Material	Code[7]	Range of application[8] °C	Shore A hardness (D)	Tensile strength[9] N/mm²	Ultimate elongation[9] %	Resistance to[11] Weathering	Ozone	Gasoline	Diesel fuel	Mineral oil
Thermoplastic elastomers										
Blend/olefin with non-linked to fully cross-linked rubber	TPE-O[14]	– 40 ... + 100 (120)	45A ... 50D	3 ... 15	250 ... 600	+	+	0	0	0
Blend/styrene block polymers	TPE-S[14]	– 60 ... + 60 (100)	30A ... 90A	3 ... 12	500 ... 900	+	+	–	–	–
Polyester elastomer	TPE-E[14]	– 50 ... + 150	40D ... 80D	9 ... 47	240 ... 800	0[10]	×	×	+	+
Polyester urethane	TPE-U[14]	– 40 ... + 100	70A ... 70D	15 ... 55	250 ... 600	0[10]	+	0	×	+
Polyetherblockamide	TPE-A[14]	– 40 ... + 80	75A ... 70D	30 ... 60	300 ... 500	0[10]	+	×	+	×

[7] DIN ISO 1629.
[8] Not continuous-service temperature.
[9] Depending upon composition of compound.
[10] Can be improved by adding protective agents.
[11] + good resistance, × limited resistance, 0 low resistance, – no resistance.
[14] No ISO standard to date.

PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

Plastics abbreviations with chemical names and trade names[3])

Code	Chemical name	Trade names
ABS	Acrylonitrile butadiene styrene	Cycolac, Novodur, Ronfalin, Terluran
ACM	Polyacrylate rubber	Cyanacryl, Hycar
EAM[1])	Ethylene acrylate rubber	Vamac
APE[1])	Ethylene propylene rubber	Arylef, APEC
ASA	Acrylate styrene acrylonitrile	Luran S
AU	Polyurethane elastomers	Urepan
CA	Cellulose acetate	Bergacell, Tenite
CAB	Cellulose acetate butyrate	Cellidor, Tenite
CM	Chlorinated polyethylene	Bayer CM, CPE
CR	Chloroprene rubber	Baypren, Neoprene
CSM	Chlorosulfonated polyethylene	Hypalon
ECO	Epichlorohydrin rubber	Herclor, Hydrin
EP	Epoxy	Araldite
EPDM	Ethylene propylene rubber	Buna AP, Dutral, Keltan, Nordel, Vistalon
EU	Polyurethane elastomers	Adiprene C
FPM	Fluorcarbon rubber	DAI-EL, Fluorel, Tecnoflon, Viton
HNBR[1])	Hydrogenated nitrile rubber	Therban, Zetpol
IR	Isoprene rubber	Cariflex IR, Natsyn
MF	Melamine-formaldehyde	Bakelite, Resinol, Supraplast, Resopal
MPF	Melamine/phenol-formaldehyde	Supraplast, Resiplast
MVQ	Silicone rubber	Rhodorsil, Silastic, Silopren
NBR	Nitrile butadiene rubber	Buna N, Chemigum, Hycar, Perbunan
PA 46[1])	Polyamide 46	Stanyl
PA 6-3-T	Amorphous polyamide	Trogamid T
PA 6	Polyamide 6 (polymers of ε-caprolactam)	Akulon, Durethan B, Grilon, Nivionplast, Perlon, Renyl, Sniamid, Technyl, Ultramid B, Wellamid
PA 66	Polyamide 66 (polymers of hexamethylene diamide and adipic acid)	Akulon, Durethan B, Grilon, Nivionplast, Perlon, Renyl, Sniamid, Technyl, Ultramid B, Wellamid
PA X[1])	X = partially aromatic polyamides	Ultramid T[4]), Amodel 1...[5]), Amodel 4...[6]), Grivory GV[7]), Grivory HTV[8]), Zytel HTN[9]), IXEF[10])
PA 11	Polyamide 11 (polymers of 11-aminoundecanoic acid)	Rilsan B
PA 12	Polyamide 12 (polymers of dodecalactam)	Grilamid, Rilsan A, Vestamid
PAI	Polyamide imide	Torlon
PAN	Polyacrylonitrile	Dralon, Orlon
PBTP	Polybutylene terephthalate	Crastin, Pocan, Ultradur, Vestodur, Celanex
PC	Polycarbonate	Makrolon, Orgalan, Sinvet, Lexan
PA 612	Polyamide 612 (polymers of hexamethylene diamine and dodecanoic acid)	Zytel
COC[1])	Cyclo-olefin copolymers	Topas
LCP	Liquid crystal polymers	Vectra, Zenite
PA 6/66	Copolyamide 6/66	Ultramid C, Technyl, Grilon TSV
SPS[1])	Syndiotactic polystyrene	Questra, Xarec
PK[1])	Polyketon	Carilon

PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

Code	Chemical name	Trade names
LFT[1])	Long-fiber reinforced thermoplastic	Celstran
(PC + ABS)	Blend of polycarbonate + ABS	Bayblend, Cycoloy
(PC + ASA)	Blend of polycarbonate + ASA	Terblend S
(PC-PBT)	Blend of polycarbonate + PBT	Xenoy
PE	Polyethylene	Hostalen, Lupolen, Stamylan, Vestolen
PEEK	Polyether ether ketone	Victrex "PEEK"
PEI	Polyether imide	Ultem
PES	Polyether sulphone	Victrex "PES", Ultrason E
PETFE[1])	Ethylene tetrafluorethylene copolymer	Hostaflon ET, Tefzel
PETP	Polyethylene terephthalate	Arnite, Crastin, Mylar, Rynite, Impet
PF	Phenol-formaldehyde	Bakelite, Supraplast, Vyncolite
PFA	Perfluoralkoxyethylene	Teflon PFA
PFEP[1])	Fluorinated ethylene-propylene copolymer	Teflon FEP
PI	Polyimide	Kapton, Kerimid, Kinel, Vespel
PMMA	Polymethyl methacrylate	Degalan, Diakon, Lucryl, Perspex, Plexiglas, Vedril
POM	Polyoxymethylene, polyformaldehyde (a polyacetal)	Delrin, Hostaform, Ultraform
PP	Polypropylene	Daplen, Hostalen PP, Moplen Stamylan P, Starpylen, Vestolen
(PPE + SB)	Blend of polyphenylene ether + SB	Noryl, Luranyl
(PPE + PA)	Blend of polyphenylene ether + PA	Noryl GTX, Ultranyl, Vestoblend
PPS	Polyphenylene sulfide	Fortron, Ryton, Tedur
PS	Polystyrene	Edistir, Hostyren, Lustrex
PSU	Polysulphone	Udel, Ultrason S
PTFE	Polytetrafluorethylene	Fluon, Hostaflon, Teflon
PUR	Polyurethane	Lycra, Vulkollan
PVC-P	Polyvinyl chloride, plasticized	Trosiplast, Vestolit, Vinoflex
PVC-U	Polyvinyl chloride, unplasticized	Trovidur, Hostalit, Vinidur, Vestolid
PVDF	Polyvinylidene fluoride	Dyflor, Kynar, Solef
PVF	Polyvinyl fluoride	Tedlar
SAN	Styrene acrylnitrile	Kostil, Luran, Tyril
SB	Styrene butadiene	Hostyren, Lustrex
SBR	Styrene butadiene rubber	Buna Hüls, Buna S, Cariflex S
TPE-A[1])	Polyether blockamide	Pebax, Vestamid E
TPE-E[1])	TPE[2]) polyester base	Arnitel, Hytrel, Riteflex
TPE-O[1])	TPE[2]) olefin base	Leraflex, Santoprene
TPE-S[1])	TPE[2]) styrene base	Cariflex, Evoprene, Kraton
TPE-U[1])	Polyester urethane	Desmopan, Elastollan
UF	Urea-formaldehyde	Bakelite, Pollopas
UP	Unsaturated polyester	Keripol, Leguval, Palatal

[1]) Material code not yet standardized.
[2]) TPE: Thermoplastic rubber.
[3]) ISO 1043/DIN 7728 (Thermoplastics, thermosetting plastics), ISO 1629 (Rubbers).
Material codes [4]) – [10]) are standardized
[4]) PA 6/6T
[5]) PA 6T/6I/66
[6]) PA 6T/66
[7]) PA 66 + PA 6I/6T
[8]) PA 6I/6T
[9]) PA 6T/MPMDT
[10]) PA MXD 6

PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

Automotive paints, structure of solid-color coatings

Layer	Layer thickness in µm	Structure	Composition					Application
			Binders	Solvents	Pigments	Extenders	Additives and SC	
1	20...25	KTL	Epoxy resins Polyurethane	Water, small amounts of water-miscible organic solvents	Anorganic (organic)	Anorganic extenders	Surface-active substances, anti-crater agents, 20% SC	ET
2a	approx. 35	Primer	Polyester, melamine, urea & epoxy resins	Aromatic compounds, alcohols	Anorganic and organic	Anorganic solids	e.g. wetting agents, surface-active substances 58...62% SC	PZ ESTA-HR
2b	approx. 35	Water extender	Water-soluble polyester, polyurethane & melamine resins	Water, small amounts of water-miscible organic solvents			43...50% SC	PZ ESTA-HR
2c	approx. 20	Thin-film water extender	Water-soluble polyurethane & melamine resins	Water, small amounts of water-miscible organic solvents		Anorganic extenders	e.g. wetting agents, surface-active substances 32...45% SC	PZ ESTA-HR
3a	40...50	Solid-color top coat	Alkyd & melamine resins	Esters, aromatic compounds, alcohols		–	e.g. leveling & wetting agents	PZ ESTA-HR
3b	10...35 (color-specific)	Water-borne solid-color base coat	Water-soluble polyester, polyurethane, polyacrylate & melamine resins	Small amounts of water-miscible co-solvents		–	Wetting agents 20...40% SC	PZ ESTA-HR
4a	40...50	Conventional clear coat	Acrylic & melamine resins	Aromatic compounds, alcohols, esters	–	–	e.g. leveling agents and light stabilizers 45% SC	PZ ESTA-HR
4b	40...50	2C-HS	HS acrylate resin polyisocyanates	Esters, aromatic compounds	–	–	e.g. leveling agents and light stabilizers 58% SC	
4c	40...50	Powder-slurry clear coat	Urethane-modified epoxy/carboxy system	–	–	–	e.g. light stabilizers 38% SC	

PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

Structure of metallic coatings

Layer	Layer thickness in µm	Structure	Composition					Application
			Binders	Solvents	Pigments	Extenders	Additives and SC	
1	20...25	KTL	Epoxy resin polyurethane	Water, small amounts of water-miscible organic solvents	Anorganic (organic)	Anorganic extenders	Surface-active substances, anti-crater agents, 20 % SC	ET
2a	approx. 35	Extender melamine, urea & epoxy resins	Polyester, alcohols	Aromatic compounds, alcohols	Anorganic and organic	Anorganic extenders	e.g. wetting agents, surface-active substances 58...62 % SC	PZ ESTA-HR
2b	approx. 35	Water extender	Water-soluble polyester, polyurethane & melamine resins	Water, small amounts of water-miscible organic solvents			43...50 % SC	PZ ESTA-HR
2c	approx. 20	Thin-film water extender		Water, small amounts of water-miscible organic solvents			e.g. wetting agents, surface-active substances 32...45 % SC	PZ ESTA-HR
3a	10...15	Metallic base coat		Esters, aromatic compounds	Aluminum particles, mica particles	–	15...30 % SC	PZ ESTA-HR
3b	10...15	Water-soluble metallic base coat	Water-soluble polyester, polyurethane, polyacrylate & melamine resins	Small amounts of water-miscible co-solvents	Aluminum and mica particles organic and anorganic pigments	–	Wetting agent	PZ ESTA-HR
4a	40...50	Conventional clear coat	Acrylic & melamine resins	Aromatic compounds, alcohols, esters	–	–	e.g. leveling agents and light stabilizers 45 % SC	PZ ESTA-HR
4b	40...50	2C-HS	HS acrylate resin, polyisocyanates	Esters, aromatic compounds	–	–	e.g. leveling agents and light stabilizers 58 % SC	
4c	40...50	Powder-slurry clear coat	Urethane-modified epoxy/carboxy system	–	–	–	e.g. light stabilizers 38 % SC	

Acronyms: DS High film build, ESTA-HR Electrostatic high rotation, ET Electrophoretic coating, SC (FK) Solids content or non-volatiles, KTL Cathodic deposition, PZ Pneumatic spray, 2C-HS 2-component high-solid (high levels of non-volatile matter).

PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

Lubricants

Terms and definitions

Lubricants provide mutual insulation for components in a state of relative motion. The lubricant's function is to prevent direct contact between these components, thereby reducing wear and minimizing, or optimizing, friction. Lubricants serve as coolants, sealants and corrosion inhibitors, and can also reduce operating noise. The lubricant can be solid, consistent, liquid or gaseous in form. Specific lubricants are selected with reference to design characteristics, materials combinations, the operating environment and the stress factors encountered at the friction surface.

Additives
Additives are substances mixed into the lubricant in order to improve specific properties. These substances modify either the lubricant's physical characteristics (e.g. viscosity index improvers, pour-point depressors) or its chemical properties (e.g. oxidation inhibitors, corrosion inhibitors). In addition, the properties of the friction surfaces themselves can be modified with additives which change the friction characteristics (friction modifiers), protect against wear (anti-wear agents), or provide protection against scoring and seizure (extreme-pressure additives). Great care must be exercised in order to ensure that the additives are correctly matched with each other and with the base lubricant.

AFC (Anti-Friction-Coating)
Solid lubricant combinations which a binding agent holds in place on the friction faces.

ATF (Automatic Transmission Fluid)
Special-purpose lubricants specifically formulated to meet stringent requirements for operation in automatic transmissions.

Ash (DIN 51575, 51803)
The mineral residue which remains after oxide and sulfate incineration.

Bingham bodies
Materials whose flow characteristics differ from those of Newtonian liquids.

Bleeding (Oil separation, DIN 51817)
Separation of the base oil and the thickener in a lubricating grease.

Cloud point (DIN ISO 3015)
The temperature at which mineral oil becomes opaque due to the formation of paraffin crystals or precipitation of other solids.

Consistency (DIN ISO 2137)
A measure of the ease with which lubricating greases and pastes can be deformed.

Doped lubricants
Lubricants containing additives for improving specific properties (e.g. aging stability, wear protection, corrosion protection, viscosity-temperature characteristics).

Dropping point (DIN ISO 2176)
Temperature at which a lubricating grease attains a specified viscosity under defined conditions.

EP Lubricants (Extreme Pressure)
See high-pressure lubricants.

Fire point/flash point (DIN ISO 2592)
The lowest temperature (referred to 1013 hPa) at which a gaseous mineral product initially flashes (fire point), or continues to burn for at least 5 secs (burning point).

Flow pressure (DIN 51805)
According to Kesternich, the gas pressure required to press a consistent lubricant through a standardized test nozzle. The flow pressure is an index of a lubricant's starting flow characteristics, particularly at low temperatures.

Friction modifiers
Polar lubricant additives which reduce friction in the mixed-friction range and increase bearing capacity after adsorption on the surface of the metal. They also inhibit stick-slip behavior.

PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

Gel-type greases
Lubricants with inorganic gelling agents (e.g. Bentonites, Aerosiles, silica gels).

Graphite
Solid lubricant with layer-lattice structure. Graphite provides excellent lubrication when combined with water (e.g. high atmospheric humidity) and in carbon-dioxide atmospheres or when combined with oils. It does not inhibit friction in a vacuum.

High-pressure lubricants
Contain additives to enhance load-bearing capacity, to reduce wear and to reduce scoring (generally provide good performance in steel-to-steel and steel-to-ceramic applications).

Induction period
The period which elapses before substantial changes occur in a lubricant (e.g. aging of an oil containing an oxidation inhibitor).

Inhibitors
Lubricant protection additives (e.g. oxidation and corrosion inhibitors).

Low-temperature sludge
Products of oil degradation which form in the engine crankcase due to incomplete combustion and condensation at low engine load. Low-temperature sludge increases wear and can cause engine damage. Modern high-quality engine oils inhibit its formation.

Metal soaps
Reaction products from metals or from their compounds with fatty acids. They are used as thickeners for grease and as friction modifiers.

Mineral oils
Mineral oils are distillates or raffinates produced from petroleum or coal. They consist of numerous hydrocarbons in various chemical combinations. Classification is according to the predominant component: paraffin-based oils (chain-shaped saturated hydrocarbons), naphthene-based oils (closed-chain saturated hydrocarbons, generally with 5 or 6 carbon atoms per ring) or aromatic oils (e.g. alkylbenzene). These substances are distinguished by major variations in their respective chemical and physical properties.

Molybdenum disulfide (MoS_2)
A solid lubricant with layer-lattice structure. Only low cohesive forces are present between the individual layers, so their mutual displacement is characterized by relatively low shear forces. A reduction in friction is only obtained when MoS_2 is applied in suitable form to the surface of the metal (e.g. in combination with a binder such as (MoS_2 anti-friction coating).

Multigrade oils
Engine and transmission oils with good resistance to viscosity-temperature change (high viscosity index VI). These oils are formulated for year-round use in motor vehicles; their viscosity ratings extend through several SAE grades.

Penetration (DIN ISO 2137)
Depth (in 10^{-1} mm) to which a standardized cone penetrates into a consistent lubricant within a defined period and at a specified temperature. The larger the number, the softer the lubricant.

Polar substances
Dipolar molecules are easily adsorbed onto metal surfaces. They enhance adhesion and bearing capacity, thus reducing friction and wear. This category includes, for example, esters, ethers, polyglycols and fatty acids.

Pour point (DIN ISO 3016)
The lowest temperature at which an oil continues to flow when cooled under defined conditions.

Rheology
Science dealing with the flow characteristics of materials. These are generally represented in the shape of flow curves.
Coordinate plotting:
Shear stress $\tau = F/A$ ($N/m^2 = Pa$)
F force, A surface area
against
Shear rate $D = v/y$ (s^{-1})
(linear shear rate)
v velocity, y thickness of lubricating film.

PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

Dynamic viscosity
$\eta = \tau/D$ (Pa · s)
The formerly-employed unit "centiPoise" (cP) is equal to the unit (mPa · s).

Kinematic viscosity
$\nu = \eta/\varrho$ (mm²/s)
ϱ density (kg/m³).
The formerly-employed unit "centiStokes" (cSt) is equal to the unit (mm²/s).

Newtonian fluids
These display a linear relationship between τ and D in the shape of a straight line through zero, with the slope increasing as a function of viscosity.

All materials not characterized by this kind of flow behavior are classified as non-Newtonian fluids.

Intrinsically viscous flow behavior
Decrease in viscosity with increasing shear rate (e.g. liquid grease, multigrade oil with VI improvers).

Dilatant flow behavior
Increase in viscosity with increasing shear rate.

Plastic flow behavior
Formability of an intrinsically viscous fluid supplemented by yield value (e.g. lubricating greases).

Thixotropy
A characteristic of those non-Newtonian fluids that display an increase in viscosity proportional to shear time, and only gradually recover their original viscosity once shearing ceases.

Rheopexy
A characteristic of those non-Newtonian fluids that display a reduction in viscosity proportional to shear time, and only gradually recover their original viscosity once shearing ceases.

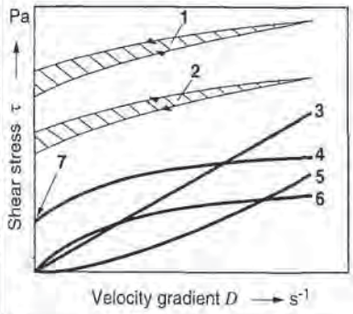
Stribeck curve
Portrays friction levels between two liquid- or grease-lubricated bodies separated by a narrowing gap (e.g. lubricated plain or roller bearings) as a function of sliding speed.

Solid-body friction
The height of the lubricant layer is lower than that of the roughness protrusions in the material's surface.

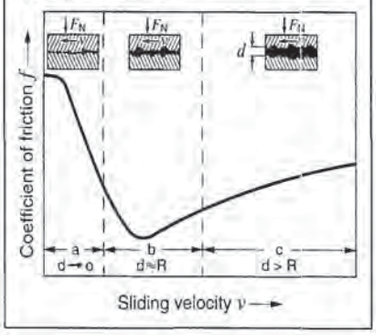
Mixed friction
The height of the lubricant layer is approximately equal to that of the roughness protrusons

Hydrodynamics
Complete separation between primary and opposed body (virtually frictionless condition).

Flow curves
1 Rheopex, 2 Thixotropic, 3 Newtonian, 4 Plastic, 5 Dilatant, 6 Intrinsically viscous, 7 Yield limit.



Stribeck curve
R Surface roughness, F_N Normal force, d Distance between basic and counter-body. Range a: solid friction, high wear; Range b: mixed friction, moderate wear; Range c: hydrodynamics, no wear.



PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

Viscosity (DIN 1342, DIN 51550)
Defines the internal friction of substances. It indicates the degree of resistance (internal friction) with which the substance's molecules oppose displacement forces (see Rheology).

Viscosity index (VI) (DIN ISO 2909)
The viscosity index VI is a mathematically-derived number expressing the change in a mineral-oil product's viscosity relative to its temperature. The greater the VI, the lower the effect of temperature on the viscosity.

Viscosity classification
Classification of oils in specific viscosity ranges. ISO viscosity classifications (DIN 51519, see Table 1).
SAE viscosity grades (DIN 51511, SAE J300, DIN 51512, SAE J306c, see Tables 2 and 3).

Yield point (DIN 13342)
The minimum shear stress at which a substance begins to flow. Above the yield point, the rheological characteristics of plastic substances are the same as those of liquids.

Table 1. Viscosity classes for industrial lubricating oils to ISO 3448 (DIN 51519)

ISO viscosity class	Medium viscosity at 40 °C mm²/s	Kinematic viscosity limits at 40 °C mm²/s	
		min.	max.
ISO VG 2	2.2	1.98	2.42
ISO VG 3	3.2	2.88	3.52
ISO VG 5	4.6	4.14	5.06
ISO VG 7	6.8	6.12	7.48
ISO VG 10	10	9.00	11.0
ISO VG 15	15	13.5	16.5
ISO VG 22	22	19.8	24.2
ISO VG 32	32	28.8	35.2
ISO VG 46	46	41.4	50.6
ISO VG 68	68	61.2	74.8
ISO VG 100	100	90.0	110
ISO VG 150	150	135	165
ISO VG 220	220	198	242
ISO VG 320	320	288	352
ISO VG 460	460	414	506
ISO VG 680	680	612	748
ISO VG 1000	1000	900	1100
ISO VG 1500	1500	1350	1650

Table 2. SAE viscosity grades for engine oils/transmission lubricants (SAE J300, Dec. 95)

SAE viscosity grade	Viscosity (ASTM D 5293) mPa·s at °C max.	Limit pumping viscosity (ASTM D 4684) with no yield point mPa·s at °C max.	Kinematic viscosity (ASTM D 445) mm²/s at 100 °C min.	max.	Viscosity under high shear (ASTM D 4683, CEC L-36-A-90, ASTM D 4741) mPa·s at 150 °C and 10^6 s^{-1} min.
0 W	3250 at –30	60000 at –40	3.8	–	–
5 W	3500 at –25	60000 at –35	3.8	–	–
10 W	3500 at –20	60000 at –30	4.1	–	–
15 W	3500 at –15	60000 at –25	5.6	–	–
20 W	4500 at –10	60000 at –20	5.6	–	–
25 W	6000 at –5	60000 at –15	9.3	–	–
20	–	–	5.6	<9.3	2.6
30	–	–	9.3	<12.5	2.9
40	–	–	12.5	<16.3	2.9 (0W–40, 5W–40, 10W–40)
40	–	–	12.5	<16.3	3.7 (15W–40, 20W–40, 25W–40, 40)
50	–	–	16.3	<21.9	3.7
60	–	–	21.9	<26.1	3.7

PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

Table 3. SAE viscosity grades for transmission lubricants (SAE J306c, March 85)

SAE viscosity grade	Maximum temperature °C for dynamic viscosity at 150,000 mPa · s (ASTM D2983)	Kinematic viscosity mm²/s at 100 °C (ASTM D 445)	
		min.	max.
70 W	–55	4.1	–
75 W	–40	4.1	–
80 W	–26	7.0	–
85 W	–12	11.0	–
90	–	13.5	24.0
140	–	24.0	41.0
250	–	41.0	–

Worked penetration (DIN ISO 2137)
Penetration of a grease sample after it is warmed to 25 °C and processed in a grease kneader.

Engine oils

Engine oils are employed primarily to lubricate contiguous components in relative motion within the internal-combustion engine. The oil also removes heat generated by friction, carries particles removed by friction away from the source, washes out contaminants and holds them in suspension, and protects metals against corrosion. The most common engine oils are mineral oils treated with additives (HD oils: Heavy Duty for extreme operating conditions). Higher stress-resistance requirements combined with extended oil-change intervals have led to widespread application of fully and semi-synthetic oils (e.g. hydro-crack oils). The quality of an engine oil is determined by its origin, the refining processes used on the mineral oil (except in the case of synthetic oils) and the additive composition.

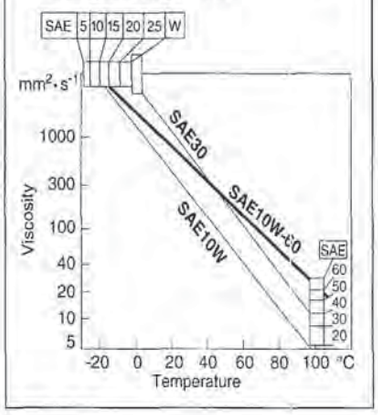
Additives are classified according to their respective functions:
- Viscosity-index (VI) improvers,
- Pour-point improvers,
- Oxidation and corrosion inhibitors,
- Detergent and dispersant additives,
- Extreme-pressure (EP) additives,
- Friction modifiers,
- Anti-foaming agents.

Oil is subjected to considerable thermal and mechanical stresses in the IC engine. The data on the oils' physical properties provide information on their operating range, but are not indicative of their other performance characteristics.

There thus exist several different procedures for evaluating engine oils (see comparison of engine-oil performance categories):
- ACEA (Association des Constructeurs Européens de l'Automobile) standards, replaced the CCMC standards (Comité des Constructeurs d'Automobiles du Marché Commun) at the beginning of 1996,
- API Classification (American Petroleum Institute)
- MIL Specifications (Military),
- Manufacturer's specifications.

The approval criteria include the following:
- Sulfate-ash content,
- Zinc content,
- Engine type (diesel or spark-ignition engines, naturally-aspirated or forced-induction engines),
- Load on power-transmissior components and bearings,
- Wear-protection properties,
- Oil operating temperature (in oil pan),

Viscosity/Temperature curves for single and multigrade engine oils



PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

- Combustion residue and chemical stress exerted upon the oil by acidic combustion products,
- The oil's detergent and residue-scavenging properties,
- Its suitability for use with gasket and sealing materials.

ACEA (CCMC) Specifications

Engine oils for gasoline engines

A1-98: Special high-lubricity oils with reduced viscosity at high temperatures and high shear.

A2-96: Conventional and high-lubricity engine oils without any restriction on viscosity class. Higher requirements than CCMC G4 and API SH.

A3-98: Oils of this category meet higher requirements than A2-96 and CCMC G4 and G5.

Engine oils for passenger-car diesel engines

B1-98: Corresponding to A1-98 for low friction losses and, consequently, reduced fuel consumption.

B2-98: Conventional and high-lubricity engine oils compliant with the current minimum requirements (higher than those of CCMC PD2).

B3-98: Exceeds B2-98.

B4-98: Corresponds to B2-98, particularly suitable for VW TDI engines.

Engine oils for commercial-vehicle diesel engines

E1-96: Oils for naturally aspirated and turbocharged engines with normal intervals between oil changes.

E2-96: Derived from MB Specification Sheet 228.1. Primarily for engine designs predating the Euro-II standard.

E3-96: For Euro-II engines, derived from MB Specification Sheet 228.3. In comparison with the predecessor category CCMC D5, these oils evince a significant improvement in soot dispersion capability and a much reduced tendency to thicken.

E4-98: Currently the highest quality category for diesel engines compliant with the Euro-I and Euro-II standards and with high requirements, particularly for extended intervals between oil changes (according to manufacturer's specification). Based to a large extent on MB 228.5.

API Classifications

S Classes (Service) for gasoline engines.

C Classes (Commercial) for diesel engines.

SF: For engines produced in the 80s.

SG: For engines built since 1989, with more stringent sludge test, improved oxidation stability and friction protection.

SH: Since mid-1993, corresponds to API SG quality level, but with more stringent process requirements for oil quality testing.

SJ: Since October 1996, more tests than API SH.

CC: Engine oils for non-turbocharged diesel engines with low stress factors.

CD: Engine oils for non-turbocharged and turbocharged diesels, replaced by API CF in 1994.

CD-2: API CD requirements, plus additional requirements relevant to 2-stroke diesel engines.

CF-2: Oils with special 2-stroke properties (since 1994).

CE: Oils with CD performance characteristics, with supplementary test operation in American Mack and Cummins engines.

CF: Replaced API CD in 1994. Specially for indirect injection, even if the sulfur content of the fuel is >0.5%.

CF-4: As API CE, but with more stringent test procedure in single-cylinder Caterpillar turbo-diesel engines.

CG-4: For diesel engines operating under very stringent conditions. Exceeds API CD and CE. Fuel sulfur <0.5%. Required for engines compliant with post-1994 emission-control regulations.

SAE viscosity grades

(DIN 51511, SAE J300, DIN 51512, SAE J306c)

The SAE (Society of Automotive Engineers) classifications are the internationally-accepted standard for defining viscosity. The standard provides no information on the quality of the oil. A distinction is drawn between single-grade and multigrade oils. Multigrade oils are the type in widespread use today.

Two series are employed for the designation (see Table 2) with the letter "W" (Winter) being used to define specific cold-flow properties. The viscosity grades including the letter "W" are rated according to max-

PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

imum low-temperature viscosity, maximum viscosity pumping temperature and the minimum viscosity at 100 °C; viscosity grades without the "W" are rated only according to viscosity at 100 °C.

Multigrade oils
Multigrade oils are characterized by a less pronounced proportional relationship between temperature and viscosity. They reduce friction and wear, can be used all year round, and provide rapid lubrication for all engine components in cold starts.

High-lubricity oils
Lubricating oils with multigrade properties, low viscosity at low temperatures, and special anti-friction additives. Extremely low engine friction under all operating conditions reduces fuel consumption.

Transmission lubricants

Specifications for transmission lubricants are defined by the type of transmission and the stresses to which it is subjected through the entire range of operating conditions. The requirements (high pressure resistance, high viscosity stability relative to temperature, high resistance to aging, good anti-foaming properties, compatibility with gaskets and seals) can only be satisfied by lubricants treated with special additives. The use of unsuitable and qualitatively inferior oils typically results in damage to bearings and gear-tooth flanks.

The viscosity must also suit the specific application. Viscosity grades for vehicular transmissions are defined in DIN 51512 and SAE J 306 (see Table 3).

Synthesized oils are being increasingly used to meet special requirements (e.g. poly-α-olefines). Advantages over standard mineral lubricants include superior temperature-viscosity properties and increased aging resistance.

API Classifications for transmission lubricants
GL1-GL3: Outdated, no longer of practical significance.
GL4: Transmission lubricants for moderately-stressed hypoid-gear transmissions and for transmissions which operate at extreme speeds and impact loads, high rotational speeds and low torques, or low rotational speeds and high torques.
GL5: Transmission lubricants for highly-stressed hypoid-gear transmissions in passenger cars and in other vehicles where they are exposed to impact loads at high rotational speeds, and at high rotational speeds and low torques, or low rotational speeds and high torques.
MT-1: Non-synchromeshed manual transmissions in American trucks.

