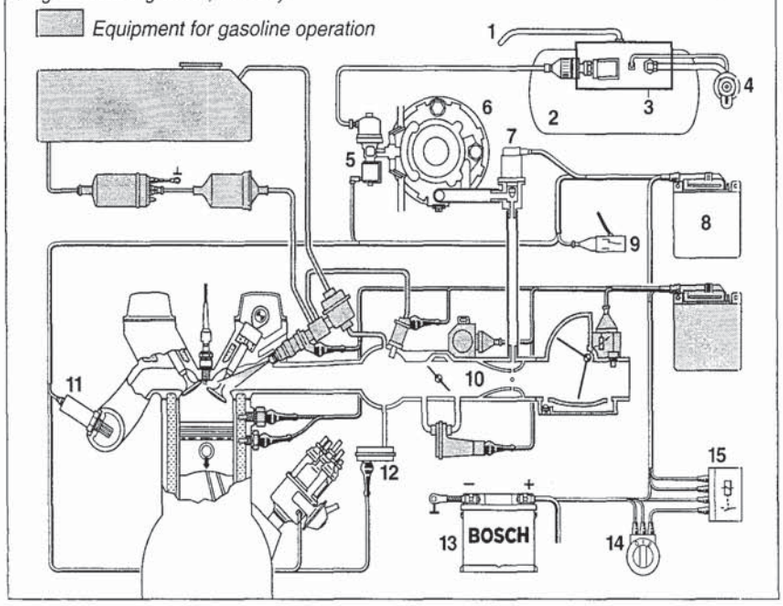
Disadvantages

- Lower cruising ranges and increased fuel volumetric consumption compared to gasoline (although actual energy use is not higher than with gasoline, lowest with diesel engine).
- Special safety precautions are necessary, as LPG is pressurized.
- Pressurized gas cylinders require a lot of space, as the actual capacity is only 80% of cylinder volume (the remainder serving as expansion room for the gas).

LPG system

In Germany, professional installation of

Schematic diagram of an LPG system (carburetor principle)
1 Ventilation line for tank fittings, 2 LPG tank, 3 Housing with tank fittings, 4 External filler valve (designed to interrupt refueling at 80% of container capacity), 5 Gas shut-off valve, 6 Pressure regulator for evaporator, 7 Servomotor for gas control, 8 ECU, 9 Gas/gasoline changeover switch, 10 Venturi mixing device, 11 Lambda oxygen sensor, 12 Vacuum sensor, 13 Battery, 14 Ignition/starting switch, 15 Relay.



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the LPG system in a specialist workshop is followed by a trip to the TÜV or TÜA (German inspection authorities) to secure operating approval. This inspection is based on the "Guidelines for inspecting vehicles with engines powered by liquefied gases" as issued by the Federal Minister of Transportation. Uniform regulations for Europe are being drawn up.

A modern LPG system will incorporate the following components:
- LPG tank,
- external filler valve designed to interrupt refueling at 80 % of container capacity,
- flow-interrupt valve,
- evaporator pressure regulator with cooling system,
- venturi mixing unit/injector,
- electronic control unit (ECU),
- servomotor for controlling gas flow,
- switch for alternating between LPG and gasoline operation.

The LPG flows from the LPG tank to the evaporator pressure regulator, where it is vaporized and its pressure reduced. The ECU processes the signals from the Lambda (O_2) and vacuum sensors, which serve as references for controlling the servomotor used to regulate the flow of liquefied gas to the venturi mixing unit.

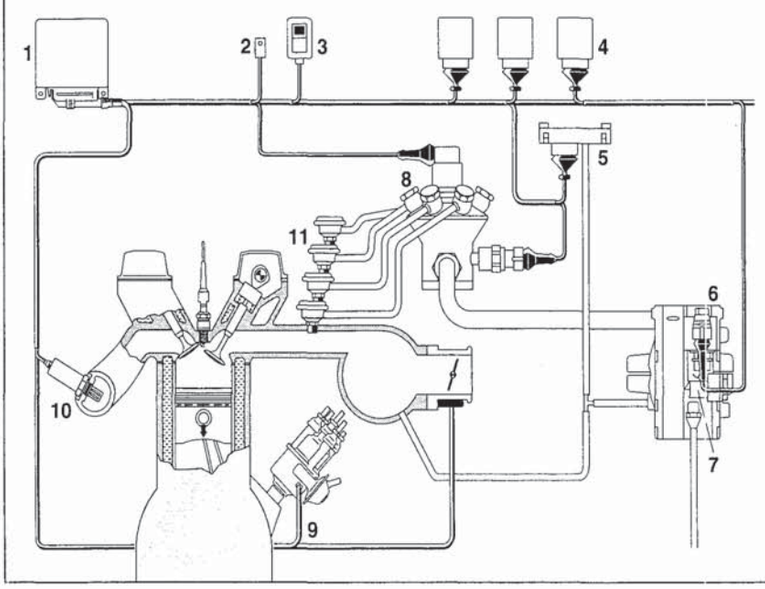
The flow-interrupt valve shuts immediately when the ignition is switched off. A gasoline/LPG switch installed in the instrument panel allows the operator to alternate between the two fuels.

LPG tanks

As LPG tanks are used to store pressurized gas, they are subject in Germany to the "TRG 380" technical regulations. At the factory, each tank receives official technical approval with certification.

They are equipped with an external filler valve (designed to interrupt refueling at 80 % of container capacity) as well as an electromagnetic discharge valve, and have a capacity of 40 ... 128 l for passenger cars.

Schematic diagram of an LPG system (injection principle)
1 ECU, 2 Diagnosis plug, 3 Fuel selection switch, 4 Relay, 5 Air-intake pressure sensor, 6 Evaporator pressure regulator, 7 Flow-interrupt valve, 8 Distributor with step motor, 9 RPM signal, 10 Lambda (O_2) sensor, 11 Gas injector nozzle.



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Natural-gas operation of spark-ignition engines

Against the backdrop of worldwide efforts to reduce CO_2 emissions and to comply with ever more stringent exhaust emission limits, natural gas is gaining increasing significance as an alternative fuel.

Properties and storage of natural gas

The main component of natural gas is methane (CH_4), making up 80...99 %. The remainder consists of inert gases such as carbon dioxide, nitrogen and other low-order hydrocarbons. A differentiation is made between L-gas (80...90 % methane) and H-gas (>90 %) depending on the gas quality.

Natural gas can be stored both in liquid form at −162 °C as LNG (Liquefied Natural Gas) or in compressed form at pressures of up to 200 bar as CNG (Compressed Natural Gas). In view of the great expense involved in storing the gas in liquid form, natural gas is used in compressed form in virtually all applications.The low energy density of natural gas is a particular disadvantage, making large storage tanks necessary. Metal-hydride storage tanks represent a further storage option, although they are not used for reasons of cost.

Mixture formation

In most systems, gas is injected into the intake manifold as in conventional multipoint gasoline-injection systems. A low-pressure common rail supplies the injector valves that inject intermittently into the intake manifold. Mixture formation is simplified by the completely gaseous supply of fuel, as natural gas does not condense on the intake manifolds and does not form a film on the walls. This has a favourable effect on emissions, particularly during the warm-up phase.

The output of the natural-gas engine is approx. 10...15 % lower than that of the gasoline engine due to the lower fuel mass necessary for stoichiometric combustion (17.2 : 1 ratio) as well as a lower volumetric efficiency due to the injected natural gas. Higher compression can boost performance while simultaneously increasing efficiency. The extremely high antiknock quality of natural gas (120 RON) enables very high compressions of approx. 13 : 1 (8 : 1 for regular gasoline). As a result, the natural-gas engine is ideally suited for turbocharging. As piston displacement decreases, efficiency increases due to additional de-throttling and reduced friction.

A further improvement in efficiency can be achieved by lean-mixture operation up to $\lambda = 1.7$. A lean mixture reduces combustion temperatures while at the same time further de-throttling the engine.