Lubricants for automatic transmissions
(ATF: Automatic Transmission Fluid)
Automatic transmissions differ from their manually shifted counterparts in the way that they transfer torque; positive mechanical and hydrodynamic force transfer being supplemented by friction coupling arrangements. Thus the friction response of automatic-transmission fluids is extremely important. Applications are basically classified according to the friction characteristics:

General Motors: Superseded classifications: Type A, Suffix A, DEXRON®, DEXRON®B, DEXRON®II C, DEXRON®II D. DEXRON® II E (valid through 1994).
DEXRON® III. Valid since 1 January, 1994. Features more stringent requirements for oxidation stability and consistency in frictional coefficient.
Ford: MERCON® (valid since 1987).
Other manufacturers: In accordance with operating instructions.

Lubricating greases

Lubricating greases are thickened lubricating oils. A great advantage that greases enjoy over oil is that they do not drain from the contact location. Complicated measures designed to seal them in place are therefore unnecessary (e.g. application in wheel bearings and moving systems such as ABS, alternators, ignition distributors, windshield-wiper motors, servo motors). Table 4 provides a general overview of the components in a consistent lubricating grease as blended from three basic components – base oil, thickener and additive.

PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

Mineral oils are usually employed as the basic oil component, although full-synthetic oils have recently become more common as a replacement (e.g. due to more stringent requirements for aging resistance, cold-flow properties, viscosity-temperature characteristics).
Thickeners are used as a binder for the base oil; metal soaps being generally employed. They bind the oil in a sponge-like soap structure (micelle) via occlusion and Van der Waals forces. The higher the proportion of thickener in the grease (depends upon the type of thickener), the greater the grease's consistency (depth of penetration of a test cone into the grease sample) and the higher the NLGI classification (see Table 5).

The additives serve to modify physical and chemical properties of the grease to achieve specific objectives (such as improvement of anti-oxidation properties, increased load capacity (EP additives), and reduction of friction and wear).

Solid lubricants (e.g. MoS_2) are also added to lubricating greases (for instance for lubricating constant-velocity joints in motor vehicles).

Specific lubricating greases are selected with reference to their physical characteristics and their effects upon the slicing surface, and to minimize interaction between the grease and the contact materials.

Example:
Mutually antagonistic effects with polymers:

- Formation of stress cracks,
- Consistency changes,
- Polymer degradation,
- Swelling, shrinkage, brittleness.

Thus, for example, mineral-oil greases and greases based on synthetic hydrocarbons should not come into contact with elastomers used together with brake fluid (polyglycol base), as this could provoke substantial swelling.

In addition, greases with varying compositions should not be mixed (changes in physical properties, grease liquefaction due to drop-point reduction).

The correct lubricating grease can substantially enhance the performance potential of products with sliding contact-surfaces (e.g. transmissions, friction and roller bearings, actuator and control systems).

Table 5. Consistency classifications for lubricating greases (DIN 51 818)

NLGI grade	Worked penetration as per DIN ISO 2137 in units (0.1 mm)
000	445 ... 475
00	400 ... 430
0	355 ... 385
1	310 ... 340
2	265 ... 295
3	220 ... 250
4	175 ... 205
5	130 ... 160
6	85 ... 150

Table 4. Composition of lubricating greases

Base oils	Thickeners	Additives
Mineral oils	Metal soaps	Oxidation inhibitors
– Paraffinic	(Li,Na,Ca,Ba,Al)	Fe, Cu ions, sequestering agents
– Naphthenic	Normal	Corrosion inhibitors
– Aromatic	Hydroxi-	Extreme-pressure additives
	Complex	(EP additives)
Poly-α-olefines		Wear-protection additives
Alkyl aromatics	Polyureas	(anti-wear additives)
Esterols		Friction reducers
Polyalcohols	PE	(friction modifiers)
Silicones	Bentonites	Adhesion improvers
Phenyletherols	Silica gels	Detergents, dispersants
Perfluorpolyethers		VI improvers
		Solid lubricants

Whatever the friction pairing, the multiplicity of lubricant components can be used to develop a high-performance lubricant.

PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

Fuels

Characteristics

Net and gross calorific values
The specific values for the net (formerly: low) and gross (formerly: high, or combustion) calorific values, or H_u and H_o respectively, provide an index for the energy content of fuels. Only the net calorific value H_u (combustion vapor) is significant in dealing with fuels with which water is produced as a byproduct of combustion.

Oxygenates, fuel constituents which contain oxygen, such as alcohols, ether, and fatty-acid methyl ester have a lower calorific value than pure hydrocarbons, because their bound oxygen does not contribute to the combustion process. Power comparable to that achievable with conventional fuels can thus only be attained at the cost of higher consumption rates.

Calorific value of air-fuel mixture
The calorific value of the combustible air-fuel mixture determines the engine's output. Assuming a constant stoichiometric ratio, this figure remains roughly the same for all liquid gases and fuels (approx. 3500 ... 3700 kJ/m³).

Fuels for spark-ignition engines (gasoline/petrol)

The minimum requirements for these types of fuels are contained in various national and international standards. European Standard EN228 defines the unleaded fuels which have been introduced within Europe ("Euro-Super"). DIN EN228 contains the corresponding German specifications along with supplementary descriptions of unleaded regular and super premium ("Super-Plus") fuels. In the USA, fuels for spark-ignition engines are defined in ASTM D439 (American Society for Testing and Materials). Fuels for spark-ignition engines are hydrocarbon compounds which may contain supplements in the form of oxygenous organic components or other additives to improve performance.

A distinction is made between regular and premium (super-grade) gasolines. Premium gasoline has higher anti-knock qualities and is formulated for use in high-compression engines. Volatility ratings vary according to region and whether the fuel is intended for summer or winter use.

Unleaded gasoline (EN 228)
Unleaded fuel is indispensable for vehicles equipped with catalytic converters for treating exhaust gases, as lead would damage the noble metals (e.g. platinum) in the catalytic converter and render it inoperative. It would also destroy the Lambda sensors employed to monitor exhaust gas composition for closed-loop control.

Unleaded fuels are a special mixture of high-grade, high-octane components (e.g. platformates, alkylates and isomerisates). The anti-knock qualities can be effectively increased with non-metallic additives such as methyl tertiary butyl ether (MTBE) in concentrations of 3 ... 15% and/or alcohol mixtures (methanol 2 ... 3%, and higher alcohols). As of the year 2000, the maximum lead content is limited to 5 mg/l.

Leaded gasoline
In Europe, sales of leaded gasoline are prohibited as of the year 2000, although for a transitional period it will still be available in exceptional cases.

Leaded fuel is still available in some countries, although their number is steadily decreasing.

Density
European Standard EN228 limits the fuel density range to 720 ... 775 kg/m³. Because premium fuels generally include a higher proportion of aromatic compounds, they are denser than regular gasoline, and also have a slightly higher calorific value.

Anti-knock quality (octane rating)
The octane rating defines the gasoline's anti-knock quality (resistance to pre-ignition). The higher the octane rating, the greater the resistance to engine knock. Two differing procedures are in international use for determining the octane rat-

PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

Table 1. Essential data for unleaded gasoline, European Standard EN 228 (valid since 1 January 2000)

Requirements	Unit	Specification
Anti-knock quality		
Premium, min.	RON/MON	95/85
Regular, min.[1])	RON/MON	91/82.5
Super Plus[1])	RON/MON	98/88
Density	kg/m³	720 ... 775
Sulfur, max.	mg/kg	150
Benzene, max.	% volume	1
Lead, max.	mg/l	5
Volatility		
Vapor pressure, summer, min./max.	kPa	45/60
Vapor pressure, winter, min./max.	kPa	60/90[1])
Evaporated vol. at 70 °C, summer, min./max.	% volume	20/48
Evaporated vol. at 70 °C, winter, min./max.	% volume	22/50
Evaporated vol. at 100 °C, min./max.	% volume	46/71
Evaporated vol. at 150 °C, min./max.	% volume	75/–
Final boiling point, max.	°C	210
VLI transitional period[3]), max.[2])		1150[1])

[1]) National values for Germany, [2]) VLI = Vapor Lock Index, [3]) spring and fall.

ing; these are the Research Method and the Motor Method.

RON, MON

The number determined in testing using the Research Method is the Research Octane Number, or RON. It serves as the essential index of acceleration knock.

The Motor Octane Number, or MON, is derived from testing according to the Motor Method. The MON basically provides an indication of the tendency to knock at high speeds.

The Motor Method differs from the Research Method by using preheated mixtures, higher engine speeds and variable ignition timing, thereby placing more stringent thermal demands on the fuel under examination. MON figures are lower than those for RON.

Octane numbers up to 100 indicate the volumetric content in percent of C_8H_{18} iso-octane (trimethyl pentane) contained in a mixture with C_7H_{16} n-heptane at the point where the mixture's knock-resistance in a test engine is identical to that of the fuel being tested. Iso-octane, which is extremely knock-resistant, is assigned the octane number 100 (100 RON and MON), while n-heptane, with low-resistance to pre-ignition, is assigned the number 0.

Increasing the anti-knock quality

Normal (untreated) straight-run gasoline displays little resistance to knock. Various refinery components must be added to obtain a fuel with an octane rating which is adequate for modern engines. The highest-possible octane level must also be maintained throughout the entire boiling range. Cyclic hydrocarbons (aromatics) and branched chains (iso-paraffins) provide greater knock resistance than straight-chain molecules (n-paraffins).

Additives based on oxygenous components (methanol, ethanol, methyl tertiary butyl ether (MTBE)) have a positive effect on the octane number, but can lead to difficulties in other areas (alcohols raise the volatility level and can damage fuel-system components).

Volatility

Gasolines must satisfy stringent volatility requirements to ensure satisfactory operation. The fuel must contain a large enough proportion of highly volatile components to ensure good cold starting, but the volatility must not be so high as to impair operation and starting when the engine is hot (vapor lock). In addition, environmental considerations demand that evaporative losses be held low. Volatility

PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

is defined in various ways.

EN228 defines 10 different volatility classes distinguished by various levels of vapor pressure, boiling-point curve and VLI (Vapor Lock Index). To meet special requirements stemming from variations in climatic conditions, countries can incorporate specific individual classes into their own national standards.

Boiling curve

Individual ranges on the boiling curve exercise a particularly pronounced effect on operating behavior. They can be defined according to volumetric fuel evaporation at three temperatures. The volume which evaporates up to 70 °C must be adequate to ensure good cold starting, (particularly important for old carburetor-type engines), but not so large as to promote formation of vapor bubbles when the engine is hot. The percentage of vaporized fuel at 100 °C determines the engine's warm-up qualities, as well as its acceleration and response characteristics once it has reached normal operating temperature. The vaporized volume up to 150 °C should be high enough to minimize dilution of the engine's lubricating oil, especially with the engine cold.

Vapor pressure

Fuel vapor pressure as measured at 38 °C/100 °F in accordance with prEN 13016-1 is primarily an index of the safety with which the fuel can be pumped into and out of the vehicle's tank. The Reid method is an alternative method of measuring fuel vapor pressure. All specifications place limits on this vapor pressure. Germany, for example, prescribes maxima of 60 kPa in summer and 90 kPa in winter. In the case of modern fuel-injected (EFI) engines though, when troubleshooting vapor-lock malfunctions, it is more important to know the vapor pressure at higher temperatures (80, 100 °C). Research procedures are in place, and a definitive standard is currently being prepared. Fuels to which methanol is added are characterized by a pronounced rise in vapor pressure at high temperatures.

Vapor/liquid ratio

This specification provides an index of a fuel's tendency to form vapor bubbles. It is based on the volume of vapor generated by a specific quantity of fuel at a set temperature. A drop in pressure (e.g. when driving over a mountain pass) accompanied by an increase in temperature will raise the vapor/liquid ratio and with it the probability of operating problems. ASTM D439, for example, defines specific vapor/liquid ratios for gasolines.

Vapor Lock Index (VLI)

This parameter is the sum of the Reid vapor pressure (in kPa x 10), and that proportion of the fuel that vaporizes up to a temperature of 70 °C. Both components in the equation are absolute data, the latter being derived from the boiling-point curve prior to multiplication by a factor of 7. The VLI provides more useful information on the fuel's influence on starting and operating a hot engine than that supplied by conventional data.

The correlation between VLI and vapor/liquid ratio is good in the case of fuels that contain no alcohol additives.

Sulfur

Direct-injection systems for gasoline engines require fuels with a low sulfur content on account of their exhaust-emission control systems. The byproducts of sulfur combustion would render the catalytic converters inoperative. Consequently, and also on account of the target reduction in SO_2 emissions, the sulfur content of fuels will have to be further reduced in future; figures in the range < 10 ppm are desirable.

Additives

Along with the structure of the hydrocarbons (refinery components), it is the additives which determine the ultimate quality of a fuel. The packages generally used combine individual components with various attributes.

Extreme care and precision are required both when testing additives and in determining their optimal concentrations. Undesirable side-effects must be avoided. To obtain optimal results, the quantities used remain the province of the manufacturer, who can add the correct ratios in the desired concentrations when

VW EX1012 (Part 2 of 4)
U.S. Patent No. 6,581,565

PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

filling the tanker at the refinery. Vehicle operators should refrain from adding supplementary additives of their own.

Anti-aging additives
These agents are added to fuels to improve their stability during storage, and are particularly important when the fuel also contains cracked components. They inhibit oxidation due to atmospheric oxygen and prevent catalytic reactions with metal ions (metal deactivators).

Intake-system contamination inhibitors
The entire intake system (injectors, intake valves) must be kept free of contamination and deposits for several reasons. A clean intake tract is essential for maintaining the factory-defined A/F ratios, as well as for trouble-free operation and minimal exhaust emissions. To achieve this end, effective detergent agents should be added to the fuel.

Corrosion protection
Moisture entrained with the fuel can lead to corrosion in the fuel system. An extremely effective remedy is afforded by anti-corrosion additives designed to form a protective layer below the film of water.

Environmentally compatible gasolines

Environmental authorities and legislative bodies are imposing increasingly stringent regulations for fuels to ensure low evaporation and pollutant emissions (ecologically sound fuels, reformulated gasoline). As defined in the regulations, these fuels' salient characteristics include reduced vapor pressure along with lower levels of aromatic components, benzene and sulfur, and special specifications for the final boiling point. In the USA, additives designed to prevent deposit formation in the intake tract are also mandatory.

Diesel fuels

Diesel fuels contain a multiplicity of individual hydrocarbons with boiling points ranging from roughly 180 °C to 370 °C. They are the product of graduated distillation of crude oil. The refineries are adding increasing amounts of conversion products to the diesel fuel; these "cracked components" are derived from heavy oil by breaking up (cracking) large molecules. Specifications for diesel fuels are defined in national standards, while European Standard EN 590 applies throughout Europe. The most important specifications in this standard are listed in Table 2.

Ignition quality, cetane number, cetane index

Because the diesel engine dispenses with an externally-supplied ignition spark, the fuel must ignite spontaneously (auto-ignition) and with minimal delay (ignition lag) when injected into the hot, compressed air in the combustion chamber. Ignition quality is an expression of the fuel's suitability for spontaneous auto-ignition in a diesel engine. The higher the cetane number, the greater the fuel's tendency to ignite. The cetane number 100 is assigned to n-hexadecane (cetane), which ignites very easily, while slow-burning methyl naphthalene is allocated the cetane number 0. The cetane number is determined using a test engine. A cetane number in excess of 50 is desirable for optimal operation in modern engines (smooth operation, emissions). High-quality diesel fuels contain a high proportion of paraffins with elevated CN ratings. Conversely, the aromatic compounds found in cracked components have a detrimental effect on ignitability.

Yet another indication of ignitabilty is provided by the cetane index, which can be calculated on the basis of density and various points on the boiling curve. In contrast to the cetane number, this index does not reflect the positive influence of "ignition enhancers" on the fuel's ignitability.

Cold-flow properties, filtration

Precipitation of paraffin crystals at low temperatures can result in fuel-filter blockage, ultimately leading to interruption of the fuel flow. Under unfavorable conditions paraffin particles can start to form at temperatures of 0 °C or even higher. Spe-

PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

cial selection and manufacturing procedures are thus necessary for winter diesel fuels in order to ensure trouble-free operation during cold seasons of the year. Normally, flow improvers are added to the fuel at the refinery. Although these do not actually prevent paraffin precipitation, they severely limit the growth of the crystals, which remain small enough to pass through the pores in the filter material. Other additives can be used to maintain the crystals in a state of suspension, extending the filtration limit downward even further.

Suitability for cold-weather operation is defined with reference to a standardized procedure for determining the "limit of filtration". This is known as the CFPP, or Cold Filter Plugging Point. European Standard EN 590 defines CFPPs for various classes. Individual classes are then selected according to the country and season (summer/winter) in which the fuel is to be used.

At one time owners frequently improved the cold response of diesel fuel by adding kerosene or regular gasoline to the contents of their vehicles' fuel tanks. Fuels conforming to modern standards have rendered this measure unnecessary, and it is no longer recommended by vehicle manufacturers.

Flash point
The flash point is the temperature at which the quantities of vapor which a combustible fluid emits into the atmosphere are sufficient to allow a spark to ignite the vapor-air mixture above the fluid. Safety considerations (transport, storage) dictate that diesel fuels must meet the requirements for Class A III (flash point >55 °C). Less than 3% gasoline in the diesel fuel is sufficient to lower the flash point to such an extent that ignition becomes possible at room temperature.

Boiling range
The boiling range affects several parameters of major importance in determining the diesel fuel's operating characteristics. Extending the boiling range downward to embrace lower temperatures improves the fuel's cold-operation properties, but at the price of a reduction in the cetane number. Particularly critical is the negative effect on the fuel's lubrication properties, with the attendant increase in the risk of wear to injection-system components. In contrast, raising the final temperature at the upper end of the boiling range, although desirable from the standpoint of efficient utilization of petroleum resources, also results in higher soot emissions as well as carbon deposits (combustion residue) in the injection nozzles.

Density
There is a reasonably constant correspondence between a diese fuel's calorific value and its density; higher densities have a higher calorific value. Assuming constant injection-pump settings (and thus constant injection volume), the use of fuels with widely differing densities in a given system will be accompanied by variations in mixture ratios stemming from fluctuations in calorific value. Higher densities provoke increased particulate emissions, while lower densities lead to reductions in engine output.

Viscosity
Leakage losses in the injection pump result if viscosity is too low, and this in turn results in lower power, while on the other hand high viscosity has a detrimental effect on fuel atomization. The viscosity of diesel fuels, therefore, is limited to a narrow bandwidth.

Lubricity
The hydrodynamic lubricity of diesel fuels is not as important as that in the mixed-friction range. The introduction of environmentally compatible, hydrogenation-desulfurized fuels resulted in huge wear problems with distributor injection pumps in the field. Lubricity enhancers have to be added to the fuel to avoid these problems. Lubricity is tested using a high-frequency reciprocating rig (HFRR). Minimum lubricity is defined in EN 590.

Sulfur
Diesel fuels contain chemically bonded sulfur, with the actual quantities depending upon the quality of the crude petroleum and the components which are added at the refinery. Cracked compo-

PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

nents are generally characterized by especially high sulfur contents, but these can be reduced with hydrogen treatment at the refinery. The sulfur converted to sulfur dioxide (SO_2) during combustion in the engine is undesirable due to its environmentally harmful acidic reactions. For this reason, the legal limits placed on sulfur levels in diesel fuels have been progressively tightened in recent years. As of 1 January, 2000, the European limit is 350mg/kg. Reducing particulate mass (sulfate in the soot) is another reason why vehicles with oxidation catalytic converters require fuel with a low sulfur content. The same applies in terms of avoiding damage to the converters used for removing nitrogen from the exhaust gas (NO_x treatment). Sulfur contents < 10 ppm are desirable.

Carbon-deposit index (carbon residue)
The carbon-deposit index describes the fuel's tendency to form residues on the injection nozzles. The mechanisms of deposit formation are complex and not easily described.

The components (particularly the cracking constituents) which the diesel fuel contains at the end of vaporization exercise a not inconsiderable influence on deposit formation.

Additives
Additives, long a standard feature in gasolines, have attained increasing significance as quality improvers for diesel fuels. The various agents are generally combined in additive packages to achieve a variety of objectives. As the total concentration of the additives generally lies < 0.1 %, the fuel's physical characteristics – such as density, viscosity and boiling range – remain unchanged.

Flow improvers: Flow improvers are polymers whose application is generally restricted to winter (see Cold-flow properties).

Cetane improvers: Cetane improvers are added to promote good ignition response; they are in the form of alcohol-derived esters of azotic acid. These agents improve the fuel's combustion properties, with positive effects upon noise and particulate emissions.

Detergent additives: Detergent additives help clean the intake system to ensure efficient mixture preparation and inhibit the formation of carbon deposits on the injection nozzles.

Corrosion inhibitors: Corrosion inhibitors resist corrosion on metallic components (when fuel-borne moisture is entrained into the system).

Anti-foaming agents: Anti-foaming agents prevent the fuel foaming, so it is easier to refuel the vehicle.

Environmentally compatible diesel fuels

Some countries and regions (including Sweden and California) are promoting reductions in exhaust emissions by providing fiscal incentives for the use of "ecologically sound/environmentally compatible" fuels and/or making their use mandatory. In these fuels the final boiling point and the aromatic content are reduced, while sulfur is virtually eliminated. Introduction of these fuels was accompanied by serious problems stemming from component wear and elastomer damage in fuel-injection systems. Special additives are required to prevent this type of damage.

Alternative fuels

Coal hydrogenation
The essential raw materials are coal and coke. These emerge from initial processing in the form of water gas (H_2 + CO), while subsequent catalytic conversion produces hydrocarbons. These, in turn, provide the basic materials for the production of gasoline and diesel fuel. The byproducts are liquefied petroleum gas (LPG) and paraffin. Fischer-Tropsch synthesis has acquired particular significance in South Africa, where it is in widespread use in large industrial installations.

PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

Properties of liquid fuels and hydrocarbons

Substance	Density	Main constituent	Boiling temperature	Latent heat of evaporation	Specific calorific value	Ignition temperature	Theoretical air requirement	Ignition limit Lower	Ignition limit Upper
	kg/l	% by weight	°C	kJ/kg[1])	MJ/kg[1])	°C	kg/kg	% by vol. of gas in air	
Spark-ignition engine fuel,									
Regular	0.715...0.765	86 C, 14 H	25...215	380...500	42.7	≈ 300	14.8	≈ 0.6	≈ 8
Premium	0.730...0.780	86 C, 14 H	25...215	–	43.5	≈ 400	14.7	–	–
Aviation fuel	0.720	85 C, 15 H	40...180	–	43.5	≈ 500	–	≈ 0.7	≈ 8
Kerosene	0.77...0.83	87 C, 13 H	170...260	–	43	≈ 250	14.5	≈ 0.6	≈ 7.5
Diesel fuel	0.815...0.855	86 C, 13 H	180...360	≈ 250	42.5	≈ 250	14.5	≈ 0.6	≈ 7.5
Crude oil	0.70...1.0	80...83 C, 10...14 H	25...360	222...352	39.8...46.1	≈ 220	–	≈ 0.6	≈ 6.5
Lignite tar oil	0.850...0.90	84 C, 11 H	200...360	–	40.2...41.9	–	13.5	–	–
Bituminous coal oil	1.0...1.10	89 C, 7 H	170...330	–	36.4...38.5	–	–	–	–
Pentane C_5H_{12}	0.63	83 C, 17 H	36	352	45.4	285	15.4	1.4	7.8
Hexane C_6H_{14}	0.66	84 C, 16 H	69	331	44.7	240	15.2	1.2	7.4
n-Heptane C_7H_{16}	0.68	84 C, 16 H	98	310	44.4	220	15.2	1.1	6.7
Iso-octane C_8H_{18}	0.69	84 C, 16 H	99	297	44.6	410	15.2	1	6
Benzene C_6H_6	0.88	92 C, 8 H	80	394	40.2	550	13.3	1.2	8
Toluene C_7H_8	0.87	91 C, 9 H	110	364	40.6	530	13.4	1.2	7
Xylene C_8H_{11}	0.88	91 C, 9 H	144	339	40.6	460	13.7	1	7.6
Ether $(C_2H_5)_2O$	0.72	64 C, 14 H, 22 O	35	377	34.3	170	7.7	1.7	36
Acetone $(CH_3)_2CO$	0.79	62 C, 10 C, 28 O	56	523	28.5	540	9.4	2.5	13
Ethanol C_2H_5OH	0.79	52 C, 13 H, 35 O	78	904	26.8	420	9	3.5	15
Methanol CH_3OH	0.79	38 C, 12 H, 50 O	65	1110	19.7	450	6.4	5.5	26

Viscosity at 20 °C in mm²/s (= cSt, P. 27): gasoline ≈ 0.6; diesel fuel ≈ 4; ethanol ≈ 1.5; methanol 0.75.

[1]) Values per *l* = values per kg x density in kg/*l*. See P. 25 for conversion tables.

PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

Properties of gaseous fuels and hydrocarbons

Substance	Density at 0 °C and 1013 mbar	Main constituents	Boiling temp. at 1013 mbar	Specific calorific value Fuel	Specific calorific value A/F mixture	Ignition temperature	Theoretical air requirement	Ignition limit Lower	Ignition limit Upper
	kg/m³	% by weight	°C	MJ/kg[1])	MJ/m³ [1])	°C	kg/kg	% by vol. of gas in air	
Liquefied gas	2.25[2])	C_3H_8, C_4H_{10}	–30	46.1	3.39	≈ 400	15.5	1.5	15
Municipal gas	0.56...0.61	50 H, 8 CO, 30 CH_4	–210	≈ 30	≈ 3.25	≈ 560	10	4	40
Natural gas	≈ 0.83	76 C, 24 H	–162	47.7	–	–	–	–	–
Water gas	0.71	50 H, 38 CO	–	15.1	3.10	≈ 600	4.3	6	72
Blast-furnace gas	1.28	28 CO, 59 N, 12 CO_2	–170	3.20	1.88	≈ 600	0.75	≈ 30	≈ 75
Sewage gas) [3])	–	46 CH_4, 54 CO_2	–	27.2[3])	3.22	–	–	–	–
Hydrogen H_2	0.090	100 H	–253	120.0	2.97	560	34	4	77
Carbon monoxide CO	1.25	100 CO	–191	10.05	3.48	605	2.5	12.5	75
Methane CH_4	0.72	75 C, 25 H	–162	50.0	3.22	650	17.2	5	15
Acetylene C_2H_2	1.17	93 C, 7 H	–81	48.1	4.38	305	13.25	1.5	80
Ethane C_2H_6	1.36	80 C, 20 H	–88	47.5	–	515	17.3	3	14
Ethene C_2H_4	1.26	86 C, 14 H	–102	14.1	–	425	14.7	2.75	34
Propane C_3H_8	2.0[2])	82 C, 18 H	–43	46.3	3.35	470	15.6	1.9	9.5
Propene C_3H_6	1.92	86 C, 14 H	–47	45.8	–	450	14.7	2	11
Butane C_4H_{10}	2.7[2])	83 C, 17 H	–10; +1[4])	45.6	3.39	365	15.4	1.5	8.5
Butene C_4H_8	2.5	86 C, 14 H	–5; +1[4])	45.2	–	–	14.8	1.7	9
Di-methylether C_2H_6O	2.05[5])	52 C, 13 H, 35 O	–25	28.8	3.43	235	9.0	3.4	18.6

[1]) Values per m³ = values per kg x density in kg/m³. See P. 25 for conversion tables.
[2]) Density of liquefied gas 0.54 kg/l, density of liquefied propane 0.51 kg/l, density of liquefied butane 0.58 kg/l.
[3]) Purified sewage gas contains 95 % CH_4 (methane) and has a heating value of 37.7 MJ/kg.
[4]) First value for isobutane, second value for n-butane or n-butene.
[5]) Density of liquefied dimethylether: 0.667 kg/l.

PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

Table 2. Essential data for diesel fuels, European Standard EN 590 (valid since 1 January 2000)

Requirements	Unit	Specification
Flame point, min.	°C	55
Water, max.	mg/kg	200
Sulfur content, max.	mg/kg	350
Lubricity, "wear scardiameter" max.	µm	460
For temperate climates:		
Density (at 15 °C), min./max.	kg/m³	820/845
Viscosity (at 40 °C), min./max.	mm²/s	2/4.5
Cetane number, min.	–	51
Cetane index, min.	–	46
up to 250 °C, distilled, max.	% volume	65
up to 350 °C, distilled, min.	% volume	85
up to 360 °C, distilled, min.	% volume	95
CFPP[1]) in 6 classes A...F, max.	°C	+5 ... –20
For arctic climates (in 5 classes 0 ... 4):		
Density (at 15 °C), min./max.	kg/m³	800/845 ... 800/840
Viscosity (at 40 °C), min./max.	mm²/s	1.5/4 ... 1.2/4
Cetane number, min.	–	49 ... 47
Cetane index, min.	–	46 ... 43
up to 180 °C, distilled, max.	% volume	10
up to 340 °C, distilled, min.	% volume	95
CFPP[1]), max.	°C	–20 ... –44

[1]) Filtration limit.

Liquefied Petroleum Gas (LPG)
The two major components of LPG are butane and propane. It is in limited use as a fuel for motor vehicles. LPG is a byproduct of the petroleum-refining process, and can be liquefied under pressure. LPG is distinguished by a high octane number (RON >100).

CNG (Compressed Natural Gas)
This usually assumes the form of compressed methane, and is suitable for obtaining ultra-low emissions from combustion engines. Due to its high H/C ratio (relative to other fuels), CNG also produces less CO_2. CNG is being tested in both diesel and spark-ignition engines. Soot production is virtually zero when this fuel is combusted in the diesel cycle. The vehicles must be specially adapted for operation on this type of alternative fuel.

Alcohol fuels
Methanol, ethanol, and their various byproducts (e.g. ether) are the main contenders as alternative fuels for spark-ignition engines, being the focus of both discussion and actual application. Methanol can be derived from the plentiful hydrocarbon reserves represented by coal, natural gas, and heavy oils, etc. In certain countries (such as Brazil, but also in the USA), biomass (sugar cane, wheat) is distilled to produce ethanol for use as an engine fuel and fuel additive.

Methanol additives ≤3 % by volume are permissible in many countries. Methanol, however, should be used only in conjunction with solubilizers. Fuels with higher concentrations (15 ... 85 %) have been subjected to large-scale testing (flexible fuel).

Ethanol fuel mixtures are commercially available in countries with high agricultural yield (e.g. E22 in Brazil).

Both calorific value and other factors differ from those of conventional fuels, so special vehicle modifications are sometimes required.

Dimethylether (DME)
This is a synthetic product with a high

PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

cetane number, producing little soot and reduced nitrogen oxide when combusted in diesel engines. Its calorific value is low, on account of its low density and high oxygen content. It is a gas-phase fuel, so the fuel-injection equipment has to be modified. Other ethers (dimethoxymethane, di-n-pentylether, and others) are also being tested.

Emulsions
Emulsions of water and alcohols in diesel fuels are undergoing testing by a number of different institutes. Water and alcohols (particularly methanol) are difficult or impossible to mix with diesel. Emulsifiers are required to keep the mixture stable and prevent it from de-emulsifying. Corrosion-inhibiting measures are also necessary. Soot and nitrogen-oxide emissions can be reduced by these emulsions, but to date their use has been restricted to certain fleet applications; there has been no broad-based testing for a variety of injection systems.

Fatty-acid methyl ester (FAME)
Fatty-acid methyl ester is the generic term applied to all vegetable and animal oils and greases transesterified with methanol, including rapeseed (RME), soy, and sunflower methyl esters, among others. The use of vegetable oils for combustion in diesel engines is a field of investigation because it would allow agricultural areas to be used for the production of regenerative energy. Considerable problems arise with pure vegetable oils, due to their high viscosity and severe tendency to cause nozzle coking; oils transesterified with methanol are free of these difficulties.

Fatty-acid methyl esters are used in pure form and as additives for diesel fuel (≤5%). In concentrations as low as 1...2% they boost the lubricity of the fuels. A definitive standard for these substances is now in preparation. This standard will have to lay down characteristics, stability and maximum permissible contaminants in order to ensure trouble-free operation of the injection equipment and engines. FAME fuels are uneconomical in comparison with conventional fuels and their use will require subsidization.

Antifreeze and brake fluid

Brake fluids

Brake fluid is the hydraulic medium employed to transmit actuation forces within the braking system. Compliance with stringent requirements is essential to ensure reliable brake-system operation. These requirements are defined in various standards of similar content (SAE J 1703, FMVSS 116, ISO 4925). The performance data contained in FMVSS 116 (Federal Motor Vehicle Safety Standard), mandatory in the US, also serve as an international reference. The US Department of Transportation (DOT) has defined specific ratings for salient characteristics (Table 1).

Requirements

Equilibrium boiling point
The equilibrium boiling point provides an index of the brake fluid's resistance to thermal stress. The heat encountered in the wheel cylinders (with the highest temperatures in the entire braking system) can be especially critical. Vapor bubbles can form at temperatures above the brake fluid's instantaneous boiling point, resulting in brake failure.

Wet boiling point
The wet boiling point is the fluid's equilibrium boiling point subsequent to moisture absorption under specified conditions (approx. 3.5%). Hygroscopic (glycol-based) fluids respond with an especially pronounced drop in boiling point. The wet boiling point is tested to quantify the response characteristics of used brake fluid. Brake fluid absorbs moisture, mostly by diffusion through brake-system hoses. This is the main reason why it should be replaced every 1...2 years. The illustration shows the reductions in boiling point that result from moisture absorption in two different brake fluids.

PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

Table 1. Brake fluids

Reference standard for testing	FMVSS 116			SAE J1703
Requirements/Date	DOT3	DOT4	DOT5	Nov. 1983
Dry boiling point min. °C	205	230	260	205
Wet boiling point min. °C	140	155	180	140
Cold viscosity at –40 °C mm²/s	1500	1800	900	1800

Viscosity

To ensure consistent reliability throughout the braking system's extended operating range (–40 °C ... +100 °C), the viscosity should remain as constant as possible, with minimal sensitivity to temperature variations. Maintaining the lowest possible viscosity at very low temperatures is especially important in ABS/TCS/ESP systems.

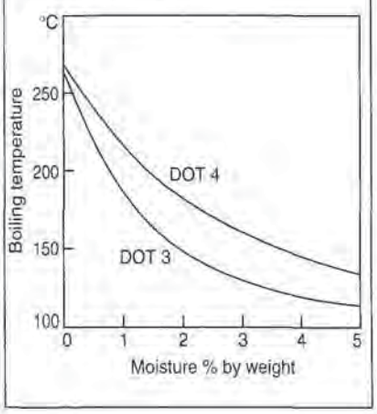
Compressibility

The fluid should maintain a consistently low level of compressibility with minimal sensitivity to temperature fluctuations.

Corrosion protection

FMVSS 116 stipulates that brake fluids shall exercise no corrosive effect on those metals generally employed in braking systems. The required corrosion protection can be achieved only by using additives.

Brake-fluid boiling points as a function of moisture content



Elastomer swelling

The elastomers employed in a particular brake system must be able to adapt to the type of brake fluid being used. Although a small amount of swelling is desirable, it is imperative that it not exceed approximately 10 %. Otherwise, it has a negative effect on the strength of the elastomer components. Even minute levels of mineral-oil contamination (such as mineral-oil-based brake fluid, solvents) in glycol-based brake fluid can lead to destruction of rubber components (such as seals) and ultimately lead to brake-system failure.

Chemical composition

Glycol-ether fluids

Most brake fluids are based on glycol-ether compounds. These generally consist of monoethers of low polyethylene glycols. Although these components can be used to produce a brake fluid which conforms to the DOT 3 requirements, their undesirable hygroscopic properties cause this fluid to absorb moisture at a relatively rapid rate, with an attendant swift reduction in the boiling point.

If the free OH (hydroxyl) groups are partially esterified with boric acid, the result is a superior DOT 4 (or "DOT4+", "Super DOT4", "DOT5.1") brake fluid capable of reacting with moisture to neutralize its effects. As the DOT 4 brake fluid's boiling point drops much more slowly than that of a DOT 3 fluid, its service life is longer.

Mineral-oil fluids (ISO 7308)

The great advantage of mineral-oil-based fluids is the fact that they are not hygroscopic, so the boiling point does not drop owing to moisture absorption. The mineral and synthetic oils for this fluid must be selected with utmost care. Viscosity-index improvers are generally added to

PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

achieve the desired relationship between viscosity and temperature.

The petroleum industry can also supply a range of further additives to improve other brake-fluid properties. It should be noted that mineral-oil-based fluids should never be added to braking systems designed for glycol ether (or vice versa), as this destroys the elastomers.

Silicone fluids (SAE J1705)

Because silicone fluids – like mineral oils – do not absorb moisture, they formerly saw occasional use as brake fluids. The disadvantages of these products include considerably higher compressibility and inferior lubrication, both of which reduce their suitability for use as hydraulic fluid in many systems. A critical factor with brake fluids based on silicone or mineral oils is the absorption of free water in a fluid state, as the water forms vapor bubbles when it heats up to more than 100 °C and freezes when it cools to less than 0 °C.

Coolants

Requirements

The cooling system must dissipate that part of the engine's combustion heat that is not converted into mechanical energy. A fluid-filled cooling circuit transfers the heat absorbed in the cylinder head to a heat exchanger (radiator) for dispersal into the air. The fluid in this circuit is exposed to extreme thermal loads; it must also be formulated to ensure that it does not attack the materials within the cooling system (corrosion).

Owing to its high specific heat and its correspondingly substantial thermal-absorption capacity, water is a very good cooling medium. Its disadvantages include its corrosive properties and limited suitability for application in cold conditions (freezing).

This is why additives must be mixed with the water for satisfactory performance.

Table 2. Ice flaking and boiling points for water-glycol mixtures

Glycol Vol. %	Ice flaking point °C	Boiling point °C
10	–4	101
20	–9	102
30	–17	104
40	–26	106
50	–39	108

Antifreeze

It is possible to lower the coolant's freezing point by adding ethylene glycol. When glycol is added to form a mixture with water the resulting coolant no longer freezes at a given temperature. Instead, ice crystals are precipitated in the fluid once the temperature drops to the flaking point. At this temperature the fluid medium can still be pumped through the cooling circuit. Glycol also raises the coolant's boiling point (Table 2).

In their owner's manuals automobile manufacturers usually specify various optional antifreeze mixture ratios for different levels of low-temperature frost protection.

Additives

Coolants must include effective additives to protect the glycol against oxidation (with formation of extremely corrosive byproducts) and to protect metallic cooling-system components against corrosion. Common additives include:

- Corrosion inhibitors: silicates, nitrates, nitrites, metal salts of organic acids, benzthiazole derivates,
- Buffers: borates,
- Antifoaming agents: silicones.

Many of these additives are subject to aging deterioration, leading to a gradual reduction in coolant performance. The manufacturers have responded to this fact by granting official approval exclusively for coolants of proven long-term stability.

PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

Names of chemicals

Hazard codes: E = explosive, O = flammable, F = readily flammable, F_+ = highly flammable, C = caustic, X_n = slightly toxic, X_i = irritant, T = toxic, T_+ = highly toxic.

Commercial name English (hazard code)	German	French	Chemical name	Chemical formula
(Glacial) acetic acid (C)	Eisessig (Essigessenz)	Acide acétique glacial	Acetic acid	CH_3COOH
Acetic ether; vinegar naphtha (F)	Essigester (Essigäther)	Ether acétique	Ethylacetate	$CH_3COOC_2H_5$
Aerosil (fumed silica)	Aerosil,		Silicon dioxide in extremely fine particles	SiO_2
Ammonia liquor (X_i, C)	Salmiakgeist	Ammoniaque hydroxide	Aqueous solution of ammonium hydroxide	NH_4OH in H_2O
Anon; pimelic ketone (X_n)	Anon	Anone	Cyclohexanone	$C_6H_{10}O$
Aqua fortis (C)	Scheidewasser	Eau forte	Nitric acid (50 % aqueous solution)	HNO_3 in H_2O
Aqua regia (C, T_+)	Königswasser	Eau régale	Mixture of nitric acid and hydrochloric acid	$HNO_3 + HCl$ (1 + 3)
Bitter almond oil (T)	Bittermandelöl	Essence d'amandes amères	Benzaldehyde	C_6H_5CHO
Bleach (C)	Chlorkalk	Chlorure de chaux	Calciumchloride-hypochlorite	$Ca(OCl)Cl$
Chloride of lime				
Blue vitriol	Kupfervitriol	Vitriol bleu	Copper sulphate pentahydrate	$CuSO_4 \cdot 5H_2O$
Borax (tincal)	Borax (Tinkal)	Borax (tincal)	Sodium tetraborate	$Na_2B_4O_7 \cdot 10H_2O$
Butoxyl	Butoxyl®		(3-methoxybutyl) acetate	$CH_3COO(CH_2)_2CH(OCH_3)CH_3$
Butter of tin (C)	Zinnbutter	Beurre d'étain	Tin tetrachloride	$SnCl_4 \cdot 5H_2O$
Calcium carbide (F)	Karbid	Carbure de calcium	Calcium carbide	CaC_2
Calomel (X_n)	Kalomel	Calomel	Mercury(I) chloride	Hg_2Cl_2
Carbitol acetate™ [1])	Carbitolacetat®		Diethylene glycol ethylether acetate	$CH_3COOCH_2CH_2OCH_2CH_2OC_2H_5$
Carbitol™ [1]) (solvent)	Carbitol®, Dioxitol®	Carbitol	Diethylene glycol monoethylether	$HOCH_2CH_2OCH_2CH_2OC_2H_5$

PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

Carbolic acid (T)	Karbolsäure	Acide carbolique	Phenol	C_6H_5OH
Caustic potash (C)	Ätzkali	Potasse caustique	Potassium hydroxide	KOH
Caustic soda (C)	Ätznatron	Soude caustique	Sodium hydroxide	NaOH
Cellosolve™ [1]) (solvent)	Cellosolve®; Oxitol®		Ethylene glycol monoethylether	$HOCH_2CH_2OC_2H_5$
Cellosolve™ [1]) acetate	Cellosolveacetat®		Ethylene glycol ethylether acetate	$CH_3COOCH_2CH_2OC_2H_5$
Chalk	Kreide	Craie	Calcium carbonate	$CaCO_3$
Chloramine-T (X_l)	Chloramin T	Chloramine-T	Sodium salt of the p-toluene-sulfonic acid chloroamide	$Na[CH_3C_6H_4SO_2NCl] \cdot 3H_2O$
Chloroprene (F, X_n)	Chloropren	Chloroprène	2-chloro 1.3-butadiene	$CH_2{=}CClCH{=}CH_2$
Chlorothene™; ("1.1.1"); (X_n)	Chlorothene®	Chlorothène	1.1.1-trichloroethane	Cl_3CCH_3
Methylchloroform	Methylchloroform			
Chromic anhydride (C, O, X_l)	Chromsäure	Anhydride chromique	Chromium trioxide (chromic acid anhydride)	CrO_3
Cinnabar	Zinnober	Cinabre; vermillon	Mercury (II) sulfide	HgS
Colophony; rosin	Kolophonium	Colophane	Naturally occurring abietic acid	$C_{19}H_{29}COOH$
Corrosive sublimate (T)	Sublimat	Sublimé corrosif	Mercury (II) chloride	$HgCl_2$
Cryolite (X_n)	Kryolith	Cryolite	Sodium hexafluoroaluminate	$Na_3[AlF_6]$
Decalin	Dekalin	Décaline	Decahydro naphthalene	$C_{10}H_{18}$
Diane	Bisphenol A; Diphenylolpropan		Dihydroxyphenyl propane-2.2	$(CH_3)_2C(C_6H_4(OH)\text{-}4)_2$
Diisobutylene (F)	Diisobutylen	Diisobutylène	2.4.4-trimethyl pentenes 1 and 2	$(CH_3)_3CCH_2C(CH_3){=}CH_2$ and $(CH_3)_3C\text{-}CH{=}C(CH_3)_2$
DMF (X_n)	DMF	DMF	N,N-dimethyl formamide	$HCON(CH_3)_2$
DMSO	DMSO	DMSO	Dimethyl sulfoxide	$(CH_3)_2SO$
Dry ice	Trockeneis	Carboglace	(Solid) carbon dioxide	CO_2
English red	Polierrot	Rouge d'Angleterre	Iron (III) oxide	Fe_2O_3
Epsomite, bitter salt	Bittersalz (Magnesiumvitriol)	Epsomite	Magnesium sulfate	$MgSO_4 \cdot 7H_2O$

[1]) methyl-, propyl-, i-propyl-, butyl-c.: names for analogous ethers containing the above mentioned groups instead of ethyl-.

PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

Names of chemicals (continued)

Commercial name			Chemical name	Chemical formula
English	German	French		
Fixing salt; (hypo)	Fixiersalz ("Antichlor")	Sel fixateur	Sodium thiosulfate	$Na_2S_2O_3 \cdot 5H_2O$
Fluorspar; fluorite	Flußspat; Fluorit	Spath fluor; fluorine	Calcium fluoride	CaF_2
Formalin (T)	Formalin®	Formol	Aqueous solution of formaldehyde	H_2CO in H_2O
Freon™(es)	Freon®(e); Frigen®(e)	Fréon(es); Frigène(s)	Compounds of C, H, F, Cl, (Br)	Numerical characterization[3])
Glauber's salt; mirabilite	Glaubersalz	Sel de Glauber	Sodium sulphate	$Na_2SO_4 \cdot 10H_2O$
Glycol (X_i)	Glysantin®; Glykol	Glycol	1.2-ethanediol	$HOCH_2CH_2OH$
Golden antimony sulphide	Goldschwefel	Soufre doré d'antimoine	Antimony (V)-sulfide	Sb_2S_5
Green vitriol	Eisenvitriol	Vitriol vert; couperose verte	Iron (II) sulfate	$FeSO_4 \cdot 7H_2O$
Halone(s)	Halon(e)	Halon(s)	Compounds of C, F, Cl, Br	Numerical characterization[3])
Halon™	Halon®		Polymer from tetrafluorethylene	$(C_2F_4)_n$
Halothane	Halothan		2-bromine 2-chlorine 1,1,1-tetrafluoroethane	$F_3CCHClBr$
Hartshorn salt	Hirschhornsalz	Sel volatil d'Angleterre	Ammonium hydrogen carbonate + ammonium carbonate	$(NH_4)HCO_3 + (NH_4)_2CO_3$
Hexalin	Hexalin®	Hexaline	Cyclohexanol	$C_6H_{11}OH$
Hexone (F)	Hexon; MIBK		(also: hexalhydronaphthalene) 4-methylpentanone 2 (Methylisobutylketon)	$(C_{10}H_{18})$ $(CH_3)_2CHCH_2COCH_3$
Hydrochloric acid (C)	Salzsäure	Esprit de sel	Aqueous solution of hydrogen chloride	HCl in H_2O
Hydrofluoric acid (T, C)	Flußsäure	(Acide fluorhydrique)	Aqueous solution of hydrogen fluoride	HF in H_2O
Hydrogen peroxyde	Perhydrol®	Eau oxygénée	Hydrogen dioxide	H_2O_2
Laughing gas (O)	Lachgas ("Stickoxydul")	Gaz hilarant	Nitrous oxide	N_2O
Lead sugar (X_n)	Bleizucker	Sel de Saturne	Lead acetate	$Pb(CH_3COO) \cdot 3H_2O$

PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

Lead vinegar (X_n)	Bleiessig	Eau blanche; vinaigre de plomb	Aqueous solution of lead acetate and lead hydroxide	$Pb(CH_3COO)_2 \cdot Pb(OH)_2$
Libavius' fuming spirit (C)	Spiritus fumans Libavii		Tin (IV) chloride	$SnCl_4$
Lime saltpeter	Salpeter, Kalk-; Norge-	Salpêtre	Calcium nitrate	$Ca(NO_3)_2 \cdot 4H_2O$
Liquid gas (F)	Flüssiggas	Gaz liquéfié	Propane, isobutane, n-butane	$C_3H_8 + C_4H_{10}$
Lunar caustic (C)	Höllenstein; "lapis infernalis"	Pierre infernale	Silver nitrate	$AgNO_3$
Marble	Marmor	Marbre	Calcium carbonate	$CaCO_3$
Microcosmic salt	Phosphorsalz		Sodium ammonium hydrogenphosphate	$NH_4NaHPO_4 \cdot 4H_2O$
Mine gas (F)	Grubengas; Sumpfgas	Grisou; gaz des marais	Methane	CH_4
Minium	Mennige	Minium	Lead (II) orthoplumbate	Pb_3O_4 (Pb_2PbO_4)
Mohr's salt	Mohr'sches Salz	Sel de Mohr	Iron (II) ammonium sulfate	$(NH_4)_2[Fe(SO_4)_2] \cdot 6H_2O$
Mordant rouge; mordant salt[4]	Tonerde, essigsaure	Mordant d'alun	Basic aluminum acetate	$(CH_3COO)_2AlOH$
Mota (X_n)	Meta®	Alcool solidifié	Tetramethyltetroxacyclooctane (metaldehyde)	$(CHCH_3)_4O_4$
Muthmann's liquid	Muthmann's Flüssigkeit		1,1,2,2-tetrabromomethane	$Br_2CHCHBr_2$
Nitroglycerin (E, F)	Nitroglycerin	Nitroglycérine	Glycerol trinitrate	$CHONO_2(CH_2ONO_2)_2$
Nitrolim; lime nitrogen	Kalkstickstoff	Chaux azotée	Calcium cyanamide	$CaCN_2$
Norway saltpeter	Salpeter, Ammon-	Nitrate d'ammonium	Ammonium nitrate	NH_4NO_3

[2]) Numerical codes for freons (fluorine-chlorine derivatives of methane and ethane, CH_4 and C_2H_6):
Number in hundreds column = number of carbon atoms – 1
Number in tens column = number of hydrogen atoms + 1
Number in ones column = number of fluorine atoms
The missing atoms for valence saturation are chlorine atoms. Examples: F 113 = $C_2F_3Cl_3$; F 21 = $CHFCl_2$

[3]) Numerical codes for halones (fully halogenated hydrocarbons):
Number in thousands column = number of C atoms
Number in hundreds column = number of F atoms
Number in tens column = number of Cl atoms
Number in ones column = number of Br atoms
Examples: halone 1211 = CF_2ClBr; halone 2402 = $C_2F_4Br_2$

[4]) Mordanting agent, for instance for dyeing textiles red. See Hawley's Condensed Chemical Dictionary, 11th ed. 1987, "Aluminium acetate".

PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

Names of chemicals (continued)

Commercial name			Chemical name	Chemical formula
English	German	French		
Oleum (C)	Oleum ("Vitriolöl")	Oléum	Sulfuric acid + disulfuric acid	$H_2SO_4 + H_2S_2O_7$
Oxalic acid (X_n)	Kleesäure	Acide oxalique	Oxalic acid	$(COOH)_2 \cdot 2H_2O$
Phosgene (T)	Phosgen	Phosgène	Carbonic acid dichloride	$COCl_2$
Phosphine (T+)	Phosphin	Phosphine (sel de phosphore)	Hydrogen phosphide	PH_3
Picric acid (T, E)	Pikrinsäure	Acide picrique	2,4,6-trinitrophenole	$C_6H_2(NO_2)_3OH$
Potash	Pottasche	Potasse	Potassium carbonate	K_2CO_3
Potash alum	Alaun, Kali-	Alun de potassium	Potassium aluminium sulfate	$KAl(SO_4)_2 \cdot 12H_2O$
Potassium chlorate (O, X_n)	Knallsalz	Sel de Berthollet	Potassium chlorate	$KClO_3$
Potassium metabisulphite	Kaliummetabisulfit	Métabisulfite de potassium	potassium disulfite	$K_2S_2O_5$
Pyrolusite (X_n)	Braunstein	Pyrolusite	Manganese dioxide	MnO_2
Quicklime; burnt lime; caustic lime	Kalk, gebrannter	Chaux vive	Calcium oxide	CaO
Red prussiate of potash	Blutlaugensalz, rotes; Kaliumferricyanid	Ferricyanure de potassium	Potassium hexacyanoferrate (III)	$K_3[Fe(CN)_6]$
Rochelle salt; salt of Seignette	Seignettesalz (Natronweinstein)	Sel de Seignette	potassium sodium tartrate	$(CHOH)_2COOKCOONa$
Sal ammoniac (X_n)	Salmiak (Salmiaksalz)	salmiac	Ammonium chloride	NH_4Cl
Saltpeter	Natronsalpeter Salpeter, Kali-	Saltpêtre	Potassium nitrate	KNO_3
Silica gel with indicator	Blaugel (Silicagel)	Gel bleu	Porous silicon dioxide with humidity indicator	SiO_2 with cobalt compound
Slaked lime	Kalk, gelöschter	Chaux éteinte	Calcium hydroxide	$Ca(OH)_2$
Soda crystals	Soda (Kristall-)	Soude (cristaux de)	Sodium carbonate	$Na_2CO_3 \cdot 10H_2O$
Soda niter; Chile saltpeter	Salpeter, Chile;	Salpêtre du Chili	Sodium nitrate	NaNO3

PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

Sorrel salt; potassium binoxalate	Kleesalz	Sel d'oseille	Potassium tetraoxalate	$(HOOCCOOK) \cdot (COOH)_2 \cdot 2H_2O$
Sulphuric ether (F)	Schwefelether	Ether sulfurique	Diethyl ether	$C_2H_5OC_2H_5$
Tetrachloroethylene (X_n)	Per	Tétrachloroéthylène	Tetrachloroethylene (perchloroethylene)	$Cl_2C{=}CCl_2$
Tetrachloromethane (T)	Tetra ("Tetraform")	Tetrachlorométhane	Perchloromethane	CCl_4
Tetralin (O, C)	Tetralin	Tétraline	1.2.3.4-tetrahydronaphthalene	$C_{10}H_{12}$
Tin foil	Stanniol	Papier d'étain	Foil tin	Sn
Tin salt	Zinnsalz	Sel d'étain	Tin (II) chloride	$SnCl_2 \cdot 2H_2O$
TNT; trotyl (E)	TNT	TNT; tolite	2.4.6-trinitro toluene	$C_6H_2(NO_2)_3CH_3$
Trichloroethylene (X_n)	Tri	Trichloréthylène	Trichloroethene	$Cl_2C{=}CHCl$
Urea	Harnstoff	Urée	Carbonic acid diamide	$CO(NH_2)_2$
Urotropine	Urotropin	Urotropine	1.3.5.7-Tetra acadamantane	$(CH_2)_6N_4$
Vichy salt; baking soda	Bullrichsalz®; Natron (Natriumbicarbonat)	Sel de Vichy	Sodium hydrogen carbonate	$NaHCO_3$
Waterglass	Wasserglas (Kali- bzw. Natron-)	Verre soluble	Aqueous solution of potassium or sodium silicates	$M_2SiO_3 + M_2Si_2O_5$ (M = K or Na)
Yellow prussiate of potash	Blutlaugensalz, gelbes; Kaliumferrocyanid	Ferrocyanure de potassium	Potassium hexacyanoferrate (II)	$K_4[Fe(CN)_6]$
	GB-Ester; Polysolvan O®		Glycol acid butyl ester (hydroxic aceiic acid butylester)	$HOCH_2COOC_4H_{10}$

PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

Corrosion and corrosion protection

Corrosion is the attrition of metal as a result of electrochemical reactions with substances in the environment. As it proceeds, metal atoms oxidize to form nonmetallic compounds in a process emanating from the affected material's surface. In thermodynamic terms, the process can be viewed as an entropic transition from an ordered, high-energy state into a less-ordered state of lower energy and consequently greater stability.

Corrosion processes are always interphase reactions. An example of this type of reaction is metal scaling, i.e. oxidation in hot gases. The following deals exclusively with the corrosion that occurs at the phase limit between the metal and aqueous phases, generally referred to as electrochemical corrosion.

Corrosive attack

Two distinct reactions occur in every corrosive attack: in the anodic subprocess – the directly visible corrosive effect – the difference in respective potentials causes the metal to oxidize as described in the reaction equation

$$Me \rightarrow Me^{n+} + ne^-,$$

freeing an equivalent number of electrons. The metal ions thus formed can either be dissolved in the electrolyte, or can precipitate out on the metal after reacting with constituents in the attacking medium.

This anodic subprocess can continue only as long as the electrons it produces are consumed in a second process. This second subprocess is a cathodic subreaction in which oxygen is reduced to hydroxyl ions in neutral or alkaline media, in accordance with the reduction equation

$$O_2 + 2\,H_2O + 4\,e^- \rightarrow 4\,OH^-.$$

These hydroxyl ions, in turn, are able to react with the metal ions, whereas in acidic media the hydrogen ions are reduced via the formation of free hydrogen, which escapes as a gas according to the following formula:

$$2\,H^+ + 2\,e^- \rightarrow H_2.$$

When two different metals moistened by the same fluid are in mutual electrical contact, the cathodic subprocess occurs at the more noble metal while the anodic subprocess progresses at the baser material. This is called contact corrosion.

However, it is also possible for both reactions to occur on the same metal in a process termed "free corrosion". Anodic and cathodic subprocesses can alternate continuously at the metal/solution interface with statistically random time and location distributions for the individual subprocesses.

Each of the individual subreactions corresponds to a partial current/voltage curve. The total current is the sum of the two currents I_a and I_c:

$$I_{total} = I_a + I_c$$

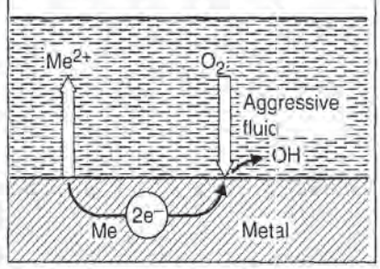
The two partial current/voltage curves combine to produce the cumulative current/voltage curve.

If no voltage is supplied externally, i.e., in free corrosion, the system assumes a state in which the anodic and cathodic partial currents are precisely balanced:

$$I_a = -I_c = I_{corr}$$

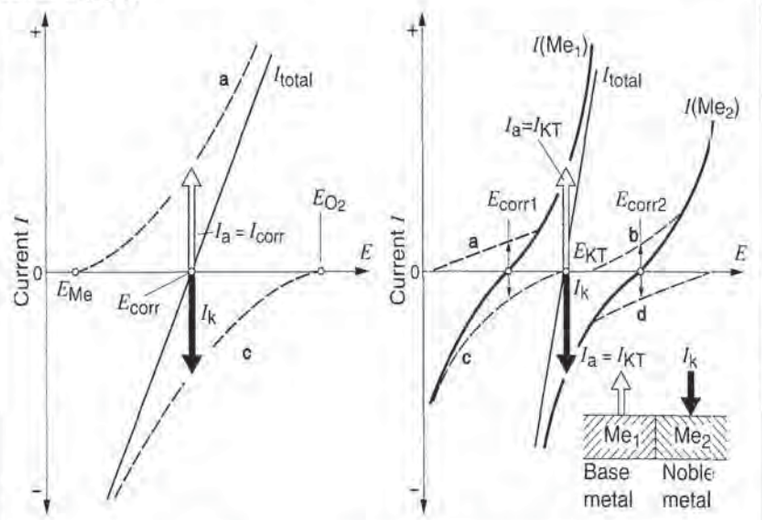
The anodic current is called the corrosion current I_{corr}, while the corresponding potential at which this current compensation

Free corrosion at the metal/corrosive-medium phase boundary
In an aggressive fluid, oxygen is reduced at the corroding metal and corrosion products are simultaneously formed.



PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

Partial and cumulative current/voltage curves for free corrosion (left) and contact corrosion (right)



a $Me_1 \longrightarrow Me_1^{++}+2e$ c $^1_2 O_2+H_2O+2e \longrightarrow 2OH^-$ (at Me_1)

b $Me_2 \longrightarrow Me_2^{++}+2e$ d $^1_2 O_2+H_2O+2e \longrightarrow 2OH^-$ (at Me_2)

occurs is called the "open-circuit potential" E_{corr}.

The open-circuit potential is a mixed potential in which there is no equilibrium; matter is converted continuously, as defined in the following general equation:

$$O_2 + 2\,H_2O + \frac{4}{n}\,Me \rightarrow \frac{4}{n}\,Me^{n+} + 4OH^-$$

The above is essentially applicable for contact corrosion, although the interrelationships are more complicated. In addition to pairs of partial current/voltage curves for each of the two metals and the resulting two cumulative current/voltage curves, the resultant total current/voltage curve for the entire system must be considered as a value which can be measured externally.

Electrochemical series of metals

Metals are often ranked in "electrochemical series of metals" corresponding to consecutively higher "standard potentials." Here, the term "standard potential" indicates that the specified values apply to standard conditions, especially to the activities (in the electrochemically active part of the concentration) of the dissolved metal ions and hydrogen in a concentration of 1 mol/l at a hydrogen pressure of 1 bar at 25°C. Such conditions are seldom found in practice; in fact, most solutions are virtually free of ions from the metal in question.

It should be emphasized that the table ("Standard electric potentials of the metals") is limited to thermodynamic values, and does not reflect the effects of corrosion kinetics, for example as encountered in the formation of protective layers. For instance, lead is shown as a base metal; as such it should dissolve in sulfuric acid. Although the "practical" and "technical" electrochemical series of metals do not have this disadvantage, their practical application range remains restricted. In contrast, electrochemical corrosion measurements provide unambiguous data.

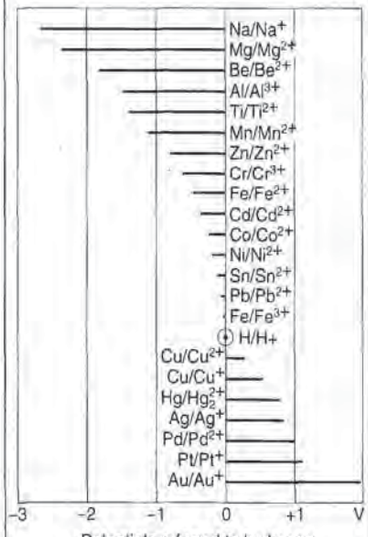
PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

For general reference purposes, the following relationships between electrical potential and corrosion susceptibility can be specified for metals with no application of external voltages provided that the metal's response is not affected by secondary reactions such as complexing reactions or the formation of protective layers:

Very base metals (potential less than – 0.4 V), e.g. Na, Mg, Be, Al, Ti and Fe, corrode in neutral aqueous solutions, even in the absence of oxygen.

Base metals (potential between – 0.5 and 0 V), e.g. Cd, Co, Ni, Sn and Pb, corrode in neutral aqueous solutions in the presence of oxygen, and corrode in acids to produce hydrogen even in the absence of oxygen.

Standard electrode potentials of the metals



Semi-noble metals (potential between 0 and + 0.7 V), e.g. Cu, Hg and Ag, corrode in all solutions only if oxygen is present.

Noble metals (potentials above + 0.7 V), e.g. Pd, Pt and Au, are generally stable.

These response patterns can vary sharply when external voltage is applied to the metals. The variations are exploited in the field of electrochemical corrosion protection (see corresponding section).

Types of corrosion

General surface corrosion
Uniform removal of material over the entire contact surface between the material and the attacking medium. This is a very frequent type of corrosion in which the material penetration rate (removal depth) can be calculated per unit of time based on the corrosion current.

Pitting corrosion
Limited localized attack by a corrosive medium which penetrates the material by forming holes, or pits, whose depth is almost always greater than their diameter. Practically no material is removed from the surface outside the pitted areas. Pitting corrosion is frequently caused by halogenide ions.