Emissions

Natural-gas vehicles are characterized by low CO_2 emissions due to the favourable hydrogen/carbon ratio (H/C ratio) of almost 4 : 1 (gasoline: 2.3 : 1) and the resulting shift in the main combustion products CO_2 and H_2O. Apart from the virtually particle-free combustion, in conjunction with a three-way closed-loop catalytic converter only very low levels of the pollutants NO_X, CO and NMHC ("non-methane hydrocarbons": the sum of all hydrocarbons minus methane) are emitted. Methane is classified as nontoxic, and is therefore not considered to be a pollutant.

In lean mixture mode the NO_X emissions are higher than in $\lambda = 1$ mode with a three-way catalytic converter. In the same way as in gasoline operation, this disadvantage can be largely eliminated by using more expensive exhaust-gas treatment methods (e.g. NO_X catalytic converter).

Natural-gas-engine applications

In view of their limited range, natural-gas engines are used almost exclusively in local public transportation fleets (e.g. buses and taxis). Bivalent systems which can be easily changed over from natural gas to gasoline operation are used primarly in passenger cars.

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Operation on alcohol (spark-ignition engines)

The limited availability of fossil fuels has led to an increased effort to develop engines and injection systems capable of using alcohols such as ethanol and methanol as alternative fuels ("Alternative fuels", P. 246). Due to its non-availability, virtually the only place where ethanol is used is Brazil. In the US (and in California in particular), increasing attention is being focused on methanol, which generates lower emissions: reduced NO_X and CO_2 along with reduced ozone and smog formation.

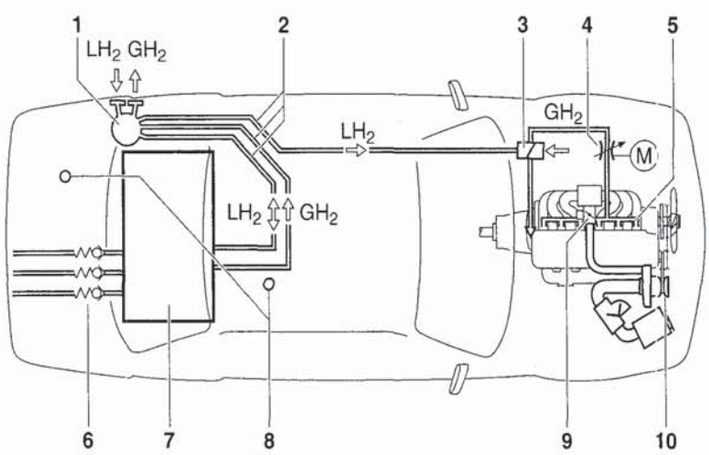
In the absence of comprehensive methanol-distribution networks to ensure universal availability, engines and engine-control systems must be designed for flexible dual-fuel operation (ranging from pure gasoline to max. 85% methanol). Alcohol places special, particularly critical demands upon engines and fuel-delivery components. Moisture, acids and gums contained in the fuel pose a hazard to metals, plastics and rubber. Because methanol has a high antiknock quality, engines designed to run exclusively on methanol have substantially higher compression ratios than gasoline engines, making them more efficient. On the other hand, methanol's low calorific value means that fuel consumption is almost doubled, necessitating higher fuel supply rates, greater tank volumes and special injectors.

Suitable Lambda (O_2) sensors can be employed for optimal emissions control with a catalytic converter. Special lubricants are able to maintain long-term stability in the face of this aggressive fuel and its combustion products.

Pilot control of the fuel mixture is facilitated by a fuel sensor which sends a signal to the ECU reflecting the proportion of methanol in the fuel.

Suitable programs implement the necessary mixture and ignition corrections applicable to a particular set of engine operating characteristics.

Hydrogen-powered passenger car with spark-ignition engine (BMW 735i)
LH_2 Liquid hydrogen, GH_2 Gaseous hydrogen. 1 Valve block for LH_2 fueling and GH_2 supply (vacuum-insulated), 2 Hydrogen lines, vacuum-insulated, 3 LH_2 Evaporator, 4 Metering valve for regulating power with electronic control, 5 Hydrogen injectors, 6 Overcurrent and safety valves, 7 Liquid-hydrogen tank with vacuum super-insulation, 8 Hydrogen sensors for automatic leak monitoring, 9 Throttle valve for gasoline operation with electronic control, 10 Variable-speed centrifugal supercharger.



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Operation on hydrogen (spark-ignition engines)

Although the production of hydrogen, the required infrastructure and refueling all pose difficult problems, technically feasible solutions are on the horizon. Producing hydrogen by means of electrolysis necessitates sufficient quantities of electrical power in the form of solar energy or nuclear power.

Storing hydrogen in the vehicle

Gaseous storage in pressurized tanks
High pressures (300 bar) are required for storage in gaseous form. This results in high weight along with safety risks.

Liquid storage (cryogenic tank)
Liquid storage represents the best alternative with regard to both weight and energy density (present operating range approx. 300 km). The extremely low temperature required (–253 °C) places substantial demands on thermal insulation. Residual heat causes loss of hydrogen via the safety valves at a rate of about 2 % per day when the vehicle is parked. An electric evaporator maintains the specified tank pressure during operation.

Metal-hydride tanks
Hydrides are produced as hydrogen is absorbed by a metallic powder. This is an exothermic process, i. e. heat must be dissipated during fueling. There are no storage losses.

The disadvantages associated with the low energy density (range: 120 km) and high materials costs are to a degree offset by uncomplicated safety technology.

Methylcyclohexanol storage
This type of storage employs a catalyst to dehydrate the hydrogenous methylcyclohexanol at 500 °C. The by-products are hydrogen and recyclable toluene.

Mixture formation

Regardless of storage mode, up to now all systems inject gaseous hydrogen into the intake manifold. Although a number of advantages could be gained by injecting extremely low-temperature hydrogen directly into the combustion chamber (improved charge for higher output, cool mixture for low NO_X emissions, no danger of backfiring), the short injector-valve service life means that this type of system is not likely to appear in production in the near future.

Current external mixture-formation concepts rely on a continuous-injection system in which a central electric metering valve and a hydrogen distributor conduct the vaporized hydrogen to the individual intake tracts.

Backfiring into the intake passage is prevented by lean mixtures or supplementary water injection. A supercharging device can be used to compensate for a portion of the power loss associated with lean operation.

One alternative, based on intermittent sequential injection of hydrogen into the intake manifold, is currently in the development stage.

This system's unlimited range of options for injection timing allows it to inhibit backfiring almost completely, even with rich mixtures.

Injector valve and electronic-control system must operate with extreme precision and short valve-opening times the technical requirements are thus substantial.

Emissions

During combustion, pure hydrogen (H_2) oxidizes to form water (H_2O). No CO_2 is produced by the combustion process. Provided no fossil fuels are used in its production, H_2 is thus the only fuel which can be used to avoid all CO_2.

Electric drive is the only alternative which can make a similar claim. Future NO_X emission limits can be met by lean mixtures or a system for catalytic control of emissions (still to be developed).

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Exhaust emissions from spark-ignition engines

Combustion products

Complete combustion

The by-products of complete gasoline combustion are carbon dioxide and water.

Incomplete combustion

Unburned hydrocarbons:
C_nH_m (paraffins, olefins, aromatic hydrocarbons)
Partially-burned hydrocarbons:
$C_nH_m \cdot CHO$ (aldehydes),
$C_nH_m \cdot CO$ (ketones),
$C_nH_m \cdot COOH$ (carboxylic acids),
CO (carbon monoxide).
Thermal crack products and derivatives:
C_2H_2, C_2H_4, H_2 (acetylene, ethylene, hydrogen, etc.), C (soot), polycyclic hydrocarbons.

Combustion by-products

From atmospheric nitrogen: NO, NO_2 (nitrogen oxide). From fuel impurities: sulfurous oxides.

Oxidants

The following oxidants are produced when exhaust gas is exposed to sunlight: organic peroxides, ozone, peroxy-acetyl-nitrates.

Properties of exhaust-gas components

Major components

The major proportion of the exhaust gas is composed of the three components nitrogen, carbon dioxide and water vapor. These are nontoxic. However, emissions of CO_2 – largely the result of fuel combustion – are becoming more and more important due to their contribution to the "greenhouse effect".

Minor components

Carbon monoxide CO: A colorless, odorless and tasteless gas. Inhalation of air with a volumetric concentration of 0.3% carbon monoxide can result in death within 30 minutes. The CO-content of the exhaust gas from spark-ignition engines is especially high at idle. It is therefore imperative that the engine never be run in a closed garage!
Nitrogen monoxide NO: A colorless, tasteless and odorless gas; in air it is gradually converted into NO_2. Pure NO_2 is a poisonous, reddish-brown gas with a penetrating odor. The concentrations found in exhaust gases and in extremely polluted air can induce irritation in mucous membranes. NO and NO_2 are generally referred to collectively as oxides of nitrogen NO_X.
Hydrocarbons are present in exhaust gases in a variety of forms. When exposed to sunlight and nitrous oxide, they react to form oxidants which irritate the mucous membranes. Some hydrocarbons are considered to be carcinogenic.
Particulates (particulate matter) in accordance with American regulatory practice, are defined as all substances (except unbound water) which under normal conditions are present in exhaust gases in a solid (ash, carbon) or liquid state.

Mixture formation

The fuel used in spark-ignition (SI) engines is more volatile than diesel fuel, while the air-fuel mixing process prior to combustion also extends over a longer period than in a diesel engine. The result is that spark-ignition engines operate on a more homogenous mixture than their diesel counterparts. Spark-ignition engines run on a mixture in the stoichiometric range ($\lambda = 1$). Diesel engines, on the other hand, always operate with excess air ($\lambda > 1$). An air deficiency ($\lambda < 1$) results in increased soot, CO and HC emissions.

Combustion characteristic

The combustion characteristic defines the combustion as a function of time, and applies the ratio of the already combusted fuel to the fuel awaiting combustion.

Here, the efficiency, the combustion temperature, and therefore the fuel consumption and NO_X emissions are particularly influenced by the position of maximum energy conversion referred to piston TDC.

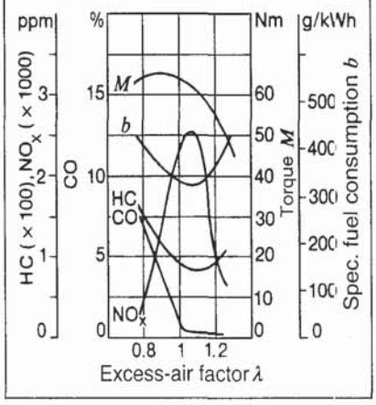
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In the spark-ignition engine, the ignition timing initiates the combustion process, while the start of injection initiates combustion in a diesel engine.

Emissions control

The methods used to influence the composition of the SI-engine exhaust gases are divided into two basic categories: engine-design measures and exhaust-gas treatment. The selection of procedures to be employed in any given country is determined by the legal regulations in force there. The major industrial nations, with their important markets, have been moving toward implementation of the stringent American exhaust emissions regulations (or have already implemented them). Compliance with this legislation necessitates using emission control systems which incorporate the 3-way catalytic converter, a principle which has already proven itself in the USA.

The excess-air factor λ influences:
Exhaust-gas composition (CO, NO_x, HC), torque (M) and specific fuel consumption (b). The values apply to the part-load range of a spark-ignition engine operating at a constant moderate speed and cylinder charge (λ values implemented in the vehicle engine range from approx. 0.85...1.15, depending on the operating point).



Engine-design measures

Setting the A/F ratio

The excess-air factor λ of the A/F mixture delivered to the engine has a dominating effect on the composition of the exhaust gas. The engine produces its maximum torque at approximately λ = 0.9; thus this ratio is generally programmed for full-load operation. A certain level of excess air is required for favourable fuel consumption. This coincides with the setting for low CO and HC emissions; oxides of nitrogen (NO_x) however, are at a maximum at this ratio. Excess-air factors of λ = 0.9...1.05 are selected at idle. An excessively lean mixture results in the engine's lean misfire limit (LML) being reached or exceeded, and as the mixture becomes progressively leaner, misfiring causes a rapid increase in HC emissions. For overrun (trailing-throttle) operation, it is frequently necessary to enrich the mixture (λ = 0.9) in order to maintain an ignitable mixture, while at the same time air is added to avoid excessive manifold vacuum.

Yet another option for overrun operation is to completely interrupt the fuel supply to the engine at speeds above idle (overrun fuel cutoff). Precise mixture control (P. 465) is achieved by the use of electronic fuel-injection systems (EFI).

Mixture formation

Mixture formation embraces not only setting the correct A/F ratio, but also the quality of the A/F mixture which actually enters the combustion chamber. The fuel's homogeneity, its stratification patterns, and its temperature at the instant of ignition are all essential factors in determining combustibility and combustion characteristic, with consequent effects upon exhaust-gas composition. Homogenous mixtures and controlled stratification (rich mixture at the spark plug, lean mixture in the vicinity of the combustion-chamber walls) are examples of two different development options.

Uniform distribution

Maximum engine efficiency can only be achieved if every cylinder is operated with the same excess-air factor. This necessitates a system which ensures that both air and fuel are distributed evenly among the individual cylinders.

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Exhaust-gas recirculation (EGR)
Exhaust gas can be conducted back to the combustion chamber to reduce peak combustion temperatures. Higher combustion temperatures induce an over-proportional increase in NO_X formation, and because exhaust-gas recirculation (EGR) reduces combustion temperatures, it represents a particularly effective means of controlling NO_X emissions. Extensive optimization of EGR can also lead to a reduction in fuel consumption. EGR can be implemented in either of two ways:
- Internal exhaust-gas recirculation is achieved with appropriate valve timing (overlap),
- External exhaust-gas recirculation employing controlled EGR valves.

Valve timing
Large valve overlaps (early opening of the inlet valve) increase the internal exhaust-gas recirculation, and can therefore help to reduce NO_X emissions. However, since the recirculated exhaust gas displaces fresh A/F mixture, early opening of the inlet valve also leads to a reduction in the maximum torque. In addition, excessively high exhaust-gas recirculation, particularly at idle, can lead to combustion misfire which in turn causes an increase in HC emissions. An optimum can be found with variable valve timing (P. 387), in which the valve timing is varied as a function of the operating point.

Compression ratio
It has long been recognized that the enhanced thermal efficiency associated with high compression ratios represents an effective means of improving fuel economy. However, the increase in peak combustion temperature also results in higher NO_X emissions.

Combustion-chamber design:
Low HC emissions are best achieved with a compact combustion chamber featuring a minimal surface area and no recesses. A centrally-located spark plug with short flame travel produces rapid and relatively complete combustion of the mixture, resulting in low HC emissions and reduced fuel consumption. Induced combustion-chamber turbulence also provides rapid combustion. Combustion chambers optimized in this way feature favourable HC emissions at $\lambda = 1$ and improve the engine's lean-mixture operation.

A thoroughly optimized combustion chamber design coupled with external measures (such as intake swirl) produce a lean-burn engine capable of running on mixtures in the range of $\lambda \approx 1.4...1.6$. Although the lean-burn engine features low exhaust emissions and excellent fuel economy, it does require catalytic exhaust-gas treatment in order to meet the most stringent emissions limits for CO, HC, and NO_X. Particularly due to the fact that developments in the aftertreatment of NO_X in lean exhaust gas are still in their infancy, the lean-burn engine has up to now only been successful in Europe and Japan, and only in the case of a few models using lean/mix concepts which compromised between emissions and fuel consumption. A new way of substantially improving the lean-running behavior of the spark-ignition engine is to inject fuel directly into the combustion chamber with stratified charge. In this case, accumulator-type catalytic converters are favoured for exhaust gas treatment.

The lean-running characteristics achieved by combustion-chamber design and induced turbulence can also be applied to implement high EGR rates for designs with $\lambda = 1$. Here, a reduction in fuel consumption can be easily achieved with exhaust-gas treatment, but not to the same degree as in lean-burn engines.