Crevice corrosion
Corrosive attack primarily occurring in narrow crevices, caused by concentration differences in the corrosive medium, e.g., as a result of long oxygen diffusion paths. This type of corrosion generates potential differences between the crevice's extremities, leading to intensified corrosion in more poorly ventilated areas.

Stress corrosion cracking
Corrosion stemming from the simultaneous concerted action of a corrosive medium and mechanical tensile stress (which can also be present as internal stress in the object itself). Intercrystalline or transcrystalline fissures form, in many

PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

cases without the appearance of visible corrosion products.

Vibration corrosion cracking
Corrosion caused by the simultaneous effects of a corrosive medium and mechanical fatigue stress, e.g., caused by vibrations. Transcrystalline fissures are formed, frequently without visible deformation.

Fretting corrosion
Corrosion caused by the simultaneous effect of a corrosive medium and mechanical friction; sometimes called frictional oxidation.

Intercrystalline and transcrystalline corrosion
Types of corrosion characterized by selective formation along the grain boundaries or roughly parallel to the deformation plane in the grain's interior.

Dezincification
Selective dissolution of zinc from brass, leaving behind a porous copper structure. Denickelification and dealuminification are analogous processes.

Rust formation
Formation of ferriferous oxide and hydroxide corrosion products on iron and steel.

Corrosion testing

Electrochemical corrosion-testing procedures
In electrochemical testing procedures, in addition to determining the relationship between potentials and object materials during the corrosion reaction, above all the corrosion currents are measured. In the case of uniform surface corrosion, these are then used for the precise definition of the periodic weight and thickness losses. The relevant conversion factors are listed in Table 2.

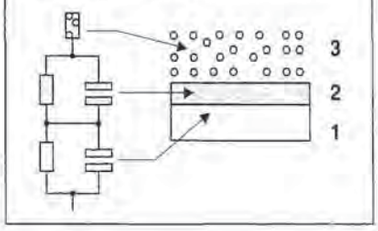
These electrochemical processes thus represent a valuable supplement to non-electrochemical methods.

The rate of free corrosion is defined based on the polarization resistance (slope of the total-current/voltage curve). Testing entails subjecting the metal to minimal, alternating anodic and cathodic pulses.

Electrochemical Impedance Spectroscopy (EIS) is employed to examine corrosion mechanisms. This alternating-current technique determines the AC resistance (impedance) and the phase angle of an electrochemical test object as a function of frequency. A low-amplitude sinusoidal alternating voltage is superimposed on the working electrode's potential, and the current response is measured. After measurement, the system is approximated in the form of an equivalent network. By way of example, the illustration captioned "Evaluation of EIS data" shows the equivalent network for the metal/coating/medium system.

The parameter-fit method is used to fit the equivalent networks to the experimental data. The impedance elements (resis-

Evaluation of EIS data
1 Metal, 2 Coating, 3 Aggressive medium.



PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

Table 1. Overview of selected standardized non-electrochemical corrosion-testing procedures

Standard	Type of corrosion-testing procedure
DIN[1]) EN ISO 196 (replaces DIN 50911)	Testing of copper and copper alloys; mercury nitrate test
DIN EN ISO 3651 (replaces DIN 50914)	Testing intercrystalline corrosion in stainless steels
DIN EN ISO 4628-3 (draft) DIN 53210	Evaluation of degradation of paint coatings, Part 3: evaluation of degree of rusting Designation of degree of rusting of paint surfaces and similar coatings (will be replaced by DIN EN ISO 4628-3)
DIN EN ISO 8565 (repl. DIN 50917-1)	Corrosion tests in atmosphere; natural weathering tests
DIN EN ISO 11306 (repl. DIN 50917-2)	Corrosion of metals; testing under natural conditions; testing in sea water
DIN 50016	Method of test in damp, alternating atmosphere
DIN 50017	Condensation water test atmospheres
DIN 50018	Testing in a saturated atmosphere in the presence of sulfur dioxide
DIN 50021	Salt spray testing
DIN 50900 -1...3	General terms, electrochemical terms, terms of corrosion testing
DIN 50905 -1...4	Corrosion testing; principles, corrosion characteristics under uniform corrosion attack, corrosion characteristics under non-uniform and localized corrosion attack without mechanical stress
DIN 50 915	Testing intercrystalline stress corrosion in non-alloy and low-alloy steels
DIN 50919	Testing contact corrosion in electrolyte solutions
DIN 50920 -1	Testing corrosion in flowing liquids
DIN 50922	Testing resistance of metallic materials against stress corrosion cracking
DIN 50928	Testing and evaluating protection of coated metallic materials against aqueous corrosive media
DIN 51213	Testing metallic wire coatings (tin, zinc)
N42AP 206	Climatic tests; testing in constant condensate climate
N42AP 209	Climatic tests; cyclic humidity tests
N42AP 213	Climatic tests; industrial-climate tests

1) Deutsches Institut für Normung e.V.

PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

tances, capacitances, inductances) are assigned physical properties. Direct conclusions can then be drawn about various characteristics, such as the effectiveness of corrosion-protection measures, porosity, thickness, a coating's ability to absorb water, the effectiveness of inhibitors, the rate of corrosion of the base metal, and so on.

The Scanning Reference Electrode Technique (SRET) is used for detecting corrosion in its early stages and for examining local corrosion processes.

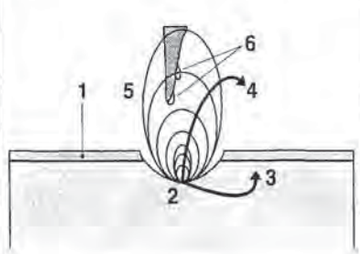
Pit, gap, and intercrystalline corrosion are examples of local processes that can significantly impair mechanical properties and even result in failure in extreme cases. A high local rate of corrosion and a change in local potential are typical of these processes.

The illustration captioned "SRET principle" is a diagrammatic view of the equipotential lines at a local, active corrosion point. A sensor consisting of two offset platinum tips measuring the micropotential changes above a rotating probe is used to pick up the signal.

Data is recorded automatically on a PC using the appropriate software for analysis and presentation in the form of 2D graphics.

SRET is used to examine pitting corrosion, activation and repassivation, to detect defects in organic coatings and at welds, and to trace delamination, etc.

SRET principle
1 Cathode, 2 Anode, 3 Electrons, 4 Metal ions, 5 Equipotential lines, 6 Platinum tips.



To measure contact corrosion, the current which flows between the two affected metals is measured directly with both immersed in the same corrosive medium.

The degradation rates determined through electrochemical measurements closely reflect those obtained in field testing. In addition to the small amount of corrosive medium required, another advantage of electrochemical procedures over non-electrochemical methods is that they provide quantitative data on corrosion rates.

Table 2. Loss of mass and thickness due to surface corrosion of various metals with a corrosion current density of 1 μA/cm²

Metals	Relative atomic mass	Density g/cm³	Loss of mass mg/(cm² · year)	Loss of thickness μm/year
Fe	55.8	7.87	9.13	11.6
Cu	63.5	8.93	10.40	11.6
Cd	112.4	8.64	18.40	21.0
Ni	58.7	8.90	9.59	10.8
Zn	65.4	7.14	10.70	15.0
Al	27.0	2.70	2.94	10.9
Sn	118.7	7.28	19.40	26.6
Pb	207.2	11.30	33.90	30.0

PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

Table 3. Degree of rusting and proportion of rust penetrations and visible sub-rusting at the surface to DIN EN ISO 4628-3

Degree of rusting	Rust surface in %
R_i0	0
R_i1	0.05
R_i2	0.5
R_i3	1
R_i4	8
R_i5	40 ... 50

Non-electrochemical corrosion-testing procedures

Non-electrochemical test procedures rely on determining the weight loss by weighing, or defining the rust level. DIN EN ISO 4628-3 (formerly DIN 53210) defines 5 different rust categories according to rust coverage or surface perforation (Table 3).

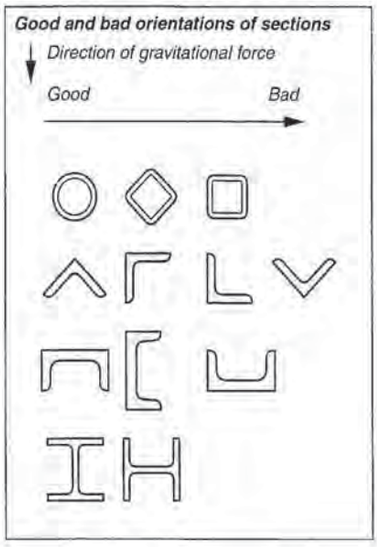
The corrosion tests have been defined to reflect actual field requirements. In addition to the standard DIN corrosion-testing procedures (Table 1), processes have evolved to reflect specific requirements, e.g. motor-vehicle testing. These tests provide reliable indices of projected service life under normal operating conditions by using short-term exposure in extremely harsh conditions to simulate long-term stresses in the real world. (These tests include operating exposure to climatic factors, splash-water tests of air/fuel mixture systems).

Corrosion protection

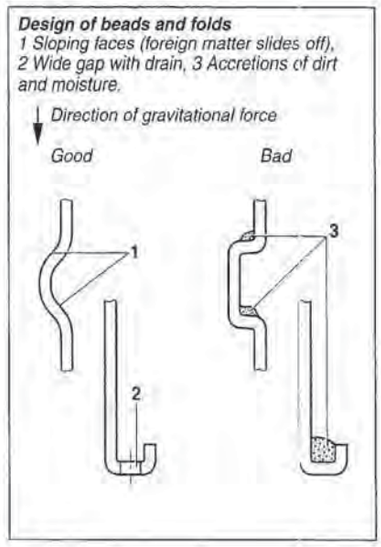
The manifestations and mechanisms of corrosion are many and varied, so widely differing methods can be adopted to protect metals against corrosion attack. Corrosion protection means intervening in the corrosive process with the object of reducing the rate of corrosion in order to prolong the service life of the components.

Corrosion protection can be achieved by applying four basic principles:

- Measures in planning and design: the choice of suitable materials and the suitable structural design of components,
- Measures that intervene in the corrosive process by electrochemical means,



Good and bad orientations of sections



Design of beads and folds
1 Sloping faces (foreign matter slides off), 2 Wide gap with drain, 3 Accretions of dirt and moisture.

PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

- Measures that separate the metal from the corrosive medium by protective layers or coatings, and
- Measures that influence the corrosive medium, for example the addition of inhibitors to the medium.

Corrosion protection by means of suitable structural design

Selecting suitable materials which feature optimum resistance to corrosion under the expected conditions can be of considerable assistance in avoiding corrosion damage. When the costs that would otherwise be incurred for upkeep and repair are factored into the long-term cost of ownership equation, a more expensive material can often be the more cost-effective alternative.

Design measures, too, are of major importance. A great deal of skill and expertise goes into design, particularly regarding the connections between parts that are made of the same material or different materials.

Corners and edges of sections are difficult to protect, and this is where corrosion can easily attack. Favorable orientation can reduce corrosion.

Beads and folds can trap dirt and moisture. Suitable surfaces and drain openings can help avoid this problem.

Welds, which generally modify the microstructure for the worse, are another weak point. In order to avoid crevice corrosion, welds have to be smooth and free of gaps.

Contact corrosion can be avoided by joining same or similar metals, or by installing washers, spacers or sleeves to ensure that both metals are electrically insulated.

Electrochemical processes

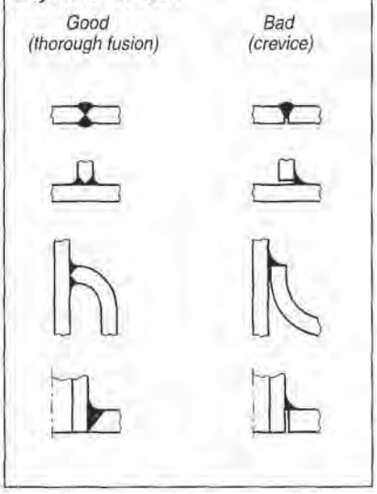
The schematic current/voltage curves for a metal suitable for passivation coating show how these processes work. The current-density values arranged in ascending order on the y-axis represent anodic currents corresponding to the corrosion reaction defined in the equation

$$Me \rightarrow Me^{n+} + ne^-$$

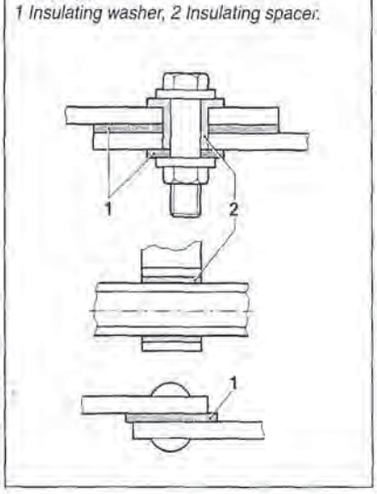
In contrast, the descending current-density values represent cathodic currents with the reaction equation progressing from right to left. The diagram incicates that external voltage can be applied to suppress corrosion. There are two basic ways of doing this:

For cathodic protection the potential is shifted so far toward the left that no an-

Design-related crevices in welds and how they can be avoided



Electric insulation to avoid contact corrosion
1 Insulating washer, 2 Insulating spacer



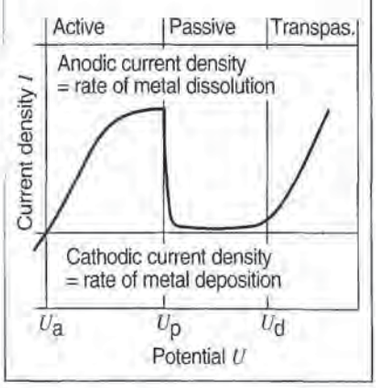
PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

odic currents flow, leading to $U < U_a$. As an alternative to applying voltage from an external source, it is also possible to shift the potential by using a base metal to act as a "sacrificial" reactive anode.

Another option is to shift the potential of the threatened electrode into the passive range, i.e., into the potential range between U_p and U_d. This is called anodic protection. The anodic currents which flow in the passive range are less than those in the active range by exponential powers of between 3 and 6, depending upon the type of metal and the corrosive medium. The result is excellent protection for the metal.

However, the potential should not exceed U_d, as oxygen would be produced in this transpassive range, potentially leading to higher rates of oxidation. Both of these effects would cause the current to increase.

Schematic current/voltage curve of a metal suitable for passivation
U_a Free corrosion potential of the metal in active state,
U_p Passivating potential,
U_d Breakthrough potential.



Coatings

Coatings inhibit corrosion by forming protective films applied directly to the metal to be protected, where they resist attack by the corrosive medium. These protective coatings should be neither porous nor electrically conductive. Because adequate application depths are also essential, considerable difficulties may be encountered when attempts are made to apply these substances in systems with close dimensional tolerances.

Inorganic nonmetallic coatings

Protective coatings can be formed interactively with the material to be protected. The formation of stable oxide coatings can be promoted in many metals by alloying in small amounts of foreign metals, such as silicon in the case of cast iron, or molybdenum with chrome-nickel steels. Phosphates, chromates and silicates can also produce coatings. Examples are found in the automotive industry, where these substances are employed in body panel manufacture ("phosphate coating") and in cooling systems.

Diffusion process

Surface treatment can be selectively combined with surface hardening by using the diffusion process to thermochemically carburize, carbonitride or chromate the metal, or treat it with boron or vanadium. The metal can also be oxidized, nitrited or sulfided without hardening.

Browning

Browning is used to produce coatings for limited protection over a short period of time. A ferrous material is browned by dipping it in hot concentrated sodium hydroxide containing $NaNO_2$.

Anodizing

Anodizing applications are restricted to aluminum. This anodic oxidation process in electrolytes containing sulfuric, chromic, or oxidic acids generally produces very durable coatings, which can be further stabilized and colored by chromating. Cadmium and magnesium surfaces can also be chromated.

PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

Enameling

Enameling furnishes high levels of corrosion protection, particularly on unalloyed steel. On the negative side, enamel coatings are brittle and thus particularly susceptible to impact and thermal shock. In addition, they are subject to consistent internal compressive stress, and tensile stress factors (e.g. when the yielding point of the carrier material is exceeded) cause fissures in the relatively inflexible enamel. To improve enamel bonding, an intermediate layer consisting of a base material is first fused to the metal; this intermediate layer is then covered with the actual silica-based protective coating.

Glass ceramics

Glass ceramics do not display the disadvantages of enamel under mechanical or thermal loading. They are applied to the material in the form of a semi-crystalline coating using controlled crystallization of thermodynamically unstable glasses. Subsequent heat treatment establishes roughly 50 % crystallinity.

Metallic coatings

Numerous options are available for applying foreign metals or metal compounds to base materials. In many cases, corrosion protection is combined with enhanced wear resistance and with the formation of decorative surfaces. Standard procedures, suitable metals and application ranges are listed in Table 4. It should be noted though that not every metal is suitable for every type of protection (e.g. aluminum applied to base materials in thermal spray processes provides no protection against wear).

Electroplating (electrolytic deposition)

In electrolytic plating, metals suspended in ionized metallic-salt solutions are deposited on conductive bases in a cathodic reaction. Under controlled conditions it is possible to produce alloys and dispersion coatings (layers of finely dispersed additives) of a type not amenable to duplication in fusion processes. Most metals are deposited from suspensions in aqueous solutions. In contrast, aluminum – an extremely base metal – can only be applied using water-free organic or molten saline solutions.

Table 4. Uses of inorganic protective coatings

Method of application	Type of metal	Primarily used for
Chemical deposition	Ni (NiB, NiP, NiPCu), Cu, dispersion coatings (NiB + SiC, NiP + SiC).	Corrosion protection, protection against wear, decorative surfaces.
Vapor deposition	Al, Au, Ta.	Corrosion protection
Electroplating (external power source)	Ag, Al, Au, Cd, Cr, Ni, Sn, Zn, ZnNi, ZnNiP.	Corrosion protection, decorative surfaces
Plating	Al, Cu, Ni, Pb, Ti, Zr and their alloys, stainless steels.	Corrosion protection
Hot immersion (including "hot-dip galvanization")	Al, Pb, Sn, Zn.	Corrosion protection
Thermal spraying	Al and alloys, Cr, Ni, NiCr, NiCrBSi, Ti, borides, carbides, oxides.	Corrosion protection, protection against wear.

PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

Chemical deposition
The basic principles are the same as in the electroplating process, but the electrons required for reduction do not come from an external power source. Instead, they are furnished by an added reducing agent (usually sodium hypophosphite or sodium boranate). This is why phosphorus and boron are present in most nickel coatings.

Hot immersion priming
The most common immersion process is the hot-dip galvanizing of iron and steel; this process is also the one most frequently used in the automotive industry. Here the pretreated (pickled and wetted with flux) material is immersed in the molten coating metal to produce alloy layers; this process progresses at the phase limit in accordance with solubility conditions in the phase diagrams. The alloy layers promote bonding between the coating and the base material. This process allows substantially greater layer depths than electroplating methods.

Evaporation
Due to its high cost, the evaporation process is limited to special applications, particularly in applying thin coatings to objects with limited surface areas (substrates). These thin layers often require additional protection from supplementary inorganic or organic coatings (example: headlight reflectors). The coating material (metal) is heated by an electric current and evaporated in an intense vacuum. It condenses on the substrate, forming a fine metal coating.

Thermal spraying
In thermal spraying, the coating material (usually in powdered form) is melted by a plasma ("plasma spraying") or an arc as it leaves the spray gun. Because this process entails accepting a certain amount of inconsistency and porosity in the final coating, thick layers (150 to 200 µm) are required to ensure effective corrosion protection.

Plating
Plating is the joining of two or more metal layers at high temperature and/or under high pressure. These metal layers do not separate under normal stress. Plating methods are divided into rolling, transfer and mechanical cladding. The materials are joined by using diffusion processes, bonding with alloying techniques, or mechanical interlocking. The application materials are several millimeters in thickness.

Organic coatings
Thermoplastics, elastomers and duromers are the primary types of organic protective coatings. They are used in the automobile industry in paints, whirl-sintered compounds, interior coatings, fiber-reinforced resins and fillers. How well these coatings protect the underlying material depends not only on the individual plastic, but also on the binder, anti-aging agent, UV stabilizers and the type of filler and pigment used. Organic coatings can be used alone or together with one of the inorganic coatings mentioned above.

Painting
Depending upon individual requirements, paint can be applied with brushes and rollers, using compressed-air or high-pressure sprays, and by immersion processes (with and without electrophoresis), and with electrostatic spray painting. An adhesion-promoting primer coat is applied to the pretreated metal surface with subsequent application of the actual protective coat on top of the primer. Solvent evaporation and/or oxygen absorption then promote formation of a homogenous, continuous surface layer.

Powder coating
Powdered plastics, which are very difficult to transform into a liquid phase at room temperature, can be applied using powder coating processes. The powder can either be fused onto the preheated parts, flame-sprayed or electrostatically sprayed. Heat treatment can be employed to form homogenous sealed layers as required. Powder coating processes are usually more economical than painting processes, while environmental and occupational safety are both less of a problem.

PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

Internal rubber and plastic coatings
Rubber and plastic coatings are frequently used to protect the insides of transport and storage tanks, vessels and pipes. Prelinked natural and synthetic rubber, and pretreated thermoplastic films are applied with interlacing adhesives; final linkage takes place either catalytically or by applying heat after the coatings have been applied to the bare carrier materials.

Plasma polymerization
Plasma polymerization is a process in which a gaseous-discharge plasma promotes polymerization of organic monomers on the object material. This type of polymer film represents a suitable coating for any metal. The result is an organic layer without gas enclosures or the volatile components frequently found in solid polymers.

Inhibitors

Inhibitors are substances added to the corrosive medium in low concentrations (up to a maximum of several hundred ppm) for absorption at the surface of the protected metal. Inhibitors drastically reduce the rate of corrosion by blocking either the anodic or the cathodic subprocess (frequently blocking both subprocesses simultaneously). Organic amines and amides of organic acids are the most frequent inhibitors. In automotive applications, for example, inhibitors are used in fuel additives; they are also added to antifreeze in order to inhibit corrosion damage in the coolant circuit.
Vapor-phase inhibitors provide only temporary protection for metallic products during storage and shipping. They must be easy to apply and remove.

The drawback of vapor-phase inhibitors is that they are potentially injurious to health.

Vapor-phase inhibitors (VPI), or volatile corrosion inhibitors (VCI), are organic substances of moderate vapor pressure. They are frequently enclosed in special-purpose packing materials or as solvents in liquids or emulsions with oil. The inhibitors evaporate or sublimate over the course of time, and are adsorbed as monomolecules on the metal, where they inhibit either anodic or cathodic corrosion subreactions, or both at once. Dicyclohexylamin nitrite is a typical example.

For optimal effectiveness, the inhibitor should form a sealed coating extending over the largest possible surface area. This is why they are generally enclosed in packing materials such as special paper or polyethylene foil. An airtight edge seal is not required; the packing can be opened briefly for inspection of contents. The duration of the packing's effectiveness depends on the tightness of the seal and the temperature (normally approx. 2 years, but less in environments substantially hotter than room temperature).

Standard commercial vapor-phase inhibitors are generally a combination of numerous components capable of providing simultaneous protection for several metals or alloys. Exceptions: cadmium, lead, tungsten and magnesium.

PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

Heat treatment of metallic materials

Heat treatment is employed to endow metallic tools and components with the specific qualities required either for subsequent manufacturing processes or for actual operation. The heat-treating process includes one or several time and temperature cycles. First, the parts to be treated are heated to the required temperature, where they are maintained for a specific period before being cooled back down to room temperature (or below in some processes) at a rate calculated to achieve the desired results.

The process modifies the microstructure to achieve the hardness, strength, ductility, wear resistance, etc., required to withstand the stresses associated with static and dynamic loads. The most significant industrial processes are summarized in Table 1 (see DIN EN 10 052 for terminology).

Hardening

Hardening procedures produce a martensitic microstructure of extreme hardness and strength in ferrous materials such as steel and cast iron.

The parts being treated are heated to the austenitizing, or hardening temperature, at which they are maintained until an austenitic structure emerges, and until an adequate quantity of carbon (released in the decay of carbides, such as graphite in cast iron) is dissolved in the treated material. The material is then quenched or otherwise cooled back to room temperature as quickly as possible in order to obtain a maximum degree of conversion to a martensitic microstructure (the time-temperature transformation chart for the specific steel in question contains the reference figures for the necessary cooling rate).

The austenitizing temperature varies according to the composition of the material in question (for specific data, consult the DIN Technical Requirements for Steels). Table 2 above furnishes reference data. See DIN 17 022, Parts 1 and 2, for practical information on hardening procedures for tools and components.

Table 2. Standard austenitizing temperatures

Type of steel	Quality specification	Austenitizing temperature °C
Unalloyed and low-alloy steels < 0.8% by mass of C	DIN EN 10 083-1 DIN 17 211 DIN 17 212	780...950
≥ 0.8% by mass of C	–	780...820
Cold- and hot-working tool steels	DIN 17 350	950...1100
High-speed steels		1150...1230
Cast iron	–	850...900

Table 1. Summary of heat-treatment processes

Hardening	Austempering	Tempering	Thermochemical treatment	Annealing	Precipitation hardening
Through hardening Surface hardening Hardening of carburized parts (case-hardening)	Isothermic transformation in the bainite stage	Tempering of hardened parts Hardening and tempering above 550 °C	Carburizing Carbonitriding Nitriding Nitrocarburizing Boron treatment Chromating	Stress-relief Recrystallization annealing Soft annealing, spheroidization Normalizing Homogenizing	Solution treatment and aging

PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

Not all types of steel and cast iron are suitable for hardening. The following equation describes the hardening potential for alloyed and unalloyed steels with mass carbon contents of between 0.15 and 0.60 %, and can be applied to estimate the hardness levels achievable with a completely martensitic microstructure:

Max. hardness = 35 + 50 · (%C) ± 2 HRC

If the microstructure does not consist entirely of martensite, then the maximum hardness will not be reached.

When the carbon content exceeds 0.6 % by mass, it may be assumed that the material's structure contains untransformed austenite (residual austenite) in addition to the martensite. This condition prevents the maximum hardness from being achieved, and the wear resistance will be lower. In addition, residual austenite is metastable, i.e. there exists a potential for subsequent transformation to martensite at temperatures below room temperature or under stress, with changes in specific volume and internal stress as the possible results. Low-temperature follow-up procedures or tempering operations at over 230 °C can be useful in cases where residual austenite is an unavoidable product of the hardening procedure.

The surface and core hardnesses remain virtually identical in components with material thicknesses up to approx. 10 mm. Beyond this point the core hardness is lower; there is a hardness progression or gradient. The rate of the progression depends upon the hardening response (testing described in DIN 50 191), which is a function of the material's composition (Mo, Mn, Cr). This factor requires particular attention with parts which do not cool well (thick parts and/or slow or graduated cooling processes designed to minimize the risk of cracks and/or distortion).

DIN 50 150 defines the method for using hardness as the basis for estimating tensile strength R_m; this method can only be applied in cases where the surface and core hardnesses are virtually identical:

$R_m \approx (34...37.7) \cdot$ Rockwell C hardness number in N/mm²

or

$R_m \approx (3.2...3.35) \cdot$ Vickers hardness number in N/mm².

The specific volume of the martensitic microstructure is approximately 1.0 % greater than that of the original material. In addition, stresses result from the rearrangement of the microstructure and from contraction during cooling. As the latter phenomenon does not take place at a uniform rate in all sections of the part, it produces variations in shape and dimensions; tensile stresses near the surface and pressure tension at the core are the common result.

Surface hardening

The process is especially suited for integration within large-scale manufacturing operations, and can be adapted to fit the rhythm of the production line.

Heating and hardening are restricted to the surface, thereby minimizing alterations in shape and dimensions. Heating is generally provided by high- or medium-frequency alternating current (inductive hardening) or by a gas burner (flame hardening). Friction (friction hardening) and high-energy beams (e.g. electron or laser beams) can also provide the heat required for austenitizing. Table 3 provides a summary of the specific heat energies for the individual procedures.

These methods can be used to treat both linear and flat surfaces, meaning that the parts can be heated either while stationary or in motion. The heat source itself can also be moved. Rotation is the best way of dealing with radially symmetrical parts, as it ensures concentric hardening. Either immersion or spraying arrange-

Table 3. Comparison of power densities when heating with different sources

Energy source	Normal power density W/cm²
Laser beam	$10^3...10^4$
Electron beam	$10^3...10^4$
Induction (MF, HF, HF pulse)	$10^3...10^4$
Flame heating	$10^3...6 \cdot 10^3$
Plasma beam	10^4
Molten salt (convection)	20
Air/gas (convection)	0.5

PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

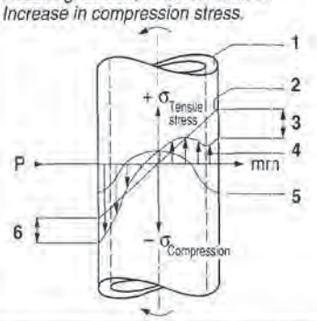
ments can be applied for quenching.

Heat rise is rapid, so the temperatures must be 50...100 °C higher than those used in furnace heating so as to compensate for the shorter dwell period. The procedure is generally employed with low-alloy or unalloyed steels with mass carbon contents of 0.35...0.60 % (consult DIN 17 212 for list of suitable steels). However, surface hardening processes can also be applied with alloyed steels, cast iron and rolling-bearing steels. The parts can be heat-treated to provide a combination of improved base strength and high surface hardness, making them suitable for high-stress applications (recessed edges, bearing surfaces, cross-sectional transitions).

Surface hardening generally results in internal compression stresses along the edge. This leads to increased fatigue resistance, especially when notched parts are exposed to inconstant vibration stress (see illustration).

The relationship defined above can be employed to estimate the potential surface hardness. There is a substantial reduction in hardness between the surface and the unhardened core region. The hardening depth R_{ht} – the depth at which 80 % of the Vickers surface hardness is found – can be derived from the hardness progression curve (see DIN 50 190, Part 2).

Cyclically alternating stress according to surface-layer hardening
+ σ Tensile stress, – σ Compression;
1 Case layer, 2 Bending stress,
3 Reduction of tensile stress,
4 Resulting tension, 5 Intrinsic stress
6 Increase in compression stress.



Austempering

The object of this process is to achieve a bainite microstructure. This microstructure is not as hard as martensite, but does display greater ductility as well as smaller changes in specific volume.

After austenitizing (see hardening), the parts for austempering are first cooled to a temperature of 200...350 °C (depending upon the exact composition of the material) at the required rate. The parts are then held at this temperature until the microstructure's transformation into bainite has been completed. The parts can then be cooled to room temperature (no special procedure required).

Austempering is an excellent alternative for parts whose geometrical configuration makes them sensitive to distortion and/or cracks, or in which high ductility is required together with substantial hardness, or which should combine hardness with a low level of residual austenite.

Draw tempering

Parts emerge from the hardening process in a brittle state; they must be tempered to increase their ductility in order to reduce the risk of damage associated with excessive internal tension, such as delayed cracking after hardening or splintering during grinding. This tempering process is based on the elimination of carbides, a phenomenon accompanied by an increase in ductility, albeit at the price of reductions in hardness.

The parts are heated in the draw to a temperature of between 180 and 650 °C; they are then held there for at least one hour before being allowed to cool to room temperature. Depending upon the specific composition of the material, tempering at temperatures in excess of 230 °C may result in any residual austenite being transformed into bainite and/or martensite.

Tempering at temperatures as low as 180 °C is enough to reduce the hardness of unalloyed and low-alloy steels by approx. 1...5 HRC. The individual materials respond to higher temperatures with specific characteristic hardness loss. The graph at the right shows a characteristic

PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

tempering curve for typical types of steel. The graph illustrates the fact that the hardness of high-alloy steels remains constant until the temperature exceeds 550 °C, after which it drops.