Ignition system
The design of the spark plug, its position in the combustion chamber, together with spark energy and spark duration all exercise a major influence on the ignition, the mixture's combustion characteristic, and therefore on emissions levels. The significance of these factors increases with the leanness ($\lambda > 1.1$) of the mixture. Ignition timing exerts a decisive effect on both exhaust emissions and fuel economy. Using the firing point for optimum fuel economy as a baseline, the timing is retarded to a point at which the exhaust valve opens before the combustion process is completed. Excess oxygen in the exhaust system can cause a thermal post-reaction to

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occur. Although this process reduces unburned hydrocarbons, it also leads to increased fuel consumption. NO_X emissions are low due to the low combustion-chamber temperatures. Fuel consumption and NO_X and HC emissions increase when the firing point is set earlier compared to the optimum.

Crankcase ventilation (blowby)

The concentration of hydrocarbons in the crankcase can be many times that found in the engine's exhaust gases. Control systems conduct these gases to a suitable point in the engine's intake tract, from where they are drawn into the combustion chamber for burning. Originally, these gases were allowed to escape untreated directly into the atmosphere; today, crankcase emission-control systems are a standard legal requirement.

Exhaust-gas treatment

Thermal afterburning

Before today's catalytic treatment of exhaust emissions became standard, initial attempts to reduce emissions utilized thermal afterburning. This method employs a specific residence time at high temperatures for burning the exhaust-gas components which failed to combust during normal combustion in the engine cylinders. In the rich range (λ = 0.7...1.0), the process must be supported with supplementary air injection (secondary air). In lean-burn engines (λ = 1.05...1.2), the residual oxygen in the exhaust gas supports the afterburning process.

In the past, mechanical pumps driven by belts directly from the engine were used for secondary-air injection. Since such air injection is only required during the engine's warm-up period, these pumps are switched using electromagnetic couplings. Lower-priced blower pumps powered by electric motors are quickly superseding the former mechanical versions.

Due to its lack of potential, particularly in maintaining low NO_X limits, thermal afterburning alone is currently considered to be of little significance. It can be used, however, to reduce emissions of HC and CO during the operating phase in which the catalytic converter has not yet reached its operating temperature. Because it greatly reduces the time taken for the catalytic converter to reach its operating temperature, thermal aftertreatment with air injection during the engine warm-up phase in combination with catalytic aftertreatment will play a major role in achieving compliance with more stringent emission limits in the future.

Catalytic afterburning

The catalytic converter is composed of a carrier substrate, which serves as a base for the catalytic material, mounted within a housing using vibration-proof, heat-insulated supports. Granulate and ceramic or metallic monolith structures are employed as substrate materials. The suitability of the monolith structure for automotive applications has been demonstrated over the course of an extended development period. It offers the following advantages: maximum utilization of catalytic surface, durability combined with physical strength, low thermal retention and limited exhaust back-pressure. The active catalytic layer consists of small quantities of noble metals (Pt, Rh, Pd), and is sensitive to lead. For this reason, it is essential that engines with catalytic converters be run on unleaded fuel exclusively, as lead destroys the effectiveness of the active layer. The conversion rate is largely a function of operating temperature; no significant conversion of pollutants takes place below an operating temperature of approx. 250 °C. Operating temperatures of approx. 400...800 °C provide ideal conditions for maximum efficiency and extended service life.

Installing the catalytic converter directly adjacent to the engine means that it more quickly reaches operating temperature since the exhaust gases are hotter. This results in improved efficiency, but at the cost of high thermal stresses. Because the maximum permissible operating temperature lies just slightly above 1000 °C, the units are generally installed at a less critical location under the floor of the vehicle. Engine malfunctions, such as ignition miss, can cause the temperature in the catalytic converter to increase to the point where the substrate melts, resulting in de-

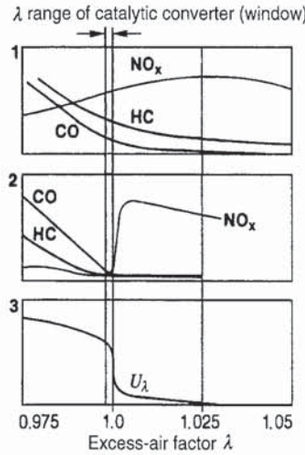
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struction of the unit. This must be prevented by using reliable, maintenance-free ignition systems. Oxidation-type catalytic converters oxidize CO and HC either by utilizing the excess air supplied by lean engine mixtures or by relying on secondary-air injection. Reduction-type converters on the other hand operate with air deficiency, and thus without air injection, to reduce NO_X levels.

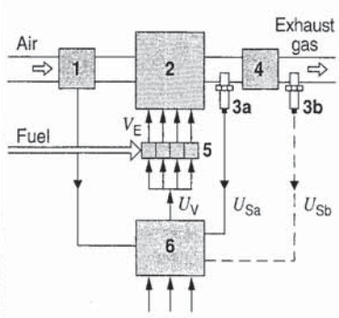
The reduction- and oxidation-type catalytic converters can also be combined in series to produce a dual-bed catalytic converter. This device uses air injection between the two converters to reduce not only NO_X emissions but also HC and CO levels. Its disadvantages include a more complex design (two converters, air injection) and the need to operate the engine in a range with higher fuel consumption ($\lambda = 0.9$).

The three-way or selective catalytic converter with lambda closed-loop control has proven to be the most effective concept for exhaust-gas aftertreatment. It is capable of providing the required reduction of all three pollutants provided the engine is operated with a stoichiometric mixture. The "window" for treatment of the three toxic components is narrow; this means that an open-loop fuel-metering system cannot be used with this concept.

Catalytic converter efficiency as a function of the excess-air factor λ
1 Exhaust emissions upstream of 3-way catalytic converter, 2 Exhaust emissions downstream of 3-way catalytic converter, 3 Electric signal from Lambda oxygen sensor.
U_λ Sensor voltage.



Functional schematic of the Lambda closed-loop control
1 Air-mass meter, 2 Engine, 3a Lambda oxygen sensor 1, 3b Lambda oxygen sensor 2 (only if required), 4 Catalytic converter, 5 Fuel injectors, 6 ECU.
U_S Sensor voltage, U_V Injector triggering voltage, V_E Injected fuel quantity.



Lambda closed-loop control

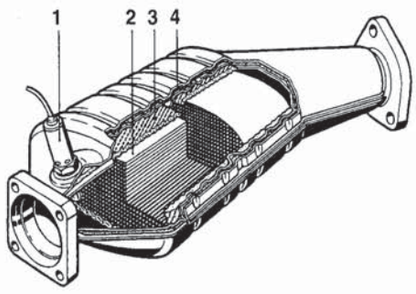
In the US, Europe and Japan, catalytic aftertreatment of the exhaust gas using a closed-loop three-way catalytic converter has proven to be a highly effective means of meeting the current low emission limits for CO, NO_X and HC.

Approximately 14.7 kg of air are required for complete combustion of 1 kg of gasoline. The excess-air factor λ (Lambda) is used to define this A/F ratio; it is the ratio of the actual A/F ratio to the stoichiometric A/F ratio. Two main control concepts for exhaust gas optimization are used in the spark-ignition engine:

Closed-loop control for λ = 1
This concept is the most effective for minimizing pollutants. The engine must be operated within a very narrow range in which $\lambda = 1 \pm 0.005$ (catalytic-converter window). Such precision can only be achieved with precise closed-loop control of the A/F mixture with a Lambda oxygen sensor installed upstream of the converter. A second Lambda sensor down-

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Two-bed, 3-way catalytic converter with Lambda oxygen sensor
1 Lambda oxygen sensor, 2 Ceramic monolith, 3 Wire screen, 4 Heat-resistant double shell.



stream of the catalytic converter increases the precision even further.

Closed-loop control for λ > 1 (lean-burn control)

The main advantage here lies in the reduction of fuel consumption as a result of lean-burn (non-throttled) operation. The success of this concept depends to a high degree upon the availability of catalytic converters which are able to reduce NO_X emissions during lean-burn operation. The lean-misfire limit (LML) for spark-ignition engines in the lean-burn range is reached at $\lambda \approx 1.7$ regardless of engine design measures.

Lambda oxygen sensors

Zirconium-dioxide sensor (λ = 1)

This sensor operates according to the principle of a galvanic oxygen concentration cell with solid electrolyte whose ceramic is made of zirconium dioxide and yttrium oxide. This oxide mix is a practically pure oxygen-ion conductor and separates the exhaust gas from the surrounding air.

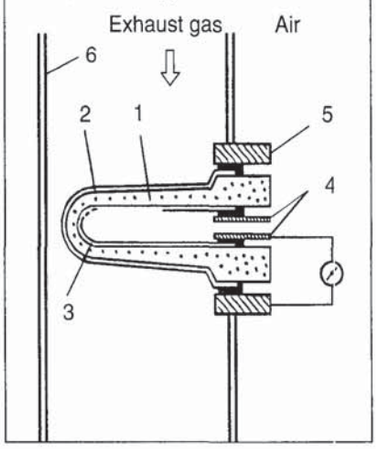
An electrical voltage is generated across the platinum-cermet electrodes in accordance with the Nernst principle.

$$U_s = R \cdot \frac{T}{4F} \cdot \ln\left(\frac{p_{O_2}''}{p_{O_2}'}\right)$$

R general gas constant, F Faraday constant, T absolute temperature, p_{O_2}'' oxygen partial pressure of reference load, p_{O_2}' oxygen partial pressure in exhaust gas.

The catalytically active electrodes must be in thermodynamic balance before measurement of the oxygen concentration can be applied for reaching conclusions on the Lambda values. In this case, the sensor's characteristic shows a jump at $\lambda = 1$. When high concentrations of exhaust-gas components which are not in thermodynamic balance are incompletely converted, the sensor's characteristic shows shifts in the excess-air factor λ.

Zirconium-dioxide oxygen sensor
1 ZrO_2 solid electrolyte, 2 Pt outer electrode, 3 Pt inner electrode, 4 Contacts, 5 Housing contact, 6 Exhaust pipe.



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These must be compensated for by design measures taken at the engine. The sensor characteristic's dependence on temperature can be minimized by heating the sensor electrically.

Resistive sensor

Due to the change in the O_2 vacancy concentration of the oxide, oxidic semiconductors such as titanium oxide or strontium titanate change their volume conductivity.

The temperature-dependency of the conductivity is superimposed upon this effect, so that the determination of the Lambda value depends upon the quality of the required temperature control – open-loop or closed-loop (by varying the amount of heat generated).

Lean sensor

In its lean range (λ in the vicinity of 1), the "voltage-jump" (zirconium-dioxide) sensor as described above has only been used to a limited extent. Special measures taken to stabilize the sensor, together with the use of a high-capacity heater (18 W) permit operation to approx. $\lambda \leq 1.5$. The lean sensor using the limit-current principle permits the measurement of any values above $\lambda = 1$.

O_2 ions are pumped from the cathode to the anode when an external voltage is applied across two electrodes deposited on a ZrO_2 ceramic substrate. Since a diffusion barrier prevents the flow of O_2 molecules from the exhaust gas to the cathode, current saturation is reached above a given pump-voltage threshold value. By approximation, the resulting limit current is proportional to the oxygen concentration. This sensor principle is particularly suitable for lean-burn concepts. However, in lean/mix concepts in which a control setpoint $\lambda = 1$ is often desired, the broadband sensor is more suitable.

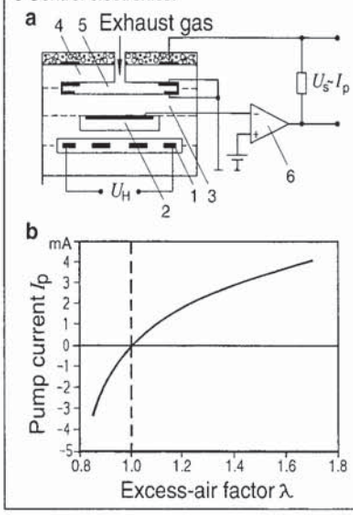
Broadband sensor

This is a combination of the lean sensor using the limit-current principle and the zirconium-dioxide sensor (Nernst oxygen concentration cell). As a two-cell sensor, and in combination with closed-loop control electronics, this sensor generates a clear signal which rises linearly within a broad lambda range ($0.7 < \lambda < 4$).

The pump cell and the concentration cell are made of ZrO_2, and each is coated with 2 porous platinum electrodes. The cells are so arranged that there is a measuring gap of 10...50 µm between them. This measuring gap is connected to the surrounding exhaust-gas atmosphere through an exhaust-gas opening in the solid electrolyte, and at the same time represents the diffusion barrier which determines the limit current. An electronic circuit controls the voltage applied to the pump cell so that the composition of the exhaust gas in the measuring gap remains constant at $\lambda = 1$. This corresponds to a voltage at the concentration cell of U_N = 450 mV. With lean exhaust gas, the pump cell pumps the oxygen from the measuring gap to the outside. On the other hand, when the exhaust gas is rich, the oxygen is pumped into the measuring gap from the surrounding exhaust gas (by decomposition of CO_2 and H_2O) and the direction of current flow is reversed. The pump current is proportional to the oxygen concentration or oxygen requirement. An integrated heater maintains an operating temperature of at least 600 °C.

Broadband sensor

a) Schematic design, b) Pump current as function of the excess-air factor λ. I_p Pump current, U_H Heater voltage, U_S Sensor voltage. 1 Heater, 2 Air reference, 3 Nernst cell, 4 Pump cell, 5 Diffusion barrier, 6 Control electronics.



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Types of control

Two-step control

The zirconium-dioxide sensor as described above with its voltage-jump characteristic at $\lambda = 1$ is suitable for two-step controls. A manipulated variable composed of the voltage jump and the ramp changes its direction of control for each voltage jump which indicates a change from rich/lean or lean/rich. The typical amplitude of this manipulated variable has been set to the 2...3 % range. This results in a limited controller response which for the most part is determined by the sum of the dead times.

This sensor's typical "false measurement", caused by variations in exhaust-gas composition, can be compensated for by selective control. Here, the manipulated-variable curve is designed to incorporate an intended asymmetry, whereby the retention of the ramp value for a controlled dwell time t_v following the sensor's voltage jump is the preferred method.

Two-step control with reference sensor downstream of the catalytic converter

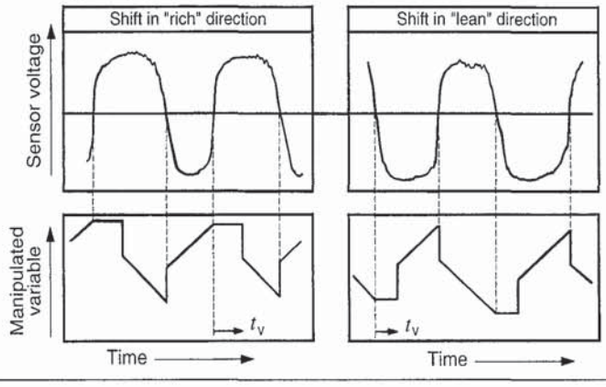
The effect of the disturbance on the accuracy at the voltage-jump point at $\lambda = 1$, as described above, has been minimized by a modified surface coating. Nevertheless, aging and environmental influences (poisoning) still have an effect. A sensor downstream of the catalytic converter is subjected to these influences to a considerably lesser extent. The principle of the two-step control is based upon the fact that the controlled rich or lean shift is changed additively by means of a "slow" corrective control loop.

The resulting long-term stability is of decisive importance for compliance with the exhaust-gas legislation now coming into force.

Continuous-action control with broadband sensor

The defined dynamic response of a two-step control can only be improved when the deviation from $\lambda = 1$ is actually measured. With the broadband sensor, it is possible to achieve continuous-action $\lambda = 1$ control with a stationary, very low amplitude together with high dynamic response. This control's parameters are calculated and adapted as a function of the engine's operating points. Above all, with this type of Lambda control, the unavoidable offset of the stationary and instationary pilot control can be compensated for far more quickly. If demanded by certain engine operating conditions (for instance, warm-up), further optimisation of the exhaust-gas emission applies the potential inherent in the control setpoints $\lambda \neq 1$, for instance, in the lean range. The expansion to a lean-mixture control system, with the advantages and disadvantages already dealt with, merely necessitates the definition of other control setpoints.