The mutual relationships between tempering temperature on the one side, and hardness, strength, yield point, fracture contraction and ductility on the other, can be taken from the tempering diagrams for the various steels (see DIN EN 10083).

Tempering of hardened parts is accompanied by a reduction in specific volume. In some cases, tempering can also induce changes in the progressive variation in internal tension at different depths in the parts.

It must be remembered that steels alloyed with manganese, chromium, manganese and chromium, chrome-vanadium, and chromium and nickel should not be tempered at temperatures of 350...500 °C, as brittleness could result. When these types of materials are cooled from higher tempering temperatures, the transition through this critical range should also be effected as rapidly as possible (see DIN 17022, Parts 1 and 2 for additional information).

Quench and draw

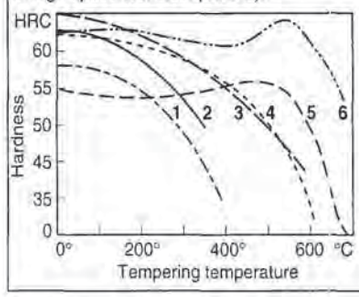
This type of quench and draw process combines hardening and tempering at temperatures above 500 °C. This procedure is designed to achieve an optimal relationship between strength and ductility. It is applied in cases where extreme ductility or malleability is required.

Particular care must be devoted to avoiding brittleness in the quench and draw operation (see above).

Thermochemical treatment

In thermochemical treatment, the parts are annealed in agents which emit specific elements. These diffuse into the surface layer of the parts being treated and modify their composition. This results in very specific properties. Of particular importance for this process are the elements carbon, nitrogen and boron.

Tempering response of various types of steel
1 Non-alloyed tempering steel (C45),
2 Non-alloyed cold-working tool steel (C80W2),
3 Low-alloyed cold-working tool steel (105WCr6),
4 Alloyed cold-working tool steel (X165CrV12),
5 Hot-working tool steel (X40CrMoV51),
6 High-speed tool steel (S6-5-2).



Carburizing, carbonitriding, case hardening

Carburizing increases the carbon content in the surface layer, while carbonitriding supplements the carbon enrichment with nitrogen. This process is generally carried out in molten salt, granulate, or gas atmospheres at temperatures ranging from 850...1000 °C. The actual hardening is performed subsequently, either by quenching directly from the carburizing/carbonitriding temperature (direct hardening), or by allowing the parts to cool to room temperature (single hardening), or by allowing them to cool to a suitable intermediate temperature (e.g. 620 °C) prior to reheating (hardening after isothermic conversion). This process produces a martensitic surface layer, while the degree of martensite at the core is a function of hardening temperature, hardenability and part thickness.

Specific temperatures can be selected for either surface hardening in the upper layers with higher carbon content (case refining), or for the non-carburized core (core refining) (see DIN 17022, Part 3).

Carburizing and carbonitriding produce a characteristic carbon declivity, with levels dropping as the distance from the surface increases (carbon curve). The distance between the surface and the point at which the mass carbon content is still

PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

0.35 % is normally defined as the carburization depth $At_{0.35}$.

The length of the carburizing or carbonitriding process depends upon the required carburization depth, the temperature and the atmosphere's carbon-diffusion properties. A reasonable approximation is possible:

$$At = K \cdot \sqrt{t} - D/\beta \text{ in mm}$$

Depending on temperature and carbon levels, K lies between 0.3 and 0.6 during carburization in a gas atmosphere, for example; the correction factor D/β is generally 0.1...0.3 mm; the time t in h must be inserted.

Generally, the objective is to achieve a carbon gradient with a concentration of at least 0.60 % mass carbon content, the ultimate goal being a surface hardness of 750 HV (corresponding to 65 HRC). Higher concentrations of carbon can lead to residual austenite and/or carbide diffusion, which could have negative effects on the performance of case-hardened parts in actual use. Control of the atmosphere's carbon level, and thus the part's ultimate carbon content, is thus extremely important.

The gradient defining the relationship between hardness and depth corresponds to the carbon concentration curve. The hardness gradient is used to define the case depth *Eht*. DIN 50190, Part 1 defines this as the maximum distance from the surface before the hardness drops below 550 HV.

The case-hardened part generally exhibits compression tension at the surface, and tensile stresses at the core. As with surface-hardened materials, this distribution pattern provides enhanced resistance to vibration loads.

In carbonitriding, nitrogen is also absorbed; it serves to improve the material's tempering properties, increase its durability and enhance its wear resistance. The positive effects are especially pronounced with non-alloyed steels. For additional, more detailed information on case-hardening procedures consult DIN 17022, Part 3, and Information Sheet 452 of the Steel Information Center, Düsseldorf.

Nitriding and nitrocarburizing

Nitriding is a thermal treatment process (temperature range: 500...600 °C) which can be used to enrich the surface layer of virtually any ferrous material with nitrogen. In nitrocarburizing, a certain amount of carbon is diffused into the material at the same time.

The nitrogen enrichment is accompanied by precipitation hardening. This strengthens the surface layer, enhancing the material's resistance to wear, corrosion and alternating cyclic stress.

Because the process employs relatively low temperatures, there are no volumetric changes of the kind associated with transformations in the microstructure, so that changes in dimensions and shape are minute.

The nitrided region consists of an outer layer, several millimeters in depth, and a transitional white layer, the hardness of which may be anywhere from 700 to over 1200 HV, depending upon the composition of the material. Still deeper is a softer diffusion layer extending several tenths of a millimeter. The thickness of the individual layers is determined by the temperature and duration of the treatment process. The process produces a hardness gradient (similar to that which results from surface and case-hardening); this gradient furnishes the basis for determining the nitriding depth *Nht*. DIN 50190, Part 3 defines this as the depth from the surface at which the hardness is still 50 HV above the core hardness.

The material's resistance to wear and corrosion is essentially determined by the white layer, which contains up to 10 mass components of nitrogen in %. The nitriding depth and the surface hardness determine the material's resistance to alternating cyclic stress (for additional details, see DIN 17022, Part 4, and Information Sheet 447 from the Steel Information Center, Düsseldorf).

Boron treatment

This is a thermochemical treatment method which employs boron to enrich the surface layer of ferrous materials. Depending upon duration and temperature (normally 850...1000 °C), an iron-boron white layer of 30 µm...0.2 mm in

PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

depth and with a hardness of 2000...2500 HV is produced.

Boron treatment is particularly effective as a means of protecting against abrasive wear. However, the comparatively high process temperature leads to relatively large changes in shape and dimensions, meaning that this treatment is only suitable for applications in which large tolerances can be accepted.

Annealing

Annealing can be applied to optimize certain operational or processing characteristics of parts. With this method, the parts are heated to the required temperature and maintained there for an adequate period before being cooled to room temperature. Table 1 lists the various processes used in specific individual applications.

Stress-relief
Depending upon the precise composition of the parts, this operation is carried out at temperatures ranging from 450...650 °C. The object is to achieve the maximum possible reduction in internal stress in components, tools and castings by inducing physical deformation.

After an oven time of 0.5...1 h, the parts are cooled back to room temperature; this cooling should be as gradual as possible to prevent new stresses from forming.

Recrystallization annealing
Recrystallization annealing is applied with parts that have been formed using non-cutting procedures. The goal is to restructure the grain pattern in order to prevent increased hardening, thereby facilitating subsequent machining work.

The temperature requirement depends upon the composition of the material and the degree of deformation: it lies between 550...730 °C for steel.

Soft annealing, spheroidization
Soft annealing is intended to facilitate the machining and/or shaping of workpieces which have hardened owing to deformation processes or other heat treating procedures.

The temperature requirement is determined by the material's composition. It is in a range of 650...720 °C for steel, and lower for nonferrous materials.

Spheroidizing of cementite is applied when a microstructure with a granular carbide pattern is desired. If the initial structure is martensite or bainite, the result will be an especially homogeneous carbide distribution.

Normalizing
Normalizing is carried out by heating the parts to austenitizing temperature and then allowing them to gradually cool to room temperature. In low-alloy and non-alloyed steels, the result is a structure consisting of ferrite and perlite. This process is essentially employed to reduce grain size, reduce the formation of coarse grain patterns in parts with limited reshaping, and to provide maximum homogeneity in the distribution of ferrite and perlite.

Precipitation hardening treatment
This process combines solution treatment with aging at ambient temperature. The parts are heated and then maintained at temperature to bring precipitated structural constituents into a solid solution, and quenched at room temperature to form a supersaturated solution. The aging process comprises one or several cycles in which the material is heated and held at above-ambient temperatures (hot aging). In this process, one or several phases, i.e. metallic bonds between certain base alloys, are formed and precipitated in the matrix.

The precipitated particles enhance the hardness and strength of the base microstructure. The actual characteristics are determined by the temperature and duration of the aging process (option of mutual substitution); exceeding a certain maximum will usually reduce the strength and hardness of the final product.

Precipitation hardening is mostly applied for nonferrous alloys, but some hardenable steels (maraging steels) can also be processed.

PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

Hardness

Hardness testing

Hardness is the property of solid materials that defines their resistance to deformation. In metallic materials the hardness is used to assess mechanical properties such as strength, machinability, malleability, and resistance to wear. DIN 50 150 defines guidelines for converting hardness to tensile strength. Measurement processes are practically nondestructive.

The test data are generally derived from the size or depth of the deformation produced when a specified indentor is applied at a defined pressure.

A distinction is made between static and dynamic testing. Static testing is based on measurement of the permanent impression left by the indentor. Conventional hardness tests include the Rockwell, Vickers and Brinell procedures. Dynamic testing monitors the rebound of a test tool accelerated against the surface of the test specimen.

Another option for obtaining an index of surface hardness is to scratch the surface with a hard test tool and then measure the groove width. The table (next page) compares the standard application ranges for the Rockwell, Vickers and Brinell hardness tests.

Rockwell hardness

(DIN EN 10 109)

This method is particularly suitable for large-scale testing of metallic workpieces. A steel or diamond indentor is positioned vertically against the surface of the test specimen and a minor load applied prior to application of the full test pressure (major load) for a period of at least 30 s. The Rockwell hardness is the resulting penetration depth e in mm (see Table 1).

The test surface should be smooth (depending on hardness range, $R_{max} \leq$ 1.2...3.4 µm) and as flat as possible. If the specimen's radius of curvature is less than 20 mm then the results must be scaled using the correction factor for the individual resistance level.

Selection of the individual test procedures enumerated in Table 1 is based upon either the specimen's thickness or the case depth of the hardened surface layer (refer to Figs. A.1...A.3 in DIN EN 10 109). The abbreviation for the selected test method should be appended to the numerical data when specifying hardness, e.g., 65 HRC, 76 HR 45 N. The Rockwell C procedure's error range is approx. ±1 HRC.

Advantages of the Rockwell test method include minimal specimen preparation and rapid measurement; this test process can also be completely automated.

Table 1. Rockwell test methods

Abbreviation	Indentor	Minor load N	Total test force N	Hardness number (e indentor penetration depth)	
HRC	Diamond cone	98 ± 2	1471 ± 9	100 − e / 0.002	20...70
HRA			588 ± 5		60 ... 381
HRB	Steel ball		980 ± 6.5	130 − e / 0.002	35 ... 100
HRF			588 ± 5		60 ... 100
HR15N HR30N HR45N	Diamond cone	29.4 ± 0.6	147 ± 1 294 ± 2 441 ± 3	100 − e / 0.001	66 ... 92 39...84 17...75
HR15T HR30T HR45T	Steel ball	3 ± 0.06	15 ± 0.1 30 ± 0.2 45 ± 0.3		50 ... 94 10 ... 94 0 ... 75

PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

Any tester vibration, or shifting and movement of either the test probe or the specimen itself can lead to testing errors, as can an uneven support surface or a damaged indentor.

Brinell hardness (DIN EN 10003)

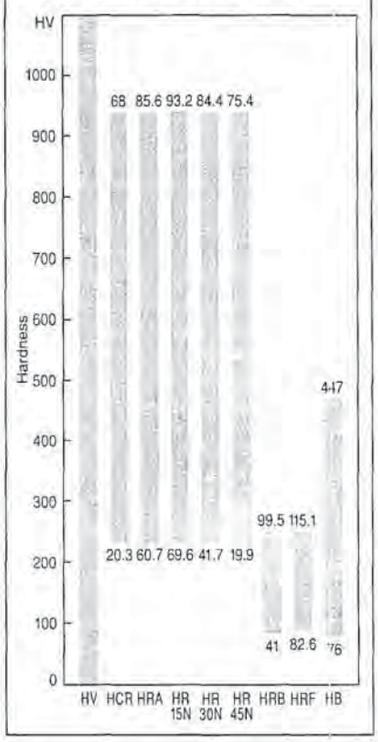
This procedure is used for metallic materials of low to medium hardness. The test tool (indentor) is a hard-metal or hardened-steel ball. It is applied to the surface of the specimen at an individually specified force F for at least 15 s[1]). Microscopic examination of the resulting deformation diameter d provides the basis for calculating the Brinell hardness. These data can be correlated with standard charts or calculated as follows:

$$\text{Brinell hardness} = \frac{0.204 \cdot F}{\pi \cdot D \cdot (D - \sqrt{D^2 - d^2})} \text{ HB}$$

F Force in N
D Ball diameter in mm
d Indentation diameter in mm

Test pressures range from 9.81 to 29,420 N. Results obtained using spheres of different diameters are only conditionally comparable, and any comparisons should be based on testing at identical force levels. Testing should always be performed using the largest possible spherical sphere, while load factors should be

Comparison of hardness ranges for different test methods
The figures at the range extremities indicate the hardness data for the respective methods. HB test procedures with force levels 30 (1); 10 (2); 5 (3); and 1 (4)



[1]) At least 30 s for lead, zinc, etc.

Table 2. Application of the Brinell hardness test

Material	Brinell hardness	Load factor 0.102 F/D^2
Steel; nickel and titanium alloys		30
Cast iron[1])	<140	10
	≥140	30
Copper and copper alloys	<35	5
	35 to 200	10
	<200	30
Light metals and their alloys	<35	2.5
	35 to 80	5 10 15
	>80	10 15
Lead and tin		1
Sintered metals	see EN 24498-1	

[1]) Ball nominal diameters of 2.5, 5 or 10 mm are specified for testing cast iron.

PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

selected to obtain an indentation diameter of (0.24 ... 0.6) · D. Table 2 lists the recommended load factors and ball diameters for a variety of materials as laid down in DIN EN 10003.

In the Brinell hardness designation the numeric data are accompanied by the procedure code, the material of the ball (W for hard metal, S for steel) and the ball diameter. The final element is the test force (in N) multiplied by a factor of 0.102. Example: 250 HBW 2.5/187.5.

To prevent test errors stemming from deformation of the ball itself, testing at hardness factors in excess of 450 HB is to be carried out exclusively with a hard-metal ball.

High test pressures producing deformations extending over a relatively wide surface area can be employed to gather data on materials with inconsistent structures. An advantage of the Brinell method is the relatively high degree of correlation between the Brinell hardness factor and the steel's tensile strength.

The range of application for the Brinell test method is limited by specifications for the thickness of the test specimen (cf. DIN EN 10003) and by the ball material. The required preparations and ensuing test procedures are more complex than those used for Rockwell testing. Potential error sources are the test's reliance on visual evaluation of the impact area's diagonal length and the possibility of inconsistencies in the impression itself.

Vickers hardness (DIN 50 133)

This test method can be used for all metallic materials, regardless of hardness. It is especially suitable for testing minute and thin specimens, while the potential application range extends to include surface and case-hardened parts as well as nitrided workpieces and parts carburized in nitrogen-based atmospheres.

The test tool is a square-based diamond pyramid with an apical angle of 136°. This is applied to the surface of the test specimen at the specified force F.

The diagonal d of the rhomboidal cavity remaining after the tool's removal is measured using a magnifying glass to obtain the basis for calculating the Vickers hardness, which can be read from tables or calculated as follows:

Vickers hardness = $0.189 \cdot F/d^2$ HV

F Applied force in N
d Diagonal of impression in mm

Table 3 gives an overview of the graduated force levels defined in DIN 50 133.

In the formula for Vickers hardness data, the actual test figure is accompanied by the abbreviation HV, the force in N (multiplied by a factor of 0.102) and, following a slash, the force application period (if other than the standard 15 s) in seconds, e.g.: 750 HV 10/25.

The test specimen's surface should be smooth ($R_{max} \leq 0.005 \cdot d$) and flat. DIN 50 133 stipulates that correction factors be used to compensate for any error stemming from surface curvature. Test pressure levels are selected with reference to either the thickness of the test specimen itself or of its hardened outer layer (cf. Fig. 2 in DIN 50 133). Practical experience indicates an error range of approximately ± 25 HV. Although the geometrical configurations are similar, hardness data vary as a function of applied-force levels.

A major advantage of this test method is that there are virtually no limitations on using it to assess thin parts or layers. It allows the use of extremely small force levels to determine the hardness of individual structural sections. The Brinell and Vickers numbers correlate up to approximately 350 HV.

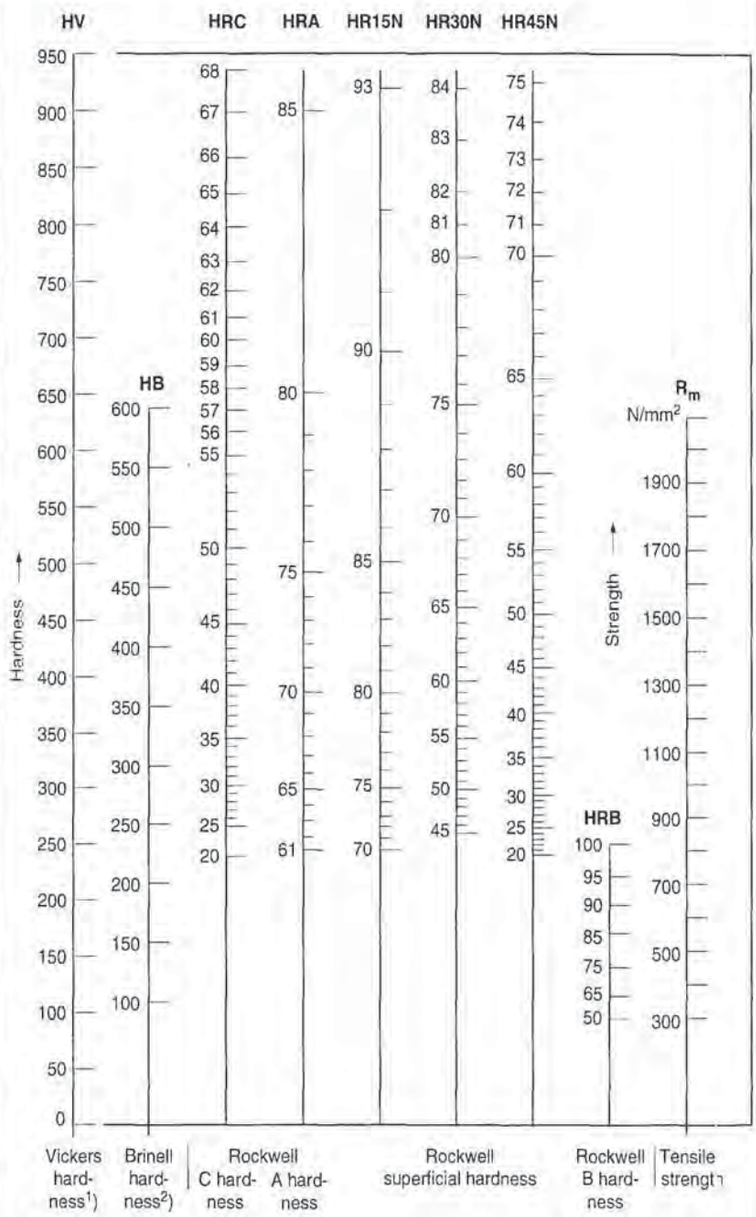
However, a certain minimal degree of surface consistency is necessary to ensure accurate results. Vickers testing furnishes roughly the same accuracy levels as the Brinell method, although the mea-

Table 3. Vickers hardness force increment

Abbreviation	HV0.2	HV0.3	HV0.5	HV1	HV2	HV3	HV5	HV10	HV20	HV30	HV50	HV100
Applied force N	1.96	2.94	4.9	9.8	19.6	29.4	49	98	196	294	490	980

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US PATENT NO. 6,581,565

Correlation between hardness and strength in accordance with DIN 50 150



¹) Applied force ≥ 98 N. ²) Force level 30.

PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

surement process is extremely sensitive to movement of the test set-up. The test tools are more expensive than the balls used in Brinell testing.

Knoop hardness

This process closely resembles the Vickers procedure. It is the method of choice for testing thin-layer hardness in the Anglo-Saxon world.

The test tool is designed to leave an impression in the form of a thin, elongated rhombus. The extended diagonal d is seven times longer than the short one and is the only one that is actually measured. The applied force is less than 9.8 N. Hardness ratings are available from standard charts, or may be calculated as follows:

$$\text{Knoop hardness} = 1.451 \cdot F/d^2 \text{ HK}$$

F Applied load in N
d Long diagonal in mm

The diagonal, which is roughly 2.8 times longer than in the Vickers method, provides more accurate results with visual assessment procedures. The penetration depth is 1/3 less than in the Vickers method, allowing evaluation of surface hardness in thin parts and layers. By making Knoop impressions in different directions, even material anisotropies can be detected.

Disadvantages of this method include pronounced sensitivity to misaligned specimen surfaces, which must be absolutely perpendicular to the force-application axis, and the fact that the maximum force is restricted to 9.8 N. There are no special German standards for this procedure. Test results cannot be correlated with those of the Vickers test, on account of this lack of standardization.

Shore hardness

This method is primarily used for hardness testing on rubber and soft plastics. The test tool is a steel pin of 1.25 mm diameter; this is forced against the surface of the test specimen by a spring. The subsequent change in the spring length (spring travel) furnishes the basis for determining Shore hardness.

In the Shore D method a pretensioned spring exerts a force of 0.55 N. The tip of the steel pin has the shape of a frustrum (for testing hard rubber). The spring rate c is 4 N/mm.

In the Shore A method, measurement uses a conical steel pin with a rounded tip and no pretension. The spring rate c is 17.8 N/mm. Each 0.025 mm of spring travel corresponds to 100 Shore units.

Ball impression hardness

(DIN 53 456)

This is the standard test for determining hardness levels in plastomers, and is also employed with hard rubber substances. The test tool is a hardened steel ball 5 mm in diameter. It is applied with a minor load of 9.81 N to the surface of the test specimen which must be at least 4 mm thick. Subsequent graduated rises in force provide application pressures of 49, 132, 358 and 961 N. After 30 s the penetration depth is measured and indicated on an analog display. The test load F should be selected to provide a penetration depth h of between 0.15 and 0.35 mm.

Ball impression hardness is defined as the ratio of the applied load to the deformed area in the test surface. It can be read from tables or calculated as follows:

$$\text{Ball impression hardness} = \frac{0.21 \cdot F}{1.25 \cdot \pi \cdot (h - 0.04)} \text{ N/mm}^2$$

F Applied load in N
h Penetration depth in mm

Scleroscope hardness

This dynamic measurement method is specially designed for heavy and large metal pieces. This process is based on the measurement of the rebound height (energy of impact) of a steel indentor (hammer) featuring a diamond or hard metal tip; this is dropped from a stipulated height onto the surface of the test specimen. The rebound serves as the basis for determining the hardness.

The method is not standardized and there is no direct correlative relationship with any other hardness testing method.

PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

Tolerances

Correlations

In the absence of special supplementary definitions, envelope analysis is mandatory for all individual form elements in prints based on standardized DIN definitions for tolerances and fits (e.g. Independence Principle from ISO 8015). The envelope concept is based on the Taylor gauge test, which specifies that maximum intolerance material mass shall not penetrate the geometric envelope at any point.

Because it is not possible to produce geometrically ideal workpieces, dimension limits (dimensional tolerances) are defined for production.

DIN 7167 includes all tolerances of form, including tolerances of parallelism, position and planar regularity as well as dimensional tolerances. The geometrical tolerance may lie anywhere within the dimensional tolerance (see diagram).

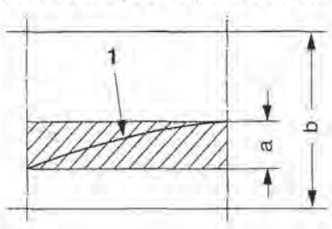
For the following tolerances of position the envelope dimension at maximum material dimension is not defined:

Tolerances of perpendicularity, slope, symmetry, coaxiality and concentricity. For these tolerances of position either direct specifications on the drawing or the general tolerances are necessary.

No generally applicable correlative data are defined for the relationship between dimensional tolerance and peak-to-valley height (general formula: $R_z \leq 0.5 \cdot T$).

ISO International Organization for Standardization.

Tolerance position
1 Actual form.
a Form tolerance, b Dimensional tolerance.



ISO system for limit allowances and fits

ISO tolerance classes are indicated by letters (for basic allowances) and numbers (for basic tolerance grades). The definitions are:

- The **letters** A to Z indicate the position of the tolerance range relative to the base line; small letters for shafts; capital letters for bores.
- The **numbers** 01 to 18 indicate the magnitude of the tolerance grade.

The ISO tolerance ranges for shafts and bores can be combined as desired to give fits; the standard bore and standard shaft systems of fits are preferred.

Standard bore: The basic allowances for all holes are identical, and different fits are obtained by selecting the appropriate shaft sizes.

Standard shaft: The basic allowances for all shafts are identical, and different fits are obtained by selecting the corresponding bore dimensions.

Most common clearance, transition and interference fits can be defined using the following selection of ISO tolerances from ISO 286.

Example for tolerance class: H7
Basic allowance
Tolerance grade

Tolerances of form and position

Tolerances of form and position should be specified only where required (e.g. in response to functional requirements, interchangeability and possible production conditions).

An element's geometrical and positional tolerances define the zone within which it must be located (surface, axis or generating line).

Within the tolerance zone the toleranced element may have any desired form or directional orientation. The tolerance applies to the entire length or surface of the toleranced element.

Symbols are employed when entering the tolerances in prints (see table).

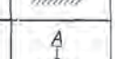
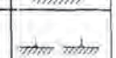
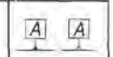
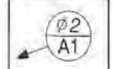
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US PATENT NO. 6,581,565

Symbols for toleranced properties

Properties		Symbols
Straightness	1)	—
Flatness		▱
Roundness (circularity)		○
Cylindricity		⌭
Profile of any line		⌒
Profile of any surface		⌓
Parallelism	2)	//
Perpendicularity		⊥
Slope		∠
Position	3)	⌖
Concentricity, coaxiality		◎
Symmetry		⌯
Path	4)	↗
Overall path		⌒↗

1) Tolerances of form, 2) Tolerances of direction,
3) Tolerances of location,
4) Running tolerances.

Supplementary symbols

Description		Symbols
Identification of the toleranced element	direct	
	with letter	
Identification of the reference	direct	
	with letter	
Reference point		
Theoretically precise dimension		50
Projected tolerance zone		(P)
Maximum material condition		(M)

Geometrical deviations

The term "geometrical deviation" refers to all disparities between the actual surface and its ideal geometrical configuration.

Critical considerations may be restricted to dimensional deviations, surface irregularities or undulations, or may encompass the entire potential deviation range, depending upon the specified application. To help distinguish between the different types of geometrical deviation DIN 4760 defines a classification system including examples of deviation types and their possible sources:

1st order geometrical deviations
Formal deviations are 1st order geometrical irregularities which can be detected when an entire surface area is examined. Usually, the general relationship between formal deviation intervals and depth is > 1000:1.

2nd to 5th order geometrical deviations
Undulations, or waviness, are geometrical irregularities of the 2nd order. These essentially periodic irregularities are defined using a representative section of the actual surface area on the geometrical element under investigation.
Roughness consists of geometrical irregularities of orders 3 through 5. It is characterized by periodic or irregular formal deviations in which the deviation intervals are a relatively low multiple of their depth.

Surface parameters

Stylus instruments are employed to record sectioning planes of surface profiles, registering vertically, horizontally, and as a function of the two components in combination. The characteristics are derived from the unfiltered primary profile (P profile), the filtered roughness profile (R profile), and the filtered waviness profile (W

PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

profile). Roughness and waviness are differentiated by means of profile filters.

Profile height P_t, R_t, W_t
Maximum excursion above the centerline Z_p plus the maximum excursion below the centerline Z_v within the traversing length l_n.

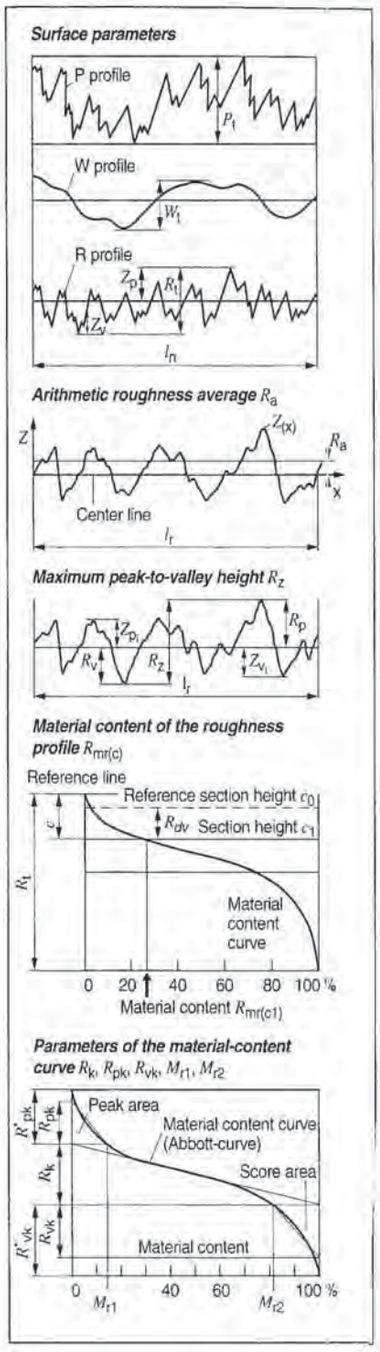
The traversing length l_n can consist of one or more sampling lengths l_r. The value of a parameter is computed from the measured data for its sampling length. As a rule, five sampling lengths are required for computing the roughness and waviness parameters. The standardized measuring conditions are based on the maximum roughness peak-to-valley height and include, for example, limit wave length λ_c, sampling length l_r, traversing length l_n, profile length l_t, stylus-tip radius $r_{SP\,max}$ and digitizing cutoff Δx_{max}). For industrial purposes, the parameters generally employed in describing surfaces are as follows:

Arithmetic roughness average R_a
Arithmetic average of the absolute values of the ordinates of the roughness profile.

Maximum peak-to-valley height R_z
Maximum excursion above the centerline R_p plus the maximum excursion below the centerline R_v within a sampling length. As the vertical distance between the highest peak and the lowest valley, R_z is a measure of scatter of the roughness ordinates.

Material content of the roughness profile $R_{mr(c)}$
The material-content curve reflects the material content as a function of section height c. The characteristic is formed by a percentage relationship between the sum of the material lengths Ml(c) of the profile elements in a specified section height c and traversing length l_n..

Parameters of the material-content curve R_k, R_{pk}, R_{vk}, Mr_1, Mr_2
The material-content curve derived from the filtered roughness profile is divided into three sections characterized by parameters (core peak-to-valley height R_k, reduced peak height R_{pk} and reduced score depth R_{vk}). The parameters Mr_1 and Mr_2 reflect the material content at the limits of the roughness core profile.



PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

Sliding bearings and rolling bearings

Sliding bearings

Different types of sliding bearings range from bearings with usually complete separation of the sliding surfaces by a lubrication film (fluid friction), through self-lubricating bearings, most of which exhibit mixed friction i.e. some of the bearing load is transformed to solid contact between the sliding surfaces), to dry sliding-contact bearings which exhibit dry friction (i.e. completely without an effective fluid lubricating film), but which nevertheless have an adequate service life.

Most of the hydrodynamic sliding bearing types used in automotive applications are circular cylindrical radial-sleeve bearings (often with oval clearance) for holding the crankshaft and camshaft, and turbocharger bearings. Thrust bearings are usually used only as axial locators, and are not subjected to large forces.

ydrodynamic sliding earings

nbols (DIN 31 652)

gnation	Symbols	Unit
bearing length bearing diameter nal diameter) diameter (nominal ter)	B D d	m m m
tricity (displacement en shaft and bearing s)	e	m
	F	N
bricant depth	h_0	m
ubricant pressure	p	Pa = N/m²
load $\bar{p} = F/(B \cdot D)$	$\bar{p}$	Pa
clearance $s = (D - d)$	s	m
rfeld number	So	–
eccentricity 2 e/s	ε	–
dynamic viscosity bricant	η_{eff}	Pa · s
bearing clearance	ψ	–
ment angle	β	°
namically effective elocity	ω_{eff}	s⁻¹

A hydrodynamic sliding bearing is reliable in service if it remains sufficiently unaffected by the following:
- Wear (sufficient separation of the contact surfaces by the lubricant),
- Mechanical stress (bearing material of sufficient strength),
- Thermal loading (observance of thermal stability of bearing material and viscosity/temperature behavior of lubricant).

The dimensionless Sommerfeld number So is used when determining loadability, i.e. when assessing the formation of the lubricating film

$$So = D \cdot \psi^2/(D \cdot B \cdot \eta_{eff} \cdot \omega_{eff})$$

As the Sommerfeld number So increases, relative eccentricity ε rises and minimum lubricating-film thickness h_0 drops:

$$h_0 = (D - d)/2 - e = 0.5\, D \cdot \psi \cdot (1 - \varepsilon)$$

with relative eccentricity

$$\varepsilon = 2e/(D - d)$$

Table 1. Orders of magnitude of the coefficients of friction for different types of friction
The coefficients of friction given below are approximate values, and are intended solely for comparison of the different types of friction.

Type of friction	Coefficient of friction f
Dry friction	0.1 .. > 1
Mixed friction	0.01 .. 0.1
Fluid friction	0.01
Friction in roller bearings	0.001

Table 2. Empirical values for maximum approved specific bearing load (Maximum values only apply for very low sliding velocities)

Bearing materials	Maximum specif. bearing load $\bar{p}_{lim}$
Pb and Sn alloys (babbitt metals)	5 ... 15 N/mm²
Bronze, lead base	7 ... 20 N/mm²
Bronze, tin base	7 ... 25 N/mm²
AlSn alloys	7 ... 18 N/mm²
AlZn alloys	7 ... 20 N/mm²

PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

The Sommerfeld number is also used to determine the coefficient of friction in the bearing, and thus for calculation of frictional power and thermal stress (cf. DIN 31 652, VDI Guideline 2204).

Because hydrodynamic bearings also operate with mixed friction a certain amount of the time, they must accommodate a certain amount of contamination without loss of function, and are additionally subjected to high dynamic and thermal stress (particularly in piston engines), the bearing material must meet a number of requirements, some of which are opposed to each another.

- Anti-seizure performance (resistance to welding),
- Conformability (compensation of misalignment without shortening service life),
- Embeddability (ability of the bearing surface to absorb particles of dirt without increasing bearing or shaft wear),
- Fatigue strength (under fatigue loading, particularly with high thermal stress),
- Mechanical loadability,
- Run-in performance (a combination of compatibility, resistance to wear and embeddability),
- Seizure resistance (bearing material must not weld to shaft material, even under high compressive load and high sliding velocity),
- Wear resistance (in the case of mixed friction),
- Wettability by lubricant.

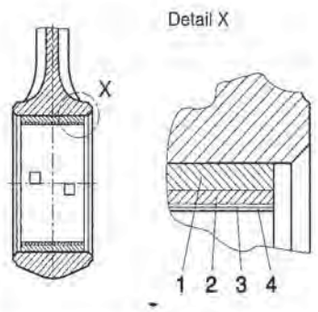
If a bearing (e.g. piston-pin bushing) is to be simultaneously subjected to high loads and low sliding velocities, high fatigue strength and wear resistance should take precedence over resistance to seizing. Bearing materials used in such cases are hard bronzes, e.g. leaded tin bronzes (Table 3).

Since they are subjected to high dynamic loads with high sliding velocities, connecting-rod and crankshaft bearings in internal-combustion engines must fulfill a number of different requirements. In these applications, **multilayer bearings**, above all trimetal bearings, have proved themselves in practice.

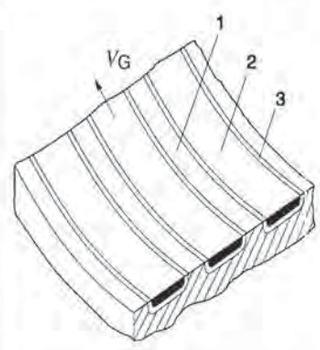
The service life of crankshaft bearings can be further increased through the use of **grooved sliding bearings**. In these bearings, fine grooves have been machined into the surface in the sliding direction, and filled with a liner made of soft material (an electroplated liner similar to that found in trimetal bearings). These contact areas are separated from one another by a harder light-alloy metal.

These bearings feature low rates of wear and high fatigue strength with good embeddability regarding lubricant impurities.

Multilayer bearing
(Design of a trimetal bearing)
1 Steel backing shell, 2 Cast lead bronze (0.4 mm), 3 Nickel barrier between lead bronze and babbitt-metal (1...2 µm), 4 Electroplated babbitt-metal liner (overlay, e.g., 20 µm).



Section through a grooved friction bearing
(MIBA patent). The liner has very fine grooves in the running direction V_G.
1 Wear-resistant light-alloy metal, 2 Soft liner, 3 Nickel barrier.



PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

Table 3. Selection of materials for hydrodynamic sliding bearings

Material	Alloy designation	Composition in %	HB hardness 20 °C	HB hardness 100 °C	Remarks, application examples
Tin-base babbitt metal	LgPbSn 80 (WM 80)	80 Sn; 12 Sb; 6 Cu; 2 Pb	27	10	Very soft, good conformance of contact surfaces to off-axis operation, excellent anti-seizure performance.
Lead-base babbitt metal	LgPbSn 10 (WM 10)	73 Pb; 16 Sb; 10 Sn; 1 Cu	23	9	Reinforcement necessary, e.g. as composite steel casting or with intermediate nickel layer on lead bronze.
Lead-base bronze	G-CuPb 25	74 Cu; 25 Pb; 1 Sn	50	47	Very soft, excellent anti-seizure performance, less resistant to wear.
	G-CuPb 22	70 Cu; 22 Pb; 6 Sn; 3 Ni	86	79	
Lead-tin-base bronze	G-CuPb 10 Sn	80 Cu; 10 Pb; 10 Sn	75	67	Improved anti-seizure performance by alloying with Pb. More resistant to off-axis performance than pure tin bronzes, therefore high-load Pb-Sn bronzes are preferable for use as crankshaft bearings. Composite bearings used in internal-combustion engine manufacture, piston-pins $\bar{p}$ to 100 N/mm².
	G-CuPb 23 Sn	76 Cu; 23 Pb; 1 Sn	55	53	Composite casting for low-load bearings (70 N/mm²). Also thick-wall bearing shells. Very good anti-seizure performance. Crankshaft and camshaft bearings, connecting-rod bearings.
Tin-base bronze	G-CuSn 10 Zn	88 Cu; 10 Sn; 2 Zn	85		Hard material. Friction-bearing shells can be subjected to moderate load at low sliding velocities. Worm gears.
	CuSn 8	92 Cu; 8 Sn	80 ... 220		High-grade wrought alloy. Good performance under high loads and in absence of sufficient lubrication. Steering-knuckle bearings. Particularly well suited for use as thin-wall sliding bearing bushings.
Red brass	G-CuSn7 ZnPb	83 Cu; 6 Pb; 7 Sn; 4 Zn	75	65	Tin partially replaced with zinc and lead. Can be used in place of tin bronze, but only for moderate loads (40 N/mm²). General sliding bearings for machinery. Piston pins, bushings, crankshaft and toggle-lever bearings.

PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

Material	Alloy designation	Composition in %	HB hardness 20 °C	HB hardness 100 °C	Remarks, application examples
Brass	CuZn 31 Si	68 Cu; 31 Zn; 1 Si	90 ... 200		Zn content is disadvantageous at high bearing temperatures. Can be used in place of tin bronze; low loads.
Aluminum bronze	CuAl 9 Mn	88 Cu; 9 Al; 3 Mn	110 ... 190		Thermal expansion comparable to that of light alloys. Suitable for use as interference-fit bearings in light-alloy housings. Better wear resistance than tin bronze, but higher friction.
Aluminum alloy	AlSi 12 Cu NiMn	1 Cu; 85 Al; 12 Si; 1 Ni; 1 Mn	110	100	Piston alloy for low sliding velocities.
Rolled aluminum cladding	AlSn 6	1 Cu; 6 Sn; 90 Al; 3 Si	40	30	Liquated tin stretched by rolling, therefore high loadability and good anti-friction properties. Improved by electroplated layer.
Electro-plated liners	PbSn 10 Cu	2 Cu; 88 Pb; 10 Sn	50 ... 60		Used in modern trimetal bearings, electroplated to thickness 10 ... 30 µm, very fine grain. Intermediate nickel layer on bearing metal.

For materials, see also DIN 1703, 1705, 1716, 17 660, 17 662, 17 665, 1494, 1725, 1743.
ISO 4381, 4382, 4383.

Table 4. Sintered-metal bearing materials
Sint-B indicates 20 % P (porosity) (Sint-A: 25 % P; Sint-C: 15 % P).

Material group	Designation Sintered ...	Composition	Remarks
Sintered iron	B 00	Fe	Standard material which meets moderate load and noise requirements.
Sintered steel, containing Cu	B 10	< 0.3 C 1 ... 5 Cu Rest Fe	Good resistance to wear, can be subjected to higher loads than pure Fe bearings.
Sintered steel, higher Cu content	B 20	20 Cu Rest Fe	More favorably priced than sintered bronze, good noise behavior and $p \cdot v$ values.
Sintered bronze	B 50	< 0.2 C 9 ... 10 Sn Rest Cu	Standard Cu-Sn-base material, good noise behavior.

PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

Sintered-metal sliding bearings

Sintered-metal sliding bearings consist of sintered metals which are porous and impregnated with liquid lubricants. For many small motors in automotive applications, this type of bearing is a good compromise in terms of precision, installation, freedom from maintenance, service life and cost. They are primarily used in motors with shaft diameters from 1.5...12 mm. Sintered-iron bearings and sintered-steel bearings (inexpensive, less likely to interact with the lubricant) are preferable to sintered-bronze bearings for use in the motor vehicle (Table 4). The advantages of sintered-bronze bearings are greater loadability, lower noise and lower friction indices (this type of bearing is used in phonograph equipment, office equipment, data systems and cameras).

The performance of sintered bearings over long periods of service is closely related to the use of optimum lubricants.

Mineral oils: Inadequate cold-flow properties, medium non-aging properties.

Synthetic oils: (e.g. esters, Poly-α olefins): Good cold-flow properties, high resistance to thermal stresses, low evaporation tendency.

Synthetic greases: (oils which include metal soaps): Low starting friction, low wear.

Dry sliding bearings

(see Table 5, P. 288)

Solid polymer bearings made of thermoplastics

Advantages: Economically-priced, no danger of seizure with metals.

Disadvantages: Low thermal conductivity, relatively low operating temperatures, possible swelling due to humidity, low loadability, high coefficient of thermal expansion.

The most frequently used polymer materials are: Polyoxymethylene (POM, POM-C), polyamide (PA), polyethylene and polybutylene terephthalate (PET, PBT), polyetheretherketone (PEEK).

The tribological and mechanical properties can be varied over a wide range by incorporating lubricants and reinforcements in the thermoplastic base material.

Lubricant additives: polytetrafluoroethylene (PTFE), graphite (C), silicone oil and other liquid lubricants.

Reinforcement additives: glass fibers (GF), carbon fibers (CF).

Application examples: windshield wiper bearings (PA and fiberglass), idle-position actuators (PEEK and carbon fiber, PTFE and other additives).

Polymer bearings made of duroplastics and elastomers

These materials, with their high levels of intrinsic friction, are seldom used as bearing materials in the motor vehicle. Duroplastics include: phenol resins (high friction, e.g. Resitex), epoxy resins (require the addition of PTFE or C in order to reduce their high intrinsic brittleness, reinforcement necessary usually by fibers), polyimides (high thermal and mechanical loadability).

Application examples: polyimide axial stop in wiper motor.

Metal-backed composite bearings

Composite bearings are combinations of polymer materials, fibers and metals. Depending upon bearing structure, they provide advantages over pure or filled polymer sliding bearings in terms of loadability, bearing clearance, thermal conductivity and installation (suitable for use with oscillating motion).

Example of bearing structure: Tinplated or copper-clad steel backing (several millimeters thick), onto which is sintered an 0.2...0.35 mm thick porous bronze layer with a porosity of 30...40%. A low-friction polymer material is rolled into this bronze layer as a liner. Liner made of a) acetal resin, either impregnated with oil or containing lubricating recesses, or b) PTFE + Pb or MoS_2 additive.

Metal-backed composite bearings are available in a number of different shapes and compositions. Metal-backed composite bearings with woven PTFE fiber inserts exhibit unusually high loadability

PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

and are suitable for use in ball-and-socket joints.

Some motor-vehicle applications for this type of bearing are: Shock-absorber-piston bearings, release-lever bearings for clutch pressure plates, brake-shoe bearings in drum brakes, ball-and-socket bearings, door-hinge bearings, bearings for seat-belt winding shafts, steering-knuckle bearings, gear-pump bearings.

Carbon-graphite bearings

Carbon-graphite bearings are members of the ceramic bearing family due to their method of manufacture and material properties. The base materials are powdered carbon; tar or synthetic resins are used as binders.

Advantages: Heat-resistant up to 350 °C (hard-burnt carbon) or 500 °C (electrographite), good antifriction properties, good corrosion resistance, good thermal conductivity, good thermal shock resistance. They are highly brittle, however.

Examples of carbon-graphite bearing applications: Fuel-pump bearings, bearings in drying ovens, adjustable blades in turbochargers.

Metal-ceramic bearings

Metal-ceramic bearings consist of material manufactured by powder metallurgy processes; in addition to the metallic matrix, the bearing material also contains finely distributed solid lubricant particles.

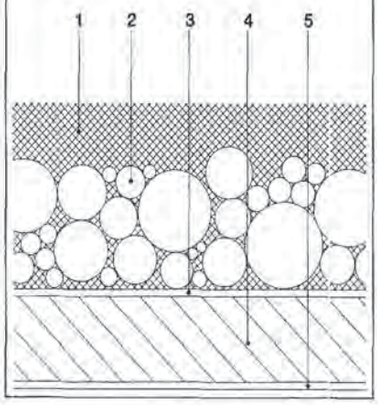
Matrix: e.g. bronze, iron, nickel.

Lubricant: e.g. graphite, MoS_2.

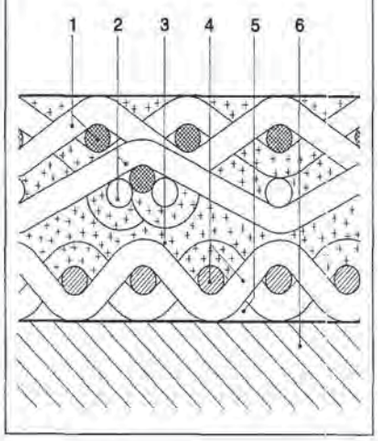
These materials are suitable for use under extremely high loads, and are at the same time self-lubricating.

Application example: Steering-knuckle bearings.

Cutaway view of a self-lubricated composite bearing
1 Polymer liner, 2 Porous bronze layer, 3 Copper layer, 4 Steel backing, 5 Tin layer.



Section through composite bearing with fabric insert made of PTFE and fiberglass
1 PTFE fiber fabric, 2 Adhesive fibers, 3 Resin, 4 Fiberglass backing, 5 Adhesive, 6 Steel backing.



PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

Table 5. Properties of maintenance-free, self-lubricating bearings

	Sintered bearings oil-impregnated		Polymer bearings		Metal-backed composite bearings Liner		Synthetic carbons
	Sintered iron	Sintered bronze	Thermoplastic polyamide	Duroplastic polyimide	PTFE + additive	Acetal resin	
Compression strength N/mm²	80 ... 180		70	110	250	250	100 ... 200
Max. sliding velocity m/s	10	20	2	8	2	3	10
Typical load N/mm²	1 ... 4 (10)		15	50 (at 50 °C) 10 (at 200 °C)	20 ... 50	20 ... 50	50
Permissible operating temperature °C Short-term	− 60 ... 180 (depends on oil) 200		− 130 ... 100 120	− 100 ... 250 300	− 200 ... 280	− 40 ... 100 130	− 200 ... 350 500
Coefficient of friction without lubrication	with lubrication 0.04 ... 0.2		0.2 ... 0.4 (100 °C) 0.4 ... 0.6 (25 °C)	0.2 ... 0.5 (unfilled) 0.1 ... 0.4 (filled)	0.04 ... 0.2	0.07[1]) ... 0.2 [1]) PTFE filled	0.1 ... 0.35
Thermal conductivity W/(m · K)	20 ... 40		0.3	0.4 ... 1	46	2	10 ... 65
Corrosion resistance	poor	good	very good		good	good	very good
Chemical resistance	none		very good		conditional	conditional	good
max. $p \cdot v$ (N/mm²) · (m/s)	20		0.05	0.2	1.5 ... 2		0.4 ... 1.8
Embeddability of dirt and abraded material	poor		good	good	poor	good	poor

PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

Rolling bearings

In rolling bearings, forces are transmitted by rolling elements (balls or rollers). Here, rolling is a form of movement comprising rolling and sliding. In rolling bearings a microslip (i.e. sliding) nearly always occurs in addition to the pure rolling movement. In the case of mixed friction, this sliding movement leads to increased wear.

Advantages:
Low static coefficient of friction (0.001...0.002), therefore particularly well suited to applications in which starting occurs frequently,
Low maintenance,
Suited to permanent lubrication,
Low lubricant consumption,
Small bearing width,
High precision.

Disadvantages:
Sensitive to impact loads, sensitive to dirt,
Bearing noise is too high for some applications,
Standard bearings are only 1-piece bearings.

Bearing materials

Bearing races and rolling elements are made of a special chromium-alloy steel (100 Cr6H) which has a high degree of purity and a hardness in the range of 58...65 HRC.

Depending upon application, the bearing cages are made of stamped sheet steel or brass. Cages made of polymer materials have come into recent use due to their ease of manufacture, improved adaptability to bearing geometry and other tribological advantages (e.g. antifriction properties). Polymer materials made of fiberglass-reinforced polyamide 66 can withstand continuous operating temperatures of up to 120 °C, and can be operated for brief periods at temperatures up to 140 °C.

Static loadability (ISO 76-1987)

The static load rating Co is used as a measure of the loadability of very slow-moving or static rolling bearings. Co is that load at which total permanent deformation of rolling elements and races at the most highly-loaded contact point amounts to 0.0001 of the diameter of the rolling element. A load equal to this load rating Co generates a maximum compressive stress of 4000 N/mm² in the center of the most highly-loaded rolling element.

Dynamic loadability

The basic load rating C is used for calculating the service life of a rotating rolling bearing. C indicates that bearing load for a nominal bearing service life of 1 million revolutions. In accordance with ISO 281:

Service life equation $L_{10} = \left(\frac{C}{P}\right)^p$

L_{10} Nominal service life in millions of revolutions achieved or exceeded by 90 % of a large batch of identical bearings,
C Basic load rating in N (determined empirically),
P Equivalent dynamic bearing load in N,
p Empirical exponent of the service life equation: for ball bearings, $p = 3$, for roller bearings, $p = 10/3$

Modified nominal service life

$$L_{na} = a_1 \cdot a_2 \cdot a_3 \cdot \left(\frac{C}{P}\right)^p$$

L_{na} Modified nominal service life in millions of revolutions,
a_1 Factor for survival probability, e.g. 90 %: $a_1 = 1$; 95 %: $a_1 = 0.62$,
a_2 Material coefficient of friction,
a_3 Coefficient of friction for operating conditions (bearing lubrication).

As the frictional coefficients a_2 and a_3 are not mutually independent, the combined coefficient a_{23} is employed.

Rolling bearings are subject to a fatigue stress. It is included, along with other influencing parameters such as contamination, in the calculation of service life (see catalogs issued by rolling-bearing manufacturers).

PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

Spring calculations

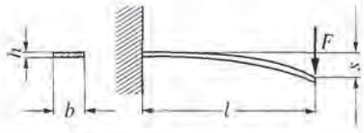
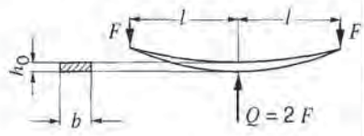
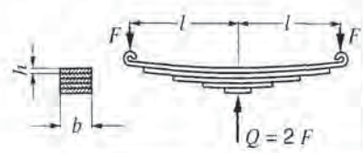
See PP. 14 ... 16 for names of units

Quantity		Unit
b	Width of spring leaf	mm
c	Spring rate (spring constant): For leaf springs and helical springs, increase in spring force per mm of spring deflection; For torsion bar springs, coiled torsion springs and flat spiral springs, increase in spring moment per degree of rotation	N/mm N · mm/°
d	Wire diameter	mm
D	Mean coil diameter	mm
E	Modulus of elasticity: for spring steel = 206,000 N/mm², for other materials, see P. 196	N/mm²

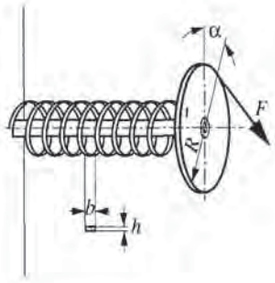
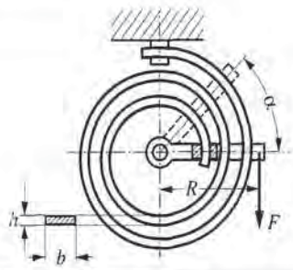
Quantity		Unit
F	Spring force	N
G	Shear modulus: for hot-formed springs ≈ 78,500 N/mm² for cold-formed springs ≈ 81,400 N/mm²	N/mm²
h	Height of spring leaf	mm
n	Number of active coils	–
M	Spring moment	N · mm
s	Spring deflection	mm
α	Angle of rotation	°
σ_b	Bending stress	N/mm²
τ	Torsional stress	N/mm²

Conversion of no longer permissible units:
1 kp = 9.81 N ≈ 10 N, 1 kp/mm ≈ 10 N/mm, 1 kp mm ≈ 10 N mm,
1 kp mm/° ≈ 10 N mm/°, 1 kp/mm² ≈ 10 N/mm².

Springs subjected to bending stress

Type	Bending stress	Spring rate
Rectangular cantilever spring (constant cross section) 	$\sigma_b = \frac{6\,l}{b \cdot h^2} F$	$c = \frac{F}{s} = \frac{E \cdot b \cdot h^3}{4 \cdot l^3}$
See fatigue limit diagram on P. 291.		
Single-leaf spring, rolled out parabolically (vehicle spring) 	$\sigma_b = \frac{3\,l}{b \cdot h_0^2} Q$	$c = \frac{Q}{s} = \frac{E \cdot b \cdot h_0^3}{4 \cdot l^3}$
Multi-leaf spring, constant cross-section of the leaves (vehicle spring) 	$\sigma_b = \frac{3\,l}{n \cdot b \cdot h^2} Q$	$c = \frac{(2 + n'/n)\, E \cdot n \cdot b \cdot h^3}{6 \cdot l^3}$
n = number of leaves n' = number of leaves at ends of spring		

PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

Type	Bending stress	Spring rate
Coil spring (DIN 2088) 	The same equations are used for torsion and spiral springs. Rectangular cross-section $\sigma_b = \frac{6}{b \cdot h^2} M$	$c = \frac{M}{\alpha} = \frac{E \cdot b \cdot h^3}{57.3 \cdot 12 \cdot l}$
	Circular section $\sigma_b = \frac{32}{\pi \cdot d^3} M$	$c = \frac{M}{\alpha} = \frac{E \cdot \pi \cdot d^4}{57.3 \cdot 64 \cdot l}$
	where $M = F \cdot R$	
	l = straight length of spring material = $2 \pi \cdot r \cdot n$ (r mean coil radius)	
Spiral torsion spring (both ends clamped) 	Under a pulsating load the stress increase due to curvature must be taken into consideration.	

Fatigue-limit diagram

For polished spring-steel strip (not for vehicle springs) in accordance with DIN 17 222 Ck 75 and Ck 85, hardened and tempered to 430...500 HV[1]), and Ck 85 and Ck 101, hardened and tempered to 500...580 HV.

Example (broken line in diagram):
For a leaf spring with a hardness of 430...500 HV, assuming a minimum stress of

$\sigma_U = 600$ N/mm²

the permissible stress range is

$\sigma_{hperm} = 480$ N/mm²

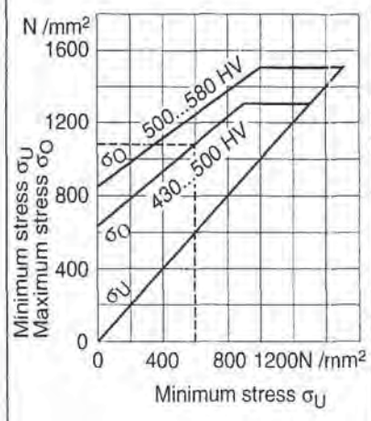
and the overstress is

$\sigma_O = \sigma_U + \sigma_{hperm} = 1080$ N/mm².

For reasons of safety, the stress range selected should be approximately 30% lower. Surface damage (corrosion pits, score marks) drastically reduces the fatigue limit.

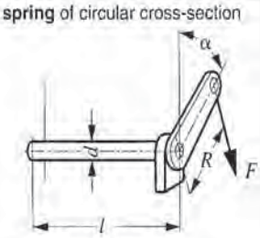
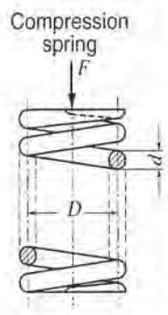
[1]) Vickers hardness; see P. 276.

Bending



PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

Springs subjected to torsional stress

Type	Torsional stress	Spring rate
Torsion-bar spring of circular cross-section (DIN 2091) 	$\tau = \frac{16}{\pi \cdot d^3} M$ where $M = F \cdot R$	$c = \frac{M}{\alpha} = \frac{G \cdot \pi}{57.3} \frac{d^4}{32 \cdot l}$
Helical springs made of round wire (DIN 2089)   Above: half "German" loop, Below: hifo® hook (see P. 295)	a) In the case of a static or infrequently alternating load, allowance need not be made for the effect of the wire curvature: $\tau = \frac{8 \cdot D}{\pi \cdot d^3} F$ b) In the case of a pulsating load, the non-uniform distribution of torsional stresses must be taken into consideration: $\tau_k = k \cdot \tau = \frac{8\,k \cdot D}{\pi \cdot d^3} F$ The curvature correction factor k depends on the spring index D/d.	$c = \frac{F}{s} = \frac{G \cdot d^4}{8 \cdot n \cdot D^3}$ In the case of compression springs with only 1 ... 3 coils, the load deflection curve is no longer a straight line; spring rate c is initially smaller and then larger than the calculated rate.

D/d	3	4	6	8	10	20
k	1.55	1.38	1.24	1.17	1.13	1.06

Avoid D/d < 3 and > 20.

The maximum permissible stresses τ_{k0} for various wire diameters at temperatures up to approximately 40 °C can be taken from the vertical boundary lines of the RDZ diagrams on the next pages (RDZ = relaxation, fatigue limit, fatigue strength for finite life). Interpolation is to be used for intermediate values. At higher temperatures, the relaxation curves (loss of load) must be considered. After 300 h, relaxation values are approximately 1.5 times as great as after 10 h. Values for other materials are given on P. 196.

In the case of extension springs with initial tension, maximum permissible stress values must be reduced to 90 % of the values indicated in the diagrams because so-called setting is not possible here.

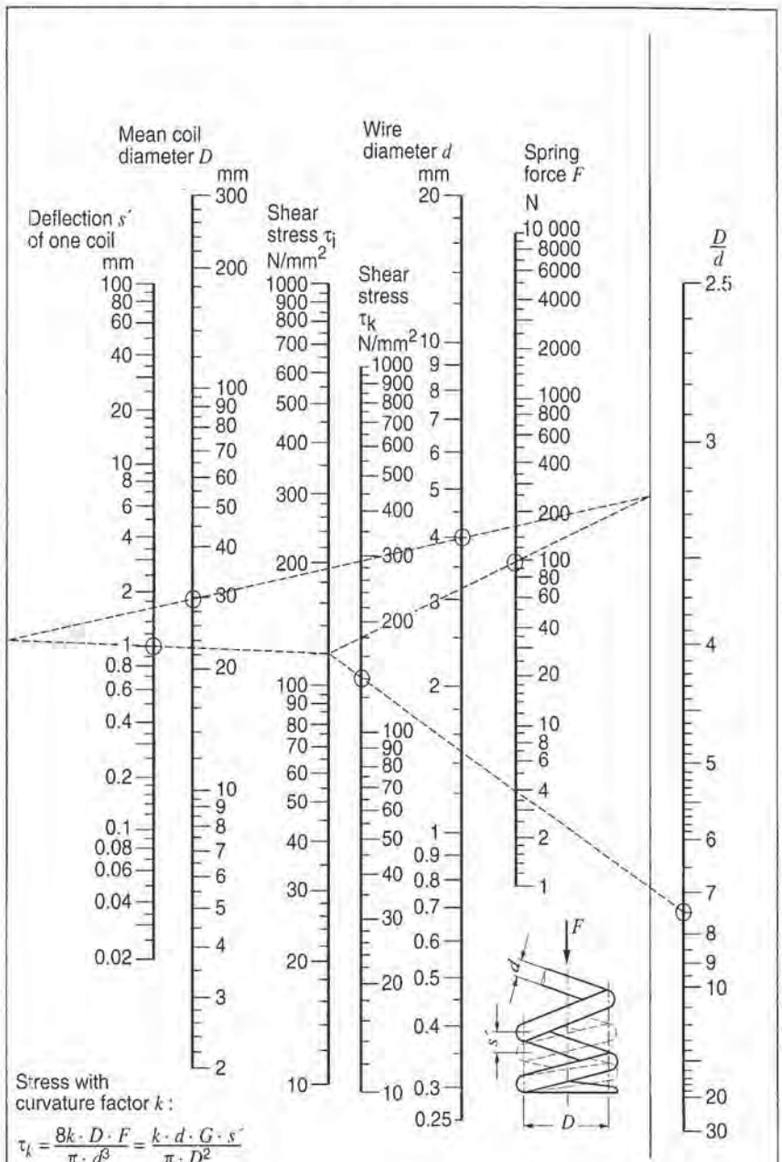
Explanation of the nomogram on the following page

The nomogram applies to cylindrical helical extension and compression springs made of round steel wire (shear modulus G = 81,400 N/mm²). In the case of materials with a different shear modulus G' spring deflection must be multiplied by G/G'.

The nomogram indicates the deflection s' of one coil. Total deflection s is obtained by multiplying (s') by the number of active coils n: $s = n \cdot s'$.

PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

Nomogram for helical spring calculation



Stress with curvature factor k:

$$\tau_k = \frac{8k \cdot D \cdot F}{\pi \cdot d^3} = \frac{k \cdot d \cdot G \cdot s'}{\pi \cdot D^2}$$

k see P. 292.