Manipulated-variable curve with closed-loop-controlled Lambda shift (two-step control)
t_v dwell time following voltage jump.



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Testing exhaust and evaporative emissions

Test program

In order to precisely determine a passenger car's emission levels, the vehicle must be tested in an emissions test cell under conditions which accurately simulate actual driving conditions. Compared to on-road driving, operation in the test cell offers the advantage of allowing tests to be conducted at precisely predefined speeds without having to take traffic flow into consideration. This is the only means of ensuring that individual emissions tests remain mutually comparable.

The vehicle to be tested is parked with its drive wheels on rollers whose rotational resistance can be adjusted to simulate friction and aerodynamic drag. Inertial masses can be added or removed to simulate the weight of the vehicle. A blower mounted a short distance in front of the vehicle provides the necessary cooling.

The measurement of emission levels is based on a simulated driving pattern which progresses through a precisely defined driving cycle incorporating various vehicle speeds. The exhaust gases produced during this driving cycle are collected for analysis of the pollutant masses at the end of the cycle (see illustration).

The methods for collecting the exhaust gases, and the procedures for determining emissions, have largely been standardized in the various countries, but the driving cycles have not. In some countries, regulations governing exhaust emissions are supplemented by limits placed on evaporative losses from the fuel system.

Chassis dynamometer

In order to compare vehicle emissions, the speeds and forces acting on the vehicle during simulation on the chassis dynamometer and on the road must agree with respect to time. It is necessary to simulate the vehicle's moments of inertia, its rolling resistance, and its aerodynamic drag. For this purpose, eddy current brakes or DC motors produce a suitable speed-dependent retarding force. This force acts on the rollers, and must be overcome by the vehicle. Rapid couplings of various sizes are used to attach inertial masses to the rollers, thus simulating vehicle weight. It is imperative that the curve of braking load versus vehicle speed, and the required inertial masses, be maintained precisely. Deviations result in measuring errors. Ambient conditions such as atmospheric humidity, temperature and barometric pressure also influence test results.

Driving cycles

To obtain comparable exhaust-emission values, the predefined speeds on the dynamometer must agree with those on the road. Testing is based on a standardized driving cycle in which gearshifts, braking, idle phases, and standstill periods, have all been selected to provide a high level of correspondence with the velocities and acceleration that characterise typical driving in the normal traffic of a large town. Seven different test cycles are employed internationally. In Europe, the driving schedule has been abbreviated to EU Stage III (as of 01/2000; the 40 s pre-run is no longer used). The USA additionally uses the SFTP test for vehicles with air conditioning, as well as further sets of driving conditions.

Usually, a driver sits in the vehicle to maintain the speed at the levels indicated on a display screen.

Sampling and dilution procedures (CVS Method)

The European adoption of the constant-volume sampling method (CVS) in 1982 means that there is now basically a single standardized procedure in force worldwide for collecting exhaust gases.

Sampling and analysis of emissions

The dilution procedure operates according to the following principle:

The exhaust gases produced by the test vehicle are diluted with fresh air at a mean ratio of 1:10, and extracted using a special system of pumps such that the flow volume composed of exhaust gas and fresh air is maintained at a fixed ratio, i.e. the admixture of air is adjusted according to the vehicle's momentary ex-

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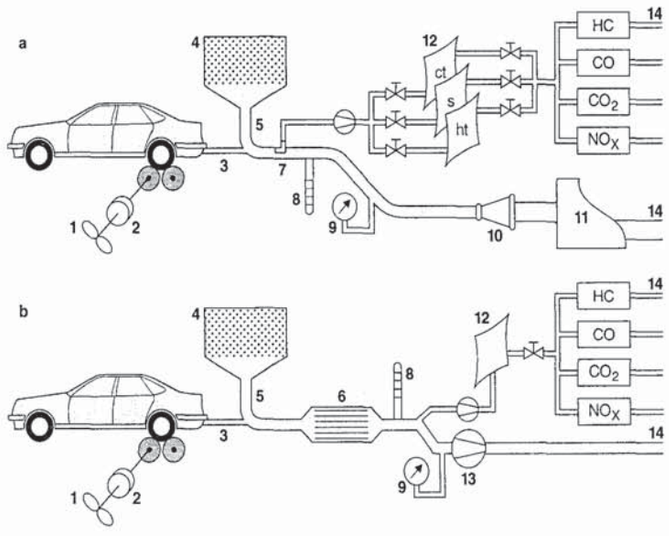
haust volume. Throughout the test a constant proportion of the diluted exhaust gas is extracted for collection in one or several sample bags. The pollutant concentration in the sample bags at the end of the test corresponds precisely to the mean concentration in the total quantity of fresh-air/exhaust mixture which has been extracted. Because the total volume of the fresh-air/exhaust mixture can be defined, the pollutant concentrations can be used as the basis for calculating the pollutant masses produced during the course of the test.

Advantages of this procedure: Condensation of the water vapor contained in the exhaust gas is avoided, which would otherwise cause a substantial reduction in the NO_X losses in the bag. In addition, dilution greatly inhibits the tendency of the exhaust components (especially hydrocarbons) to continue to react with one another. However, dilution does mean that the concentration of the pollutants decreases proportionally to the mean dilution ratio, necessitating the use of more sensitive analyzers. Standardized equipment is available for analysis of the pollutants in the bags.

Dilution systems

Either of two different but equally acceptable pump arrangements is generally used to maintain a constant flow volume for the test. In the first, a normal blower extracts the fresh-air/exhaust mixture through a venturi tube; the second arrangement makes use of a special rotary-piston blower (Roots blower). Either method is capable of measuring the flow volume with an acceptable degree of accuracy.

Test setups
a) For the US federal test (here with a Venturi system),
b) For the Europe test (here with rotary-piston blower).
1 Chassis dynamometer, 2 Inertial mass, 3 Exhaust gas, 4 Air filter, 5 Dilution air, 6 Radiator, 7 Sampling venturi tube, 8 Gas temperature, 9 Pressure, 10 Venturi tube, 11 Blower, 12 Sample bags, 13 Rotary-piston blower, 14 System outlet.
ct Emissions in transition phase, s Emissions in stabilized phase, ht Emissions during hot test



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Determining evaporative losses from the fuel system

(Evaporation tests)

Apart from the combustion pollutants produced in the engine, a motor vehicle emits additional quantities of hydrocarbons (HC) through evaporation of fuel from the fuel tank and fuel system (dependent on design and fuel temperature). Some countries (e.g. the USA and Europe) have regulations which limit these evaporative losses.

SHED Test

The SHED (Sealed Housing for Evaporative Determination) test is the most common procedure for determining evaporative emissions. It comprises two test phases with varying conditioning processes in a gas-tight chamber.

The first part of the test is carried out with the fuel tank approx. 40% full. The actual testing of the HC concentration in the chamber begins at 15.5 °C as the test fuel is warmed (initial temperature: 10...14.5 °C). The test concludes after one hour when the temperature of the fuel has risen by 14 °C; at this time the HC concentration is measured once again. Evaporative emissions are determined by comparing the initial and final measurements. The vehicle's windows and trunk lid must remain open during the test.

For determining the evaporative emissions in the second portion of the test, the vehicle is first warmed up by being run through the test cycle valid for the country concerned. The engine is then turned off with the vehicle in the chamber. The increase in the HC concentration is measured for a period of one hour as the vehicle cools.

The sum of both measurements must lie below the current limit value of 2 g of evaporated hydrocarbons. Meanwhile, a more stringent SHED test has been decided upon for implementation in the USA.

ECE/EC test cycle and limits

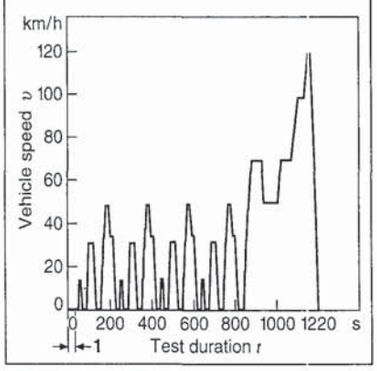
The ECE/EC test cycle incorporates a synthetically generated driving curve (see illustration) calculated to provide a reasonable approximation of driver behaviour in urban traffic. Since 1993, the test cycle has been supplemented by a suburban section with speeds of up to 120 km/h. This new ECE/EC test cycle is presently required by law in the following countries: Germany, Netherlands, Belgium, Luxembourg, France, Denmark, Great Britain, Ireland, Italy, Spain, Finland, Austria, Sweden, Greece and Portugal.

The exhaust emissions test is conducted as follows:

After the vehicle has been appropriately conditioned (vehicle allowed to stand with engine off at a room temperature of 20...30 °C for at least 6 hours) the actual test begins after a cold start and 40 second pre-running phase (this phase does not apply as of EU Stage III). During measurement, exhaust gas is collected in a sample bag using the CVS method. The pollutant masses determined by analysis of the contents of the bag are converted in the Europe test to the driven distance. In addition, hydrocarbons and oxides of nitrogen are combined to form a composite limit ($HC+NO_X$). These substances are considered separately as of EU Stage III.

More stringent limits have been in force since 1992 irrespective of the vehicle engine displacement. This directive designated 91/441 EEC (EU Stage I) is listed in the Section "Exhaust emission limits",

ECE/EC test cycle with suburban section
1 Pre-run (no measurement): to date 40 s; does not apply as of EU Stage III.
Cycle distance: 11 km
Average speed: 32.5 km/h
Maximum speed: 120.0 km/h



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Table 1. It also places limits on evaporative losses. A further reduction of the limits was imposed in 1996/97 by directive 94/12/EEC (EU Stage II).

Further reductions of limits are planned for Europe (Stage III and IV, 2000 and 2005):
- Cold start at –7 °C (as of 2002),
- EOBD (European On-Board Diagnosis) of parts relevant to exhaust emissions,
- More stringent evaporative-emissions test,
- Durability (80,000; 100,000 km) and in-field monitoring,
- Exhaust-gas sampling immediately after start

Test cycles in the USA

FTP 75 test cycle

The FTP 75 test cycle (Federal Test Procedure), which consists of three test sections, represents actual speeds measured in the USA on the streets of Los Angeles in morning commuter traffic (see illustration 3a):

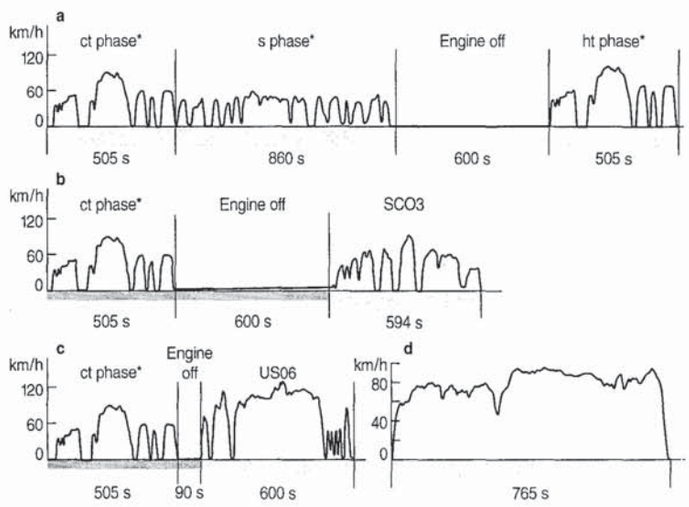
The vehicle to be tested is first conditioned (allowed to stand with engine off for 12 hours at a room temperature of 20...30 °C), then started and run through the prescribed test cycle:

Phase ct: Collection of diluted exhaust gas in bag 1 during the cold transition phase.

Phase s: Changeover to sample bag 2 at the beginning of the stabilized phase (after 505 s) without any interruption in the

USA test cycles

	a	b	c	d
Test cycle	**FTP 75**	**SC03**	**US06**	**Highway**
Cycle distance:	17.87 km	5.76 km	12.87 km	16.44 km
Cycle duration:	1877 s + 600 s pause	594 s	600 s	765 s
Average cycle speed:	34.1 km/h	34.9 km/h	77.3 km/h	77.4 km/h
Maximum cycle speed:	91.2 km/h	88.2 km/h	129.2 km/h	96.4 km/h



** ct Transition phase; s Stabilized phase; ht Hot test*
Phases during which exhaust gas is collected
Conditioning (other driving schedules are also possible)

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driving cycle. The engine is turned off for a 10-minute pause immediately following the stabilized phase (after 1372 s).
Phase ht: The engine is restarted for the hot test (505 s in duration). The speeds used in this phase correspond directly to those in the cold transition phase. Exhaust gases are collected in a third sample bag. The bag samples from the previous phases are analyzed during the pause before the hot test, as the samples should not remain in the bags for longer than 20 minutes.

The exhaust-gas sample in the third sample bag is analyzed once the driving sequence has been completed. The weighted sums of the pollutant masses (HC, CO and NO_x; ct 0.43, s 1, ht 0.57) from all three bags are correlated with the distance covered during the test, and then expressed as emissions per mile. The limits on pollutant emissions vary among individual countries. This test procedure is used in the USA including California ("Emission limits", Table 2) and in other countries (Table 4).

SFTP schedules
The tests in accordance with the SFTP standard will be phased-in in stages between 2001 and 2004. Together they consist of three driving schedules: the FTP 75, the SC03 and the US06 schedule.

They are intended to additionally examine the following driving patterns (see illustration, USA test cycles b and c):
- aggressive driving,
- major changes in driving speed,
- engine start and driving-off,
- frequent minor changes in driving speed,
- parking times and
- operation with the air conditioner running.

In the SC03 and US06 schedules, the ct phase of the FTP 75 test cycle is applied after preconditioning with no collection of exhaust gases, however other conditioning procedures are also possible.

The SC03 schedule is carried out at a temperature of 30 °C and 40 % relative humidity (vehicles with air conditioning only). The individual driving schedules are weighted as follows:

- Vehicles with air conditioner:
  35 % FTP 75 + 37 % SC03 + 28 % US06
- Vehicles without air conditioner:
  72 % FTP 75 + 28 % US06

The SFTP and FTP 75 test cycles must be successfully completed independently of each other (Tables 2 and 3 in the section "Exhaust emission limits").

Test cycles for determining fleet fuel consumption
Every vehicle manufacturer must determine the fuel consumption of his vehicle fleet. If a manufacturer exceeds certain limits, he is fined. He is awarded a bonus if consumption lies below certain limits. Fuel consumption is determined from the exhaust emissions produced during two test cycles: the FTP 75 test cycle (55 %) and the highway test cycle (45 %).

The highway test cycle is conducted once after preconditioning (vehicle allowed to stand with engine off for 12 hours at 20...30 °C) without measurements being conducted. The exhaust emissions from a second test run are then collected. Fuel consumption is determined based on the emissions (test cycle d).

Every new vehicle is required to comply with the limits for 50,000 miles, regardless of vehicle weight and engine displacement. Under certain conditions, the USA grants exemptions for the various model years. The legislation imposes different limits at 50,000 and 100,000 miles; permissible limits are higher at 100,000 miles (deterioration factors).

In addition to many measures designed to protect the environment, the Clean Air Act stipulated more stringent emission limits for vehicles since 1994 (Table 2). California decided to enforce the more stringent limits as early as 1993, and is planning to take further drastic steps.

Cold-start enrichment, which is necessary when a vehicle is started at low temperatures, produces particularly high emissions. These cannot be measured in current emissions testing, which is conducted at ambient temperatures of 20...30 °C. The Clean Air Act attempts to limit these emissions as well by prescribing an emissions test at −6.7 °C.

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However, a limit is prescribed for carbon monoxide only.

Japanese test cycle

Two test cycles with differing hypothetical driving curves are combined to provide the complete test.

Following a cold start, the 11-mode cycle is run four times, with evaluation of all four cycles. The 10·15-mode test as a hot test is run through once (see illustration).

Preconditioning for the hot start also includes the stipulated idle emissions test, and is conducted as follows:

After warming up the vehicle at 60 km/h for approx. 15 minutes, the HC, CO and CO_2 concentrations in the exhaust pipe are measured. Following a secondary warm-up period of 5 minutes at 60 km/h, the 10·15-mode hot test is then started. In the 11-mode test as well as in the 10·15-mode test, analysis is performed using a CVS system. The diluted exhaust gas is in each case collected in a bag. In the cold test, the pollutants are specified in terms of g/test, whereas in the hot test they are correlated with the distance driven, i.e. they are converted to g/km (see Table 5 in the section "Exhaust emission limits").

The exhaust gas regulations in Japan include limits on evaporative emissions which are measured using the SHED method.

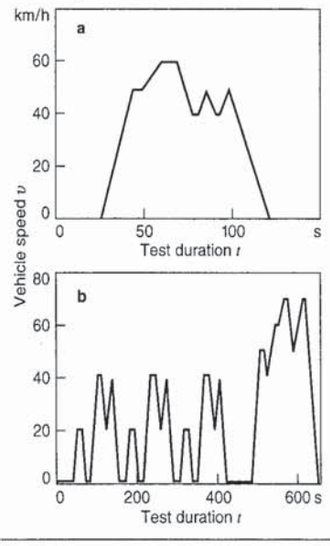
Japanese test cycles

a 11-mode cycle (cold test)

Cycle distance:	*1.021 km*
Cycles per test:	*4*
Average speed:	*30.6 km/h*
Maximum speed:	*60 km/h*

b 10·15-mode cycle (hot test)

Cycle distance:	*4.16 km*
Cycles per test:	*1*
Average speed:	*22.7 km/h*
Maximum speed:	*70 km/h*



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Exhaust-gas analyzers

Legislation requires that emission testing also be carried out on vehicles which are already on the road.

In addition, exhaust-gas analyzers are also indispensable service tools in the workshop, necessary both for optimal A/F-mixture adjustment and for effective trouble-shooting on the engine.

Test procedure

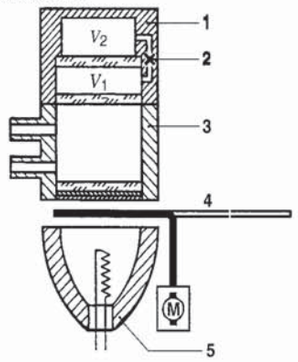
In automotive service operations, the infrared method has proved to be the only efficient and reliable means for testing exhaust gases.

This method bases on the fact that individual exhaust-gas components absorb infrared light at different specific rates, according to their characteristic wavelengths. The various designs include both single-component analyzers (e.g. for CO only) and multi-component analyzers (for CO/HC, CO/CO_2, CO/HC/CO_2 etc.).

Test chamber

Infrared radiation is transmitted by an emitter which has been heated to approx. 700 °C. The beam passes through a measuring cell before entering the receiver chamber. When measuring the CO content, the sealed receiver chamber contains gas with a defined CO content which absorbs a portion of the CO-specific radiation. This absorption causes the gas temperature to increase, which in turn causes gas with a volume of V_1 to flow via a flow sensor into the compensation volume with a volume of V_2. A rotating chopper disk induces a rhythmic interruption in the beam, producing an alternating basic flow between the two volumes V_1 and V_2. The flow sensor converts this motion into an alternating electrical signal. When a test gas with a variable CO content flows through the measuring cell, it absorbs radiant energy in a quantity proportional to its CO content; the energy is then no longer available in the receiver chamber. The result is a reduction of the base flow in the receiver chamber. The deviation from the alternating basic signal is therefore a measure of the CO content in the test gas.

Test chamber using infrared method
(schematic diagram)
1 Receiver chamber with compensation volume V_1 and V_2, 2 Flow sensor, 3 Measuring cell, 4 Rotating chopper disk with motor, 5 Infrared emitter.



Catalytic-converter testing

A representative component can be used to obtain an indirect measurement of catalytic-converter operation on vehicles with closed-loop-controlled catalytic devices. The best-suited is CO, which must not exceed 0.3 % by volume downstream of the converter, whereby lambda must be exactly 1.00 (±0.01). Lambda, in turn, can be determined using the composition of the exhaust gas at the catalytic converter's outlet. The exhaust-gas analyzer calculates the value for lambda using the CO, HC, CO_2 and O_2 exhaust gas components, with constants used for NO and fuel composition HC_V. The O_2 content is measured by an electrochemical probe.

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Current (1998) emissions limits for gasoline engines

Table 1
EU emissions limits as measured in ECE/EU test cycle

Standards	Introduction	CO g/km	HC g/km	NO_x g/km	HC+NO_x g/km
EU Stage I	07.92	2.72	–	–	0.97
EU Stage II	01.96	2.2	–	–	0.5
EU Stage III	01.00	2.3	0.2	0.15	–
EU Stage IV	01.05	1.0	0.1	0.08	–

Table 2
US Federal (49 states) and California emissions limits. FTP 75 test cycle

	Model year	Standards	CO g/mile	HC g/mile	NO_x g/mile
US Federal	1994	Level 1	3.4	0.25	0.4
	2004 [1]	Level 2	1.7	0.125 [2]	0.2
California	[3]	TLEV [4]	3.4	0.125 [2]	0.4
	[3]	LEV [5]	3.4	0.075 [2]	0.2
	[3]	ULEV [6]	1.7	0.04 [2]	0.2

[1] Proposal. [2] NMOG = Non Methanic Organic Gases. [3] Introduction varies according to manufacturer's NMOG fleet average (both vehicle and total fleet are certified).
[4] Transitional Low Emission Vehicles. [5] Low Emission Vehicles. [6] Ultra Low Emission Vehicles.

Table 3
US emissions limits. STFP test cycle

	NMHC[1]+NO_x g/mile	$CO_{Composite}$ [2] g/mile	CO_{SC03} [2] g/mile	CO_{US06} [2] g/mile
up to 50,000 miles	0.65	3.4	3.0	9.0
50,000 to 100,000 miles	0.91	4.2	3.7	11.1

[1] Non Methane HC. [2] The manufacturer has the option of selecting $CO_{Composite}$ or CO_{SC03} and CO_{US06} limits.

Table 4
Emissions limits for Argentina, Australia, Brazil, Canada, Mexico, Norway, Switzerland, and South Korea measured with FTP 75 test cycle

Country	Introduction	CO g/km	HC g/km	NO_x g/km	Evap. emissions (HC) g/test
Argentina	01.97	2.0	0.3	0.6	6.0
Australia	01.97	1.9	0.24	0.57	1.9
Brazil	01.97	2.0	0.3	0.6	6.0
Canada	01.98	2.1	THC [2] 0.25; NMHC [3] 0.16	0.24	2.0
Mexico	01.95	2.1	0.25	0.62	2.0
Norway	01.89	2.1	0.25	0.62	2.0
Switzerland [1]	10.87	2.1	0.25	0.62	2.0
South Korea	01.91 01.00	2.1	0.25 0.16	0.62 0.25	2.0

[1] EU/ECE regulations recognized since 10/95. [2] THC = Total HC. [3] NMHC = Non Methane HC.

Table 5
Japanese emissions limits measured in Japanese test cycle

Test procedure	CO	HC	NO_x	Evap. emissions
10-15-mode (g/km)	2.1...2.7 (0.67)	0.25...0.39 (0.08)	0.25...0.48 (0.08)	–
11-mode (g/test)	60.0...85.0 (19.0)	7.0...9.5 (2.2)	4.4...6.0 (1.4)	–
SHED (g/Test)	–	–	–	2.0

( ) planned figures

PETITIONER'S EXHIBIT 1002
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