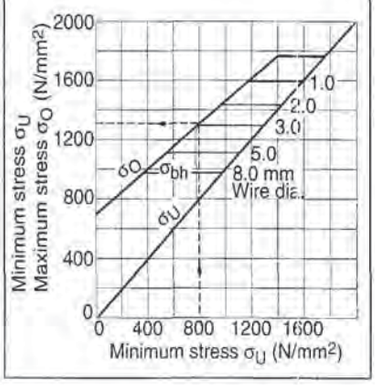
Example: given: D = 30 mm, d = 4 mm, F = 100 N
found: τ_i = 120 N/mm², τ_k = 140 N/mm², s' = 1.02 mm.

PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

Fatigue-limit diagram for torsion springs

Spring-steel wire, type C (DIN 17 223, Sheet 1). Torsion springs not shot-peened.
Example (broken line in diagram):
For a torsion spring with a wire diameter d = 2 mm, the permissible stress range to which the spring can be subjected for 10^7 load cycles and more without fracture is σ_{bh} = 500 N/mm² at a minimum stress of σ_U = 800 N/mm².
The maximum stress is then
$\sigma_O = \sigma_U + \sigma_{bh}$ = 1300 N/mm².

Bending, Type C



Diagrams of relaxation fatigue limit and fatigue strength for finite life (RDZ)

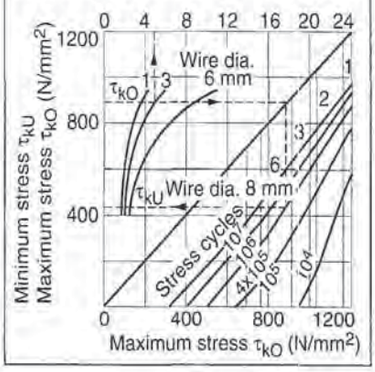
Spring-steel wire, type C (DIN 17 223, Sheet 1). Helical compression springs not shot-peened, cold-set.
Example (broken line in diagram):
A helical compression spring with a wire diameter of 3 mm a maximum stress of τ_{kO} = 900 N/mm² at an operating temperature of 80 °C, has a loss of spring force after 10 h of

ΔF_{10} = 5.5 %.

Given $N = 4 \times 10^5$ load cycles to failure, the stress range would be
$\tau_{kh} = \tau_{kO} - \tau_{kU}$ = 900 – 480 = 420 N/mm².

Torsion, Type C

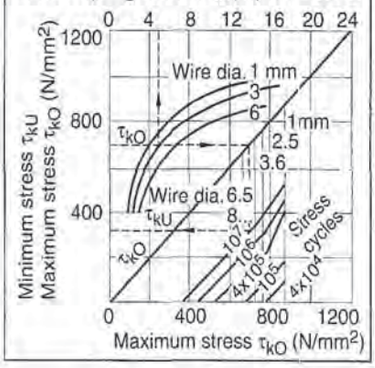
Loss of spring force ΔF_{10} (%) at t = 80 °C



Valve-spring steel wire, VD (DIN 17 223, Sheet 2). Helical compression springs not shot-peened, cold-set.
Example (broken line in diagram):
A helical compression spring with a wire diameter of 3 mm and a maximum stress of τ_{kO} = 700 N/mm² at an operating temperature of 80 °C, has a loss of spring force after 10 h of ΔF_{10} = 5 %.
For an infinite number of load cycles ($N \geq 10^7$), a stress range of
τ_{kh} = 700 – 320 = 380 N/mm²
is permissible.

Torsion, VD

Loss of spring force ΔF_{10} (%) at t = 80 °C



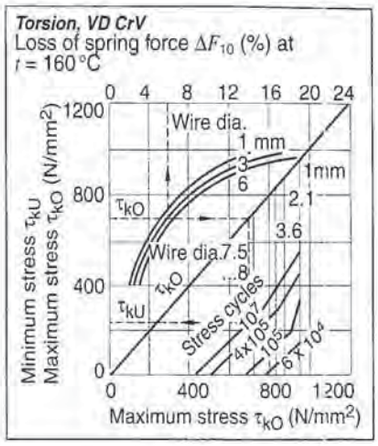
PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

Valve-spring steel wire, VD CrV.[1])
Helical compression springs not shot-peened, cold-set.
Example (broken line in diagram):
A helical compression spring with a wire diameter of 3 mm and a maximum stress of τ_{kO} = 700 N/mm² at an operating temperature of 160 °C, has a loss of spring force after 10 h of ΔF_{10} = 6 %.

For an infinite number of load cycles ($N \geq 10^7$), a stress range of

τ_{kh} = 700 – 240 = 460 N/mm²

is permissible.

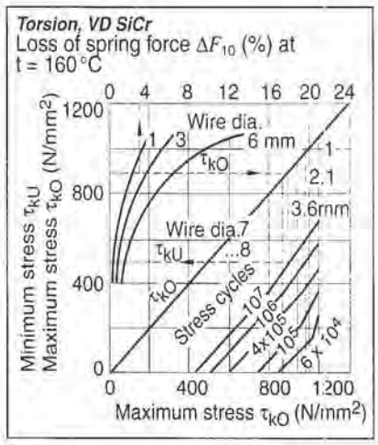


Valve-spring steel wire, VD SiCr.[1])
Helical compression springs not shot-peened, cold-set.
Example (broken line in diagram):
A helical compression spring with a wire diameter of 3 mm and a maximum stress of τ_{kO} = 900 N/mm² at an operating temperature of 160 °C, has a loss of spring force after 10 h of ΔF_{10} = 3.5 %.

For an infinite number of load cycles ($N \geq 10^7$), a stress range of

τ_{kh} = 900 – 510 = 390 N/mm²

is permissible.



If the danger of natural oscillation or other additional loads is present, reduce the selected stress range by approximately 30 %. Shot peening increases the permissible stress range by approximately 40 %.

In the case of **compression springs**, force eccentricity is reduced as far as possible by winding the spring such that the wire end at each end of the spring touches the adjoining coil (closed ends). Each end of the spring is then ground flat perpendicular to the axis of the spring (wire end = $d/4$). The following values then apply:
Total number of coils $n_1 = n + 2$.
The solid length from coil to coil $\leq (n + 2)\, d$.

[1]) According to DIN 17 223, Part 2.

Helical extension springs are overstressed by roughly 50 % at standard hooks and loops when the body of the spring is subjected to permissible stress ranges. Hifo® hooks (German Patent, and others) exhibit infinite fatigue strength due to internal high initial force F_O.

Due to their favorable internal stresses, twist-wound extension springs can be subjected to stresses as high as those to which compression springs are subjected, and exhibit smaller relaxation than the latter. For high initial forces F_O (= hifo), these springs can be up to 50 % shorter at $F1$ than standard extension springs

PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

Gears and tooth systems

(with involute flanks)

Quantities and units (DIN 3960)

Quantity		Unit
a	Center distance	mm
b	Face width	mm
c	Bottom clearance	mm
d	Reference diameter	mm
d_a	Outside diameter	mm
d_b	Base circle diameter	mm
d_f	Root diameter	mm
d_w	Pitch diameter	mm
h_a	Addendum = $m \cdot h^*_{aP}$	mm
h_f	Dedendum = $m \cdot h^*_{fP}$	mm
i	Transmission ratio = z_2/z_1	–
j_n	Normal backlash	mm
m	Module m = d/z	mm
n	Rotational speed	min^{-1}
p	Pitch $p = \pi \cdot m$	mm
s	Tooth thickness (circular)	mm
W	Span measurement	mm
x	Addendum modification coefficient	–

Quantity		Unit
z	Number of teeth	–
α	Pressure angle	°
β	Helix angle	°
ε	Contact ratio	°
*	Specific value, to be multiplied by m	
Superscripts and subscripts		
1	referred to gear 1	
2	referred to gear 2	
a	referred to tooth tip	
b	referred to base circle	
f	referred to root	
n	referred to normal profile	
t	referred to transverse profile	
w	referred to operating pitch circle	
F	referred to root tooth load	
P	referred to basic rack	
W	referred to contact pressure	

Gear type and shape are determined by the position of the shafts which are joined by the gears in order to transmit forces or movements.

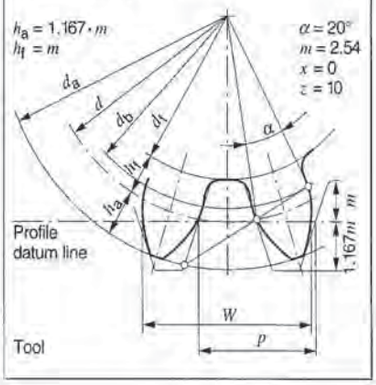
Cycloidal teeth: Cycloidal teeth are used primarily in the watch and clock industry. They permit small numbers of teeth without cutter interference (undercut). They are characterized by low contact pressure, but are sensitive to variations in the distance between centers.

Involute teeth: Involute teeth, on the other hand, are not sensitive to variations in the distance between centers.

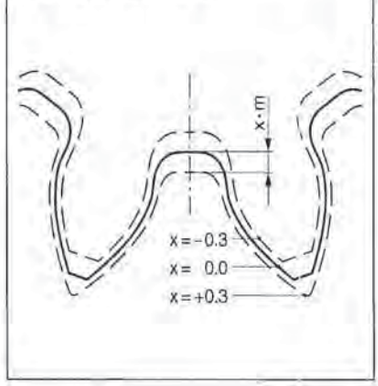
They can be produced with relatively simple tools using the generating method. The automobile industry uses involute teeth almost exclusively, therefore the following information is limited to this type of gear system.

All spur gears with the same module (and the same pressure angle) can be produced using the same generating tool, regardless of the number of teeth and addendum modification. A number of modules have been standardized in order to

Parameters for spur gears *(cycloidal teeth)*



Effect of addendum modification at spur gear *(cycloidal teeth)*



PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

Definitions

Table 1. Gear types

Position of shafts	Gear type	Properties	Examples of application in automotive engineering
Parallel	External or internal spur-gear pair with straight or helical teeth	Helical teeth run more smoothly but are subject to axial thrust	Manual transmissions
Intersecting	Bevel-gear pair with straight, helical, or curved teeth	Sensitive to fluctuations in angular and axial clearance	Differentials
Crossing	Offset bevel-gear pair with helical or curved teeth		Rear-axle drives
	Crossed helical-gear pair	For small loads	Distributor drives
	Worm-gear pair	High-ratio, single-stage transmission	Windshield wiper drives
Coaxial	Toothed shaft and hub	Sliding shaft coupling	Starters (Bendix shaft and pinion)

limit the number of such tools and the number of master gears required for gear testing:

Module for spur and bevel gears: DIN 780 (see table on P. 299), module for worms and worm gears: DIN 780, module for toothed shafts and stubs: DIN 5480. The module in the normal profile for crossed helical gears is also selected in conformance with DIN 780 in most cases. A series of modules suited to the manufacturing process is generally used for curved-tooth bevel gears.

Tooth shape

Basic rack for spur gears: DIN 867, DIN 58 400; basic rack for bevel gears: DIN 3971; basic rack for worms and worm gears: DIN 3975; basic rack for toothed shafts and hubs: DIN 5480.

The tooth shape for crossed helical gears can also be designed in conformance with the basic rack as per DIN 867. In addition to the standardized pressure angles (20° for running gears and 30° for toothed shafts and hubs), pressure angles of 12°, 14°30', 15°, 17°30', 22°30' and 25° are used.

Table 2. List of standards

Standard	Title
DIN 3960	Definitions, parameters and equations for involute cylindrical gears and gear pairs
DIN 3961...4	Tolerances for cylindrical gear teeth
DIN 3990/1...5	Calculation of load capacity of cylindrical gears
DIN 58 405/1...4	Spur gear drives for precision engineering
DIN 3971	Definitions and parameters for bevel gears and bevel gear pairs
DIN 5480	Involute spline joints
DIN 3975	Terms and definitions for cylindrical worm gears with shaft angle 90°

PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

Table 3. Basic equations for spur gears

Designation	Straight gear ($\beta = 0$)	Helical gear ($\beta \neq 0$)
Reference-circle diameter	$d = z \cdot m$ Module m, see P. 296	$d = \frac{z \cdot m_n}{\cos \beta}$ Module m_n, see P. 296
	The neg. value of z is to be used for internal teeth; center distance a becomes neg.	
Base-circle diameter	$d_b = d \cdot \cos \alpha_n$	$d_b = d \cdot \cos \alpha_t$ α_t from $\tan \alpha_t = \frac{\tan \alpha_n}{\cos \beta}$
Center distance, no backlash; if normal backlash $j_n = 0$	$a = m \frac{z_1 + z_2}{2} \cdot \frac{\cos \alpha}{\cos \alpha_w} = \frac{d_{b1} + d_{b2}}{2 \cdot \cos \alpha_w}$ as per DIN 867, $\alpha = 20°$ α_w from: $\text{inv}\, \alpha_w = \frac{2 (x_1 + x_2) \sin \alpha + j_n/m}{(z_1 + z_2) \cos \alpha} + \text{inv}\alpha$	$a = m_t \frac{z_1 + z_2}{2} \cdot \frac{\cos \alpha_t}{\cos \alpha_{wt}} = \frac{d_{b1} + d_{b2}}{2 \cdot \cos \alpha_{wt}}$ $m_t = \frac{m_n}{\cos \beta}$ $\cos \beta_b = \frac{\sin \alpha_n}{\sin \alpha_t}$ $\text{inv}\, \alpha_{wt} = \frac{m_n \cdot 2(x_1 + x_2) \sin \alpha_n + j_n/\cos \beta_b}{m_t (z_1 + z_2) \cos \alpha_t} + \text{inv}\alpha_t$
Involute	$\text{inv}\, \alpha = \tan \alpha - \text{arc}\, \alpha$ P. 146.	when $\alpha = 20°$, $\text{inv}\, \alpha = 0.014904$
Pitch-circle diameter	$d_w = d_b / \cos \alpha_w$	$d_w = d_b / \cos \alpha_{wt}$
Root-circle diameter	$d_f = d + 2\, xm - 2\, h^*_{fP} \cdot m$	$d_f = d_b + 2\, xm_n - 2\, h^*_{fP} \cdot m_n$
	with dedendum $h^*_{fP} \cdot m$ ($h^*_{fP} = 1.167$ or 1.25 as per DIN 3972)	
Tip-circle diameter	$d_a = d + 2\, xm + 2\, h^*_{aP} \cdot m$	$d_a = d \cdot 2\, xm_n + 2\, h^*_{aP} \cdot m_n$
	with addendum $h^*_{aP} \cdot m$ ($h^*_{aP} = 1.0$ as per DIN 867)	
Bottom clearance	$c_1 = a_{min} - d_{a1}/2 - d_{f2}/2$; $c_2 = a_{min} - d_{a2}/2 - d_{f1}/2$	$c \geq 0.15\, m_n$
Tooth thickness in pitch circle (circular)	$s = m\, (\pi/2 + 2\, x \cdot \tan \alpha)$	Normal profile $s_n = m_n (\pi/2 + 2\, x \cdot \tan \alpha_n)$
		Transverse profile $s_t = s_n/\cos \beta = m_t\, (\pi/2 + 2 \cdot x \cdot \tan \alpha_n)$
Ideal number of teeth	$z_i = z$	$z_i = z \frac{\text{inv}\, \alpha_t}{\text{inv}\, \alpha_n}$
Number of teeth spanned	$k = z_i \frac{\alpha_{nx}}{180} + 0.5$ for k take the next whole number	α_{nx} from $\cos \alpha_{nx} = \frac{z_i}{z_i + 2x} \cos \alpha_n$
Base tangent length over k teeth	$W_k = \{[(k' - 0.5)\, \pi + z_i\, \text{inv}\, \alpha_n] \cos \alpha_n + 2x \cdot \sin \alpha_n\}\, m_n$	
Back-reckoning of x from the base tangent	$x = \frac{(W/m_n) - [(k - 0.5) \cdot \pi + z_i \cdot \text{inv}\, \alpha_n] \cdot \cos \alpha_n}{2 \cdot \sin n\alpha}$	
Transverse contact ratio for gears without allowance for cutter interference	$\varepsilon_\alpha = \frac{\sqrt{d_{a1}^2 - d_{b1}^2} + \frac{z_2}{\lvert z_2 \rvert} \sqrt{d_{a2}^2 - d_{b2}^2} - (d_{b1} + d_{b2}) \tan \alpha_w}{2 \cdot \pi \cdot m_t \cdot \cos \alpha}$ (straight teeth)	
	$\varepsilon_\alpha = \frac{\sqrt{d_{a1}^2 - d_{b1}^2} + \frac{z_2}{\lvert z_2 \rvert} \sqrt{d_{a2}^2 - d_{b2}^2} - (d_{b1} + d_{b2}) \tan \alpha_{wt}}{2 \cdot \pi \cdot m_t \cdot \cos \alpha}$ (helical teeth)	
Overlap ratio	–	$\varepsilon_\beta = \frac{b \cdot \sin \lvert \beta \rvert}{m_n \cdot \pi}$
Total contact ratio	$\varepsilon_\gamma = \varepsilon_\alpha$	$\varepsilon_\gamma = \varepsilon_\alpha + \varepsilon_\beta$
	Apply only if: $\varepsilon_\gamma > 1$ for $d_{a\,min}$ and a_{max}	

PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

Addendum modification
Addendum modification (see diagram) is used to avoid undercut when the number of teeth is small, to increase tooth root strength and to achieve a specific distance between centers.

Gear pair with reference center distance
With a gear pair at reference center distance, the addendum modifications to the gear and pinion are equal but opposite, such that the distance between centers does not change. This design is preferred for crossed helical and bevel gear pairs.

Gear pair with modified center distance
The addendum modifications to the gear and pinion do not cancel each other, so the distance between centers changes.

Terms and errors
Terms and errors are explained in detail in the following standards: DIN 3960, DIN 58 405 (spur gears), DIN 3971 (bevel gears), and DIN 3975 (worms and worm gears).

Inserting $j_n = 0$ in the preceding formulas results in gearing with zero backlash. The tooth thickness and span measurement deviations required to produce backlash can be specified in accordance with DIN 3967 and DIN 58 405, taking into consideration the quality of the gears. It is necessary to ensure that the minimum backlash is great enough to compensate for tooth error (such as total composite error, alignment deviation, center-to-center distance error, etc.) without reducing the backlash to zero, and without leading to gear jamming. Other parameters for which the tolerances must be determined (with reference to the tooth errors given above in parentheses) are those for (two-flank) total composite error and (two-flank) tooth-to-tooth composite error

DIN gear qualities (DIN 3961 to DIN 3964)

Table 4. Manufacture and applications

Quality	Examples of applications	Manufacture
2	Primary-standard master gears	Form grinding (50 ... 60 % scrap rate)
3	Master gears for the inspection department	Form grinding and generative grinding
4	Master gears for the workshop, measuring mechanisms	
5	Drives for machine tools, turbines, measuring equipment	
6	As 5, also highest gears of passenger-car and bus transmissions	
7	Motor-vehicle transmissions (highest gears), machine tools, rail vehicles, hoisting and handling equipment, turbines, office machines	Non-hardened gears (carefully manufactured) by hobbing, generative shaping and planing (subsequent shaving is desirable); additional grinding is required for hardened gears
8 and 9	Motor-vehicle transmissions (middle and lower gears), rail vehicles, machine tools and office machines	Hobbing, generative shaping and planing (non-ground but hardened gears)
10	Transmissions for farm tractors, agricultural machinery, subordinate gear units in general mechanical equipment, hoisting equipment	All of the usual processes apply, plus extrusion and sintering, and injection molding for plastic gears
11 and 12	General agricultural machinery	

PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

(DIN 3963, DIN 58 405), alignment deviation (DIN 3962 T2, DIN 3967, DIN 58 405), and distance between centers (DIN 3964, DIN 58 405).

Module series for spur and bevel gears in mm (extract from DIN 780)

0.3	**1**	**3**	**10**	**32**
0.35	1.125	3.5	11	36
0.4	**1.25**	**4**	**12**	**40**
0.45	1.375	4.5	14	45
0.5	**1.5**	**5**	**16**	**50**
0.55	1.75	5.5	18	55
0.6	**2**	**6**	**20**	**60**
0.65	2.25	7	22	70
0.7	**2.5**	**8**	**25**	
0.75	2.75	9	28	
0.8				
0.85				
0.9				
0.95				

Modules given in bold type are preferred.

Addendum modification coefficient x

Table 5. Addendum modification coefficient x for straight-tooth gearing, $\alpha = 20°$,
Basic rack I, DIN 3972 ($h_{fP} = 1.167 \cdot m$)

1	2	3	4
Number of teeth z, for helical teeth z_i	Tooth free from undercut if $x \geq$	Top land width $0.2 \cdot m$ if $x \approx$	Tooth pointed if $x \geq$
7	+ 0.47	–	+ 0.49
8	+ 0.45	–	+ 0.56
9	+ 0.4	+ 0.4	+ 0.63
10	+ 0.35	+ 0.45	+ 0.70
11	+ 0.3	+ 0.5	+ 0.76
12	+ 0.25	+ 0.56	+ 0.82
13	+ 0.2	+ 0.62	+ 0.87
14	+ 0.15	+ 0.68	+ 0.93
15	+ 0.1	+ 0.72	+ 0.98
16	0	+ 0.76	+ 1.03

If the number of teeth is greater than $z = 16$, x should not be less than
$x = (16 - z)/17$, for helical teeth
$x = (16 - z_i)/17$.

Starter-tooth design

The system of tolerances for gears "Standard distance between centers" as specified in DIN 3961, which is customary in mechanical engineering and by which the required backlash is produced by negative tooth thickness tolerances, cannot be used for starters. Starter gear teeth require far more backlash than constant-mesh gears due to the starter engagement process. Such backlash is best achieved by increasing the distance between centers.

The high torque required for starting necessitates a high transmission ratio (i = 10 to 20). For this reason, the starter pinion has a small number of teeth (z = 8...12). The pinion generally has positive addendum modification. In the case of pitch gears, this addendum modification is expressed using the following notation outside Germany: Number of teeth, for instance = 9/10.

This means that only 9 teeth are cut on a gear blank with a diameter for 10 teeth; this corresponds to an addendum modification coefficient of + 0.5. Slight deviations of $x = + 0.5$ are quite common and do not affect the above-mentioned notation method: Number of teeth = 9/10. (This notation is not to be confused with the P 8/10 notation, see below and next page.)

Table 6. Customary starter gearing

Module m mm	Diametral pitch P 1/inch	Pressure angle of basic rack	American standard	European standard
2.1167	12	12°	SAE J 543 c	ISO 8123 1991 E
2.5	–	15°		ISO 9457–1 1991 E
2.54	10	20°	SAE J 543 c	ISO 8123 1991 E
3	–	15°		ISO 9457–1 1991 E
3.175	8	20°	SAE J 543 c	ISO 8123 1991 E
3.5	–	15°		ISO 9457–1 1991 E
4.233	6	20°	SAE J 543 c	ISO 8123 1991 E

PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

American gear standards

Instead of the module, in the USA standardization is based on the number of teeth on 1 inch of pitch diameter = diametral pitch (P).

$P = z/d$

The conversion is as follows:
Module m = 25.4 mm/P

The tooth spacing in the pitch circle is called circular pitch (CP):

$$CP = \frac{1 \text{ inch}}{P} \cdot \pi$$

Pitch p = 25.4 mm · $CP = \pi \cdot m$

Full-depth teeth

Full-depth teeth have an addendum $h_a = m$ in German standards, however the dedendum is frequently somewhat different.

Stub teeth

The formulas are the same as for full-depth teeth, however calculation of the addendum is based on a different module from that used for the other dimensions. Notation (example):

P 5/7 — P 7 for calculation of the addendum
P 5 for calculation of all other dimensions

Notation and conversions:

Outside Diameter:
$OD = da$

Pitch Diameter:
$PD = z/P = d$ in inches

Root Diameter:
$RD = d_f$

Layout Diameter:
$LD = (z + 2x)/P$ in inches
$LD = (z + 2x) \cdot$ m in mm
$LD \approx d_w$
d_w Pitch circle diameter

Measurement over D_M-pins:
M_d = measurement over D_M pins

Table 7. Diametral pitches P and modules derived from them

Diametral pitch P 1/inch	Corresponds to module m mm	Diametral pitch P 1/inch	Corresponds to module m mm	Diametral pitch P 1/inch	Corresponds to module m mm
20	1.27000	**6**	4.23333	**2**	12.70000
18	1.41111	5.5	4.61818	1.75	14.51429
16	1.58750	**5**	5.08000	**1.5**	16.93333
14	1.81429	4.5	5.64444	**1.25**	20.32000
12	2 11667	**4**	6.35000	**1**	25.40000
11	2.30909	3.5	7.25714	0.875	29.02857
10	2.54000	**3**	8.46667	**0.75**	33.86667
9	2.82222	2.75	9.23636	**0.625**	40.64000
8	3.17500	**2.5**	10.16000	**0.5**	50.80000
7	3.62857	2.25	11.28889		

PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

Calculation of load-bearing capacity

The following can be employed for rough estimations as an alternative to DIN 3990 "Calculation of load-bearing capacity of spur bevel gears". It applies to 2-gear pairs in a stationary transmission unit.
The quantities which appear in the following formulas are to be entered for the units given below:

Term		Unit	Term		Unit
P	Power	kW	φ	Life factor	–
P_{PS}	Metric horsepower	PS*	HB	Brinell hardness	
M	Torque	N · m	HRC	Rockwell hardness	
n	Rotational speed	min^{-1}	b_N	Effective flank width	mm
F	Peripheral force	N	k	Contact pressure	N/mm^2
u	Gear ratio	–	L_h	Service life	h

Table 8. Calculating load-bearing capacity of spur gears

Power	$P = 0.736 \cdot P_{PS}$ $P = M \cdot n/9549$ $P = F_t \cdot d \cdot n/(19.1 \cdot 10^6)$
Peripheral force in pitch circle	$F_{tw} = 2000 \cdot M/d_w = 19.1 \cdot 10^6\, P/(d_w \cdot n)$
in reference circle	$F_t = 2000 \cdot M/d = 19.1 \cdot 10^6\, P/(d \cdot n)$
Gear ratio	$u = z_2/z_1 = n_1/n_2$ for pairs of internal gears $u < -1$
Life factor	$\varphi = \sqrt[6]{5000/L_h}$ or from the table on P. 303
Contact pressure for the small gear	$k_{perm} = \frac{(HB)^2}{2560 \cdot \sqrt[6]{n}} = \frac{(HRC)^2}{23.1 \cdot \sqrt[6]{n}}$ or from the table on P. 303
Straight teeth	$k_{actual} = \frac{F_{tw}}{b_N \cdot d_{w1}} \cdot \frac{4\,(u+1)}{u \cdot \sin 2\,\alpha_w}$
Helical teeth	$k_{actual} = \frac{F_{tw}}{b_N \cdot d_{w1}} \cdot \frac{4\,(u+1) \cdot \cos^2\beta}{u \sin 2\,\alpha_{wt}}$ $\cos^2\beta$ only for full contact, otherwise = 1
Resistance to pitting	$S_w = \varphi \cdot k_{perm}/k_{IST} \geq 1$

Resistance to pitting and wear due to excessive contact pressure is provided if the equations for S_w for the smaller gear (gear 1) yield a value equal to or greater than 1. In the case of gear pairs where $z_1 < 20$, select $S_w \geq 1.2 \ldots 1.5$ on account of the greater contact pressure at the inside single engagement point. Because the contact pressure k is equal in magnitude for both gears of the pair, for k_{perm} a material can be selected for gear 2 from the table which has at least the same contact pressure as for gear 1 at rotational speed n_2.

PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

The k_{perm}-values in the table below apply when both gears are made of steel. For cast iron on steel or bronze on steel, the values should be roughly 1.5 times higher; for cast iron on cast iron or bronze on bronze, they should be approximately 1.8 times higher. For the gear with non-hardened surfaces, 20 % higher k_{perm} values are permissible if the other gear in the pair has hardened tooth flanks. The values in the table apply to a service life L_h of 5000 hours. A different service life L_h is allowed for in the equation for resistance to pitting S_w by means of the life factor φ.

Table 9. Permissible contact pressure k_{perm} in N/mm² for a service life L_h = 5000 h

Hardness of teeth		Rotational speed min⁻¹ (for 1 load change per revolution)											
HB	HRC	10	25	50	100	250	500	750	1000	1500	2500	5000	10000
90		2.2	1.9	1.7	1.5	1.3	1.1	1.05	1.0	0.94	0.86	0.77	0.68
100		2.7	2.3	2.0	1.8	1.6	1.4	1.3	1.2	1.15	1.06	0.94	0.84
120		3.8	3.3	2.9	2.6	2.2	2.0	1.9	1.8	1.66	1.53	1.36	1.21
140		5.2	4.5	4.0	3.6	3.0	2.7	2.5	2.4	2.26	2.08	1.85	1.65
170		7.7	6.6	5.9	5.2	4.5	4.0	3.75	3.6	3.34	3.06	2.73	2.43
200		10.7	9.1	8.1	7.3	6.2	5.6	5.2	4.9	4.6	4.24	3.78	3.37
230		14.1	12.1	10.8	9.6	8.2	7.3	6.9	6.5	6.1	5.61	5.0	4.45
260		18.0	15.4	13.8	12.2	10.5	9.4	8.8	8.4	7.8	7.17	6.39	5.69
280		20.9	17.9	16.0	14.2	12.2	10.9	10.2	9.7	9.0	8.31	7.41	6.6
300		24.0	20.6	18.3	16.3	14.0	12.5	11.7	11.1	10.4	9.54	8.5	7.6
330		29.0	24.9	22.2	19.8	17.0	15.1	14.1	13.5	12.6	11.6	10.3	9.2
400		42.6	36.6	32.6	29.0	24.9	22.2	20.7	19.8	18.5	17.0	15.1	13.5
	57	96.0	82.3	73.3	65.3	56.0	49.9	46.7	44.5	41.6	38.2	34.0	30.3
	≥ 62	112	96.5	86.0	76.6	65.8	58.6	54.8	52.2	48.8	44.8	39.9	35.6

Corresponding materials are given in the table "Strength values" on P. 305.

Table 10. Life factor φ

The life factor is used to convert the values in the table (referred to a service life L_h of 5000 hours) to values which correspond to a different service life period.

Service life in hours of operation L_h	10	50	150	312	625	1200	2500	5000	10,000	40,000	80,000	150,000
Life factor φ	2.82	2.15	1.79	1.59	1.41	1.27	1.12	1	0.89	0.71	0.83	0.57

Guidelines for selection of service life: Drives in continuous operation at full load: 40,000...150,000 hours of operation; drives run intermittently at full load or only intermittently: 50...5000 hours of operation.

PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

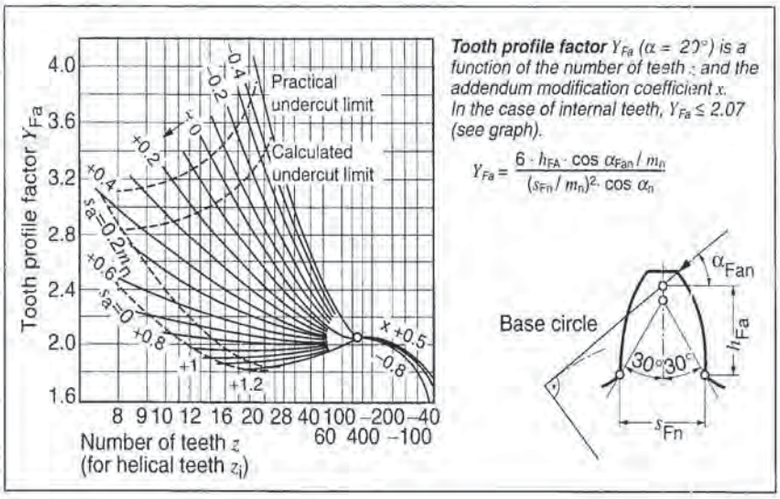
Teeth calculations for bending and tooth fracture

Table 11. Formulae for calculating the strength of spur gears

Peripheral velocity $v_1 = v_2$ m/s	$v_1 = \frac{\pi \cdot d_1 \cdot n_1}{60{,}000}$ d_1 in mm, n_1 in min^{-1}	
Velocity factor f_v	Take from table below, or calculate.	
Perm. root stress $\sigma_{F\,perm}$ N/mm²	Take $\sigma_{F\,lim}$ and Y_{NT} from table P. 305; estimate intermediate values. $\sigma_{F\,perm} = \sigma_{F\,lim} \cdot Y_{NT} \cdot Y_L$ $Y_L = 1$ for pulsating loads $Y_L = 0.7$ for alternating loads.	
Tooth profile factor Y_{Fa}	Take from graph below.	
Root stress of tooth $\sigma_{F\,actual}$ N/mm²	For straight teeth $\sigma_{F\,actual} = \frac{F_t}{b \cdot m} \cdot \frac{Y_{Fa}}{f_v \cdot \varepsilon_\alpha}$ ε_α, see P. 298 m, m_n, b in mm, F_t in N see P. 302	For helical teeth $\sigma_{F\,actual} = \frac{F_t}{b \cdot m_n} \cdot \frac{Y_{Fa}}{f_v \cdot \varepsilon_\alpha} \left(1 - \frac{\varepsilon_\beta \cdot \beta}{120°}\right)$ where $(1 - \varepsilon_\beta \cdot \beta/120°)$ must be ≥ 0.75. ε_β see P. 298, β in °
Resistance to tooth fracture S_F	$S_F = \sigma_{F\,perm}/\sigma_{F\,actual} \geq 1$	

Resistance to tooth fracture is provided if the equations for S_F for the smaller gear (gear 1) yield a value greater than or equal to 1.

If a better material is selected for gear 1 than for gear 2, the calculation for bending must also be made for gear 2.



Tooth profile factor Y_{Fa} *(α = 20°) is a function of the number of teeth z and the addendum modification coefficient x. In the case of internal teeth, $Y_{Fa} \leq 2.07$ (see graph).*

$$Y_{Fa} = \frac{6 \cdot h_{FA} \cdot \cos \alpha_{Fan} / m_n}{(s_{Fn}/m_n)^2 \cdot \cos \alpha_n}$$

Table 12. Velocity factor f_v

Materials	Peripheral velocity v m/s 0.25 | 0.5 | 1 | 2 | 3 | 5 | 10	Basic equation
Steel and other metals	Velocity factor f_v 0.96 | 0.93 | 0.86 | 0.75 | 0.67 | 0.55 | 0.38	$f_v = \frac{A}{A + v}$ [1]
Fabric-base laminates and other non-metals	0.85 | 0.75 | 0.62 | 0.5 | 0.44 | 0.37 | 0.32	$f_v = \frac{0.75}{1 + v} + 0.25$

PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

Gear materials

Table 13. Material parameters

	Material	Condition	Tensile strength R_m N/mm² min.	Fatigue strength for reversed bending stress σ_{bW} N/mm² min.	Hardness HB or HRC min.	Permissible root stress[2] $\sigma_{F\,lim}$ N/mm²	Life factor Y_{NT}[3] at number of load changes $N_L = L_h \cdot 60 \cdot n$ ≥ 3 · 10⁶	10⁶	10⁵	10⁴	10³
Heat-treatable steel	St 60-2, C 45	annealed	590	255	170 HB	160					
		heat-treat.	685	295	200 HB	185					
		heat-treat.	980	410	280 HB	245					
	St 70-2, C 60	annealed	685	295	200 HB	185					
		heat-treat.	785	335	230 HB	209					
		heat-treat.	980	410	280 HB	245	1	1.25	1.75	2.5	2.5
	50 Cr V 4	annealed	685	335	200 HB	185					
		heat-treat.	1130	550	330 HB	294					
		heat-treat.	1370	665	400 HB	344					
	37 Mn Si 5	annealed	590	285	170 HB	160					
		heat-treat.	785	355	200 HB	230					
		heat-treat.	1030	490	300 HB	270					
Case-hardening steel	RSt 34-2, C 15	annealed	335	175	100 HB	110	1	1.25	1.75	2.5	2.5
		surf.-hard.[4]	590	255	57 HRC	160	1	1.2	1.5	1.9	2.5
	16 Mn Cr 5	annealed	800	–	150 HB	–	–	–	–	–	–
		surf.-hard.[4]	1100	–	57 HRC	300	1	1.2	1.5	1.9	2.5
	20 Mn Cr 5	annealed	590	275	170 HB	172	–	–	–	–	–
		surf.-hard.[4]	1180	590	57 HRC	330	1	1.2	1.5	1.9	2.5
	18 Cr Ni 8	annealed	640	315	190 HB	200	–	–	–	–	–
		surf.-hard.[4]	1370	590	57 HRC	370	1	1.2	1.5	1.9	2.5
Gray iron	GG-18	–	175		200 HB	50	1	1.1	1.25	1.4	1.6
	GS-52.1	–	510		140 HB	110	1	1.25	1.75	2.5	2.5
	G-SnBz 14		195		90 HB	100	1	1.25	1.75	2.5	2.5
Polyamide PA 66 Pa 66 + 30 % GF		at 60 °C	– 140	40 43	– –	27 29	1	1.2	1.75	–	–
Fabric-base fine laminate coarse			– –	– –	– –	75 50	1 1	1.15 1.2	1.4 1.6	1.65 2.1	2.0 2.8

For the heat-treatable steels, two values are given in each case for the heat-treated condition. For smaller gears up to roughly module 3, the larger of the two values can be specified. With very large gears, however, only the smaller value can be achieved with certainty.

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1) Values are valid for $A = 6$ (average tooth quality). With cast gears and high-precision gears, $A = 3$ and 10 respectively.
2) For pulsating loads with a material fatigue strength $N_L \geq 3 \cdot 10^6$. In the case of alternating loads (id er gears), take into account the alternating load factor Y_L.
3) Within fatigue strength of finite life, root bending stress is multiplied by the factor Y_{NT}, depending upon the number of load changes $N_L = L_h \cdot 60 \cdot n$.
4) Surface-hardened.

PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

Belt drives

Friction belt drives

Units and symbols

Quantity		Unit
A	Belt cross section	mm²
F	Contact force	N
F_1	Belt force on load side	N
F_2	Belt force on slack side	N
F_c	Centrifugal force of belt	N
F_f	Centrifugal force per side	N
F_R	Frictional force	N
F_u	Peripheral force	N
F_v	Pretensioning force	N
F_w	Tensioning force of shaft	N
P	Required power transmission	kW
k_1	Pretension factor, with reference to operating conditions and wrap angle	–
k_2	Factor for centrifugal force	–
v	Belt speed	m/s
z	Number of belts (V-belt drives) or ribs (V-ribbed belts)	–
α	Groove angle	°
β	Wrap angle	°
μ	Coefficient of friction	–
μ'	Wedge coefficient of friction	–
ϱ	Mean density of belt material	g/cm³

Transmission of force

The general equation for friction

$$F_R = \mu \cdot F$$

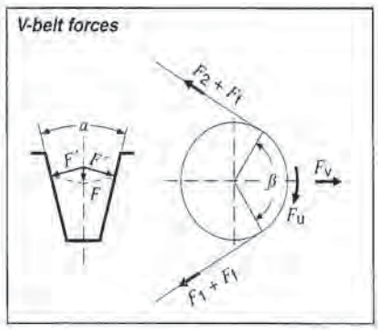
yields the following equation for V-belt pulleys (see illustration)

$$F_R = \mu \cdot 2 \cdot F'$$

or $F_R = \mu' \cdot F$

where $\mu' = \mu/\sin(\alpha/2)$

The Eytelwein equation describes the transition from static friction to sliding friction:



$$F_1/F_2 = e^{\mu'\beta}$$

where $\mu' = 0.5 \pm 0.15$

according to the specifications of the V-belt manufacturers with the incorporation of various safety factors.

As long as the ratio of forces on the two belt sides is

$$F_1/F_2 \le e^{\mu'\beta}$$

the belt will not slip during transmission of the peripheral force

$$F_u = F_1 - F_2 = P \cdot 1020/v$$

A pretensioning force F_v is required in order to transmit the peripheral force F_u; at high rotational speeds, the centrifugal force constituent F_F of the belt must be taken into consideration. The pretensioning force is

$$F_v = F_w \cdot F_F$$

where $F_w = F_u \cdot [(e^{\mu'\beta}+1)/(e^{\mu'\beta}-1)] \cdot \sin(\beta/2)$

$$F_F = 2 \cdot z \cdot F_f \cdot \sin(\beta/2)$$

$$F_f = \varrho \cdot A \cdot v^2 \cdot 10^{-3}$$

or in simplified form

$$F_F = 2 \cdot z \cdot k_2 \cdot v^2 \cdot \sin(\beta/2)$$

In practice, the following approximation process is frequently sufficient for calculating the pretensioning force (according to the Continental Co.):

$$F_v = (k_1 \cdot F_u + 2 \cdot z \cdot k_2 \cdot v^2) \cdot \sin(\beta/2)$$

The following rule of thumb can often be used for drives with 2 pulleys:

$$F_v = (1.5 \ldots 2) \cdot F_u$$

In order to check belt pretensioning, the static force applied to each side of each belt

$$F_s = F_v/[2z \cdot \sin(\beta/2)]$$

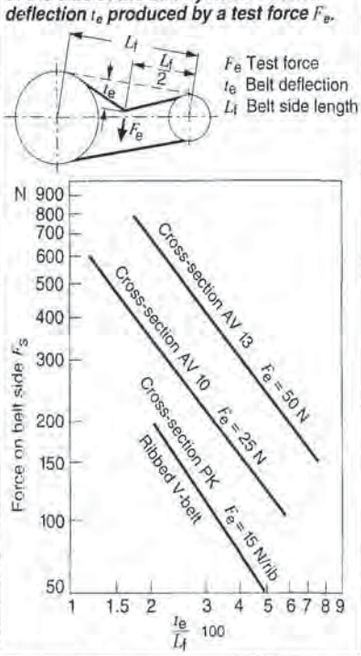
is compared with the value obtained by measuring belt deflection.

Belt drive calculations are performed in accordance with DIN 2218, or in accordance with the specifications supplied by the belt manufacturers. Manufacturer's belt performance specifications are based on a theoretical service life of 25,000 h. Computer programs are used to perform service life calculations. In properly designed drives, belt creep is less than 1%, and belt efficiency is 94...97%.

Belt service life is shortened if the maximum permissible belt speed and bending frequency are exceeded, if the belt pulleys are smaller than the minimum permissible pulley diameter or, in the case of V-belts, belt tensioners are used to apply force to the back of the belt.

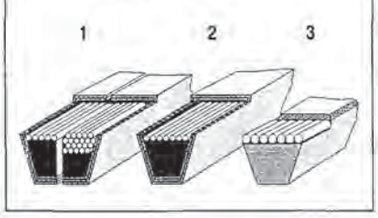
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US PATENT NO. 6,581,565

Determination of the static force F_S of the side of the belt by means of belt deflection t_e produced by a test force F_e.



Standard V-belts (classical V-belts)
In accordance with DIN 2215, for domestic appliances, agricultural machinery, heavy machinery. Ratio of top width to height: 1.6 : 1. Belt designs which use corded cable and bundled fibers as the tension members transmit considerably less power than equally wide narrow V-belts. Due to their high tensile strength and transverse stiffness, they are suitable for use under rough operating conditions with sudden load changes.

V-belt types
1 Wrapped standard V-belt
2 Wrapped narrow V-belt
3 Raw-edge narrow V-belt



Pretension factor k_1				
Belt	β	Drive: Light constant loads	Moderate loads	Heavy shock loads
V-belts	180°	1.5	1.7	1.9
	90°	2.6	2.8	3.0
V-ribbed belt	180°	1.8	2.0	2.2
	90°	3.3	3.5	3.7

Intermediate values for other wrap angles may be interpolated.

Belt	Cross section	Centrifugal factor k_2
V-belt	SPZ	0.07
	SPA	0.12
V-ribbed belt	K	0.02

Belt velocities up to 30 m/s and bending frequencies up to 40 s^{-1} are permissible. See DIN 2217 for corresponding pulley dimensions.

Narrow V-belts
In accordance with DIN 7753, Part 1; machine construction, motor vehicles built in the 1960s and 1970s. Ratio of top width to height: 1.2:1. The narrow V-belt is a modified version of the standard V-belt in which the central section, which in any case transmitted only limited forces, has been omitted. Higher capacity than standard V-belts of equal width. Toothed version exhibits less creep when bending around small pulleys. Belt velocities up to 42 m/s and bending frequencies up to 100 s^{-1} are permissible. (See DIN 2211 for corresponding pulley dimensions).

Raw-edge V-belts
Raw-edge standard V-belts in accordance with DIN 2215, and raw-edge narrow V-belts for motor vehicles in accordance with DIN 7753, Part 3 (draft). Subsurface belt fibers perpendicular to the direction of belt motion provide a high degree of flexibility while at the same time giving extreme transverse stiffness and high wear resistance. They also provide excellent support for the specially treated tension member. Particularly with small-diameter pulleys, this design increases belt capacity and provides a longer service life than wrapped narrow V-belts.

PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

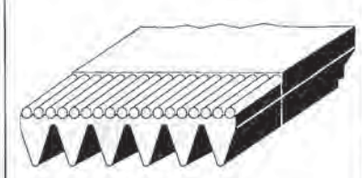
Further developments

The latest development are V-belts with tension members made of Kevlar. Kevlar has very high tensile strength with a very small degree of elongation, and is more highly temperature-resistant.

V-ribbed belts (poly-grooved belts)

In accordance with DIN 7867. Very flexible; the back of the belt may also be used to transmit power. This capability makes it possible for one such belt to be used to drive several vehicle accessories simultaneously (alternator, fan, water pump, air-conditioner compressor, power-steering pump, etc.) if the wrap angle around each driven pulley is sufficiently large. Optional cross sections include the PH, PJ, PK, PL, and PM, of which the PK cross section has been widely used in motor vehicles in recent years. This allows use of narrower pulley diameters ($d_{b\,min} \approx 45$ mm) than those possible with narrow V-belts (cross-section: AVX 10). A pretension approximately 20% greater than that employed on a narrow V-belt is recommended to provide the same force-transmission capabilities.

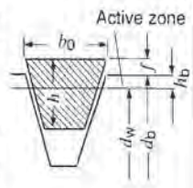
V-ribbed belts *(poly-grooved belts)*



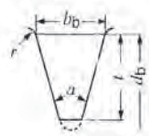
Narrow V-belts for motor vehicles in accordance with DIN 7753 Part 3
Example of designation: narrow V-belt DIN 7753 AVX 10 × 750 La,
Raw-edge, toothed with belt cross-section code AVX 10 L_a = 750 mm.

Narrow V-belt		Wrapped		Raw-edge Solid cross section (V)		Toothed (G)	
Belt cross-section	Code	9.5	12.5	AVP10	AVP13	AVX10	AVX13
	ISO code	AV 10	AV 13	AV 10	AV 13	AV 10	AV 13
Top width	$b_0 \approx$	10	13	10	13	10	13
Belt height	$h \approx$	8	10	7.5	8.5	8	9
Eff. line differential	h_b	1.8	2.6	0.9			
Belt runout	$h_{a\,max}$	–		2.4			
Effective length	L_a	500 to 2550: in increments of 25 mm					

Dimensions in mm



Groove cross-section code		AV 10		AV 13	
Effective diameter	d_b	< 57	≥ 57	< 70	≥ 70
Groove angle	$\alpha \pm 0.5°$	34°	36°	34°	36°
Effective width	b_b	9.7		12.7	
Groove depth	t_{min}	11		14	
Radius of curvature	r	0.8			
Distance bet. grooves	e_{min}	12.6		16	



PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

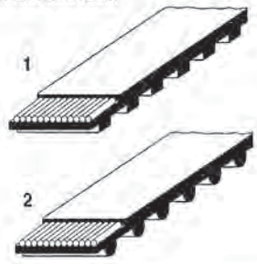
Positive belt drives

Synchronous drive belts (toothed belts) in accordance with DIN/ISO 5296
Toothed belts are used in motor vehicles as camshaft drives, and in some cases as ignition distributor drives.

Synchronous drive belts with trapezoidal or rounded teeth combine the advantages of a belt drive (any desired distance between pulley centers, quiet operation, low maintenance) with the advantages of a positive transmission (synchronous operation, low bearing stress due to low shaft load).

Synchronous drive belts must be guided on both sides to prevent them from running off. This is accomplished by using either a toothed pulley with two flanges or two toothed pulleys with one flange each on opposite sides.
See DIN/ISO 5294 for corresponding toothed-pulley dimensions.

Synchronous drive belts (toothed belts)
1 with trapezoidal teeth
2 with rounded teeth



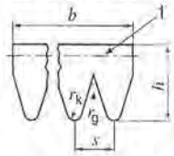
V-ribbed belts and pulleys as per DIN 7867

Designation for a 6-ribbed V-belt, cross-section code PK and reference length 800 mm:
V-ribbed belt DIN 7867-6 PK 800.

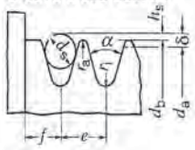
Designation for the corresponding V-belt pulley with 90 mm reference diameter:
V-ribbed belt pulley DIN 7867-6 Kx 90

Dimensions in mm	Belt	Groove profile
Cross-section code	PK	K
Rib or groove spacing s or e	3.56	3.56
Perm. deviation tolerance for s or e	± 0.2	± 0.05
Sum of deviation tolerances s or e	± 0.4	± 0.30
Groove angle α		40° ± 0.5°
r_k at rib / r_a at groove head	0.50	0.25
r_g at rib / r_i at groove seat	0.25	0.50
Belt height h	6	
Nominal dia. of test pin d_s		2.50
$2 \cdot h_s$ nominal dimension		0.99
$2 \cdot \delta$ (see illustration)		2.06
f (see illustration)		2.5

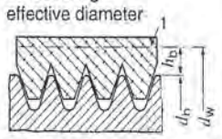
Belt cross-section



Groove cross-section



Determining effective diameter



1 Position of tension member

Belt width $b = n \cdot s$
with number of ribs = n

Effective diameter for K cross-section $d_w = d_b + 2\,h_b$
$h_b = 1.6$ mm

PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

Basic principles of threaded fasteners

Units and symbols

Quantity		Unit
A_s	Stress area of thread, see P. 312	mm²
D_{Km}	Effective diameter for determining the friction torque at the fastener head or nut bearing surface	mm
F_M	Installed tension force (gen.)	N
F_{sp}	(Axial) installed tension force of threaded fastener at 90% of yield strength	N
M_M	Tightening torque for mounting	N · m
M_{sp}	Tightening torque required to load a threaded fastener to F_{sp}	N · m
P	Thread pitch	mm
d	Fastener diameter (major diameter of the thread)	mm
d_2	Mean flank diameter of the thread	mm
d_3	Root diameter of the thread	mm
μ_G	Coefficient of thread friction (friction factor)	–
μ_{tot}	Mean coefficient of friction for the fastener and head or nut bearing faces	–
μ_K	Coefficient of friction for the head and nut bearing faces	–
$\sigma_{0.2}$	0.2% yield strength of the fastener material (for low-strength fasteners, the yield point σ_s applies instead; see footnote[1]) on P. 43)	N/mm²

Basic rules

Threaded fasteners (screws, bolts, nuts) should clamp parts together with a force great enough to hold them in contact with one another without dislocation in spite of all applied operating forces. Bolts must not be subjected to shear.

A correctly preloaded fastener needs itself only to absorb a fraction of those operating forces which are applied roughly parallel to the bolt axis and which are usually dynamic in nature. When the parts to be joined are themselves as rigid as possible and elastic fasteners are used (viz. long and reduced in the body to d_3 and of property classes 8.8...12.9 with optimum tension force), the lowest fastener fatigue loading is achieved. Highly stressed joints are also the best provision against subsequent working loose (setting of the joint after tightening). As a rule of thumb, setting of the joint reduces the installed tension force of a joint using 8.8 fasteners by 10...20% if the maximum temperature is 100 °C and the parts so joined are metals with a tensile strength of at least 300 N/mm².

The tensioned section of the threaded fastener should have a minimum threaded length of 0.5 d, the object being to achieve 1.0 d whenever possible.

The spring force of installed locking devices (spring lock washers and the like) should be as great as the tension force of the fastener! Spring lock washers are usually disadvantageous when used with screws of property classes ≥ 8.8, grip lengths > 2.5 d and metallic components (because $F \ll F_{sp}$, and joint setting in the bearing faces additionally occurs). In the case of 4.8 and 5.6/5.8 fasteners, spring lock washers are generally superfluous if grip lengths are ≥ 5 d.

The compressive load per unit area under fastener heads and nuts must not exceed the crushing yield point (corresponds to at least the 0.2% permanent elongation limit or yield strength) of the material of the joint parts (if required, use large washers or flanged hex screws).

Flange mountings using four bolts are considerably more reliable than such joints which use 3 bolts in a 120° arrangement (in the latter case, failure of one bolt leads to failure of the entire bolted joint).

Property classes

The designation system for threaded fasteners consists of two numbers separated by a decimal point. The first number represents ≈ 1/100 of the minimum tensile strength in N/mm², while the second number is 10 times the ratio of the minimum yield point (or 0.2% yield strength) to the minimum tensile strength.

The property classes of standard nuts (with nominal heights ≥ 0.8 d and width across flats ≥ 1.45 d) are indicated by a number which corresponds to 1/100 of the stress under proof load in N/mm². This stress corresponds to the minimum tensile strength of a fastener in the same property class. The ability of a nut to with-

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US PATENT NO. 6,581,565

Table 1. Threaded fasteners: Mechanical strength properties and designations
Fasteners made of steels alloyed with boron which have a low carbon content are indicated by a solid line below the numerical designation, e.g. for property class: 10.9 (see also DIN-EN 20 898, Part 1).

Property class		3.6	4.6	4.8	5.6	5.8	6.8	8.8	10.9	12.9
Tensile strength σ_B N/mm²	min	330	400	420	500	520	600	800	1040	1220
	max =	490	550		700		800	1000	1200	1400
Yield point R_{el} N/mm²	min	180	240	320	300	400	480	–	–	–
0.2% yield strength $\sigma_{0.2}$	min	–	–	–	–	–	–	640	900	1080
Vickers hardness HV30	min	95	120	130	155	160	190	250	320	385
	max	250						320	380	435

stand the stress under proof load is determined using a bolt thread of greater strength (min. 45 HRC); after the stress is relieved, the nut must still be movable on the thread.

Property classes for nuts:

4	5	6	8	10	12

For nuts with limited loadability:

04	05

(e.g. flat nuts)

For thread diameters of 5 mm and larger,

- Fasteners in property class 8.8 and greater, as well as
- Nuts of all property classes

must be marked with the property class on the fastener head or nut face. If the fasteners are small, the decimal point between the two numbers can be omitted; for nuts, a code system can also be used, whereby the chamfers of the nuts are marked with notches and dots.

The fatigue limit of cold-formed steel fasteners of sizes up to M12 is $\sigma_D = \sigma_m \pm 50 \ldots 60$ N/mm² (referred to the root cross-section), and applies up to a mean stress of $\sigma_m = 0.6 \cdot \sigma_{0.2}$ and for all property classes from 4.8 to 12.9. Fasteners of sizes up to M8 whose threads are rolled after hardening and tempering can achieve σ_D values up to 450 ± 100 N/mm², whereas gray cast iron and aluminum nuts and fasteners can achieve values of ± 120 N/mm².

Table 2. Yield strength for threaded fasteners made of other materials

Material	Designation	Thread	0.2% yield strength N/mm²
X5CrNi 1911	A2-70	≤ M20	450
CuZn37	Cu2	≤ M6	340
		M8 ... M39	250
CuSn6	Cu4	≤ M12	340
		M14 ... M39	200
AlMgSi1	Al3	≤ M39	250

PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

Tightening of threaded fasteners

Sufficient tightening torque for threaded fasteners can be realized by hand using a screwdriver up to sizes M5 – 8.8, using a hexagon socket up to a maximum of M8 – 10.9, and using a ring wrench or box-end wrench up to a maximum of M12 – 10.9 (provided that $\mu = \mu_G = \mu_K = 0.14$).

When fasteners are tightened using (conventional) torque measurement, variations in tension force occur in a ratio of approx. 1:2, and are caused by

- Variations in the coefficients of friction μ_K and μ_G (for screw head or nut and thread contact surface),
- Variations in the tightening torque (when tightened by hand and using power screwdrivers).

For $\mu_G = \mu_K = 0.1$, the effective force for producing the tension force is 17 % of the complete tightening torque, whereas the corresponding figure for $\mu_G = \mu_K = 0.2$ is only 9.5% of the complete tightening torque.

The highest possible tension forces with very small variations can be achieved with threaded fasteners using the angular tightening method and the yield-point-controlled tightening method. It must be noted that these procedures – employed to greatest advantages with grades 8.8 and above – necessitate the use of pre-calculations and empirical testing and/or special electronically-controlled wrenches. In addition, if the yield-point-controlled tightening method is used, the grip (total thickness of the bolted parts without female thread) should be at least as thick as the nominal diameter of the bolt which should be ≥ M4.

Basic formulas for tightening with torque measurement

The maximum permissible installed tension force in the fastener is limited by the effective stress (tension and torsion) which is equal to 0.9 times the minimum yield strength; this installed tension force is designated F_{sp}. The effective stress for the stress area is calculated according to the formula

$$A_s = \frac{\pi}{4} \cdot \left(\frac{d_2 + d_3}{2}\right)^2$$

with which the static, axial tensile stresses and the torsional stresses (generated by thread friction) can be viewed as if the fastener were a smooth component (A_s is greater than the root cross-section due to the fact that the threads inhibit deformation). Thus

$$F_{sp} = \frac{0.196\,(d_2 + d_3)^2 \cdot 0.9 \cdot \sigma_{0.2}}{\sqrt{1 + 4.86\left(\frac{P + \mu_G \cdot 3.63\,d_2}{d_2 + d_3}\right)^2}}$$

The values given in Table 5 for the installed tension forces F_{sp} were calculated for various coefficients of thread friction μ_G using the above formula.

In determining tightening torques it was formerly standard practice to assume that $\mu_G = \mu_K = \mu_{tot}$ in order to arrive at the simplified formula

$$M_{sp} = F_{sp}\left[0.16\,P + \mu_{tot}\left(0.58\,d_2 + \frac{D_{Km}}{2}\right)\right]$$

and corresponding graphs. In the case of differing fastener head and thread coefficients of friction (see example at end of chapter), however, this formula cannot be used to correctly determine the necessary tightening torque (see also the explanation above Table 4).

A simple, practical way of calculating tightening torque in the case of differing coefficients of friction μ_G and μ_K is the K method (proposed by Gill, used in the USA and contained in VDI Recommendation 2230, 1977 edition):
According to this method, the installed tightening torque is

$$M_M = K \cdot F_M \cdot d$$

and for tightening to 90% of the yield strength

$$M_{sp} = K \cdot F_{sp} \cdot d$$

where

$$K = \frac{0.16 \cdot P + \mu_G \cdot 0.58 \cdot d_2 + \mu_K \cdot D_{Km}/2}{d}$$

For differing coefficients of friction μ_G and μ_K and a wide range of fastener dimensions, K can be given independently of d in tabular form (see Table 4).

PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

Coefficients of friction

Using Table 3 below, both the coefficient of friction for fastener head or nut (μ_K) and thread friction (μ_G) at the contact surface are determined separately and one after the other.

These coefficients of friction are significantly increased by the use of self-locking screws, washers with sharp roughly radial edges and lock nuts; connections using these fastening devices must be tightened according to special specifications.

Tightening threaded-fastener connections

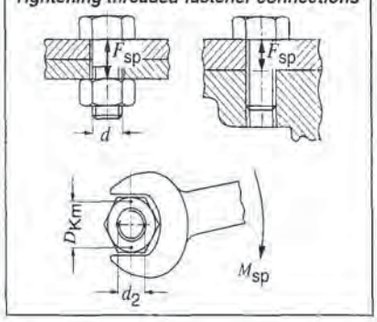


Table 3.
Coefficients of friction μ_K and μ_G for various surface and lubrication conditions
$\mu_{G\,min}$ (lower value in Table 3) is used to determine tension force F_{sp} in Table 5.

Surfaces on mating party: Bearing face on part (μ_K) or *nut thread* (μ_G)		Lubrication condition	Fastener surfaces: Bearing face of head or underside of nut (μ_K) or *fastener thread* (μ_G)		
			Steel, oil-quenched or Zn-phosph. pressed rolled	Steel, oil-quenched or Zn-phosph. turned cut	Steel zinc-plated 6 µm
Steel,	rolled	Lightly oiled	0.13...0.19	0.10...0.18	0.10...0.18
	turned, cut planed, milled		0.10...0.18	–	0.10...0.18
	ground		0.16...0.22	0.10...0.18	0.10...0.18
Gray cast iron	planed, milled, turned, cut		0.10...0.18	–	0.10...0.18
All-black	malleable iron, ground		0.16...0.22	0.10...0.18	0.10...0.18
Steel	cadmium-electroplated, 6 µm		0.08...0.16	0.08...0.16	–
	zinc-electroplated, 6 µm		0.10...0.18	0.10...0.16	0.16...0.20
	zinc-electropl. female threads		–	–	0.10...0.18
	ground, rolled, phosphated		0.12...0.20	–	–
	machined, phosphated		0.10...0.18	–	–
Al-Mg	alloy, processed, cut		0.08...0.20	–	–
Steel,	cadmium-electroplated, 6 µm	Dry	0.08...0.16	–	–
	cadm.-electropl., fem. threads		0.08...0.14	–	–
	zinc-electroplated, 6 µm		0.10...0.18	–	0.20...0.30
	zinc-electropl., female threads		0.08...0.16	–	0.12...0.20

PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

Table 4. K-Values

The K-values apply to standard threads of sizes M1.4 to M42 and bolt head and nut sizes corresponding to hexagon bolts and screws in accordance with DIN 931, 933; fillister-head slotted screws in accordance with DIN 84, hexagon cap screws in accordance with DIN 912, 6912, 7984; and nuts in accordance with DIN 934.

For M16 to M42 threads, the K-value is to be reduced by 5%, because the transition from M14 to M16 represents a significant jump in the coarseness of thread pitch a/P.

Intermediate values can be interpolated with sufficient accuracy.

Note: In special cases where only μ_{tot} has been determined previously, the K-value is determined for the case in which $\mu_K = \mu_G = \mu_{tot}$ (see also "Tightening of threaded fasteners" P. 304).

Coefficient of friction for threads μ_G	Coefficient of friction for fastener head and nut bearing face μ_K										
	0.04	0.06	0.08	0.10	0.12	0.14	0.16	0.18	0.20	0.24	0.28
0.08	0.094	0.108	0.120	0.134	0.148	0.162	0.176	0.190	0.204	0.232	0.260
0.10	0.104	0.118	0.132	0.146	0.158	0.172	0.186	0.200	0.214	0.242	0.270
0.12	0.114	0.128	0.142	0.156	0.170	0.184	0.196	0.210	0.224	0.252	0.280
0.14	0.124	0.138	0.152	0.166	0.180	0.194	0.208	0.222	0.234	0.262	0.290
0.16	0.134	0.148	0.162	0.176	0.190	0.204	0.218	0.232	0.246	0.272	0.300
0.18	0.146	0.160	0.172	0.186	0.200	0.214	0.228	0.242	0.256	2.284	0.312
0.20	0.156	0.170	0.184	0.198	0.210	0.224	0.238	0.252	0.266	0.294	0.322
0.24	0.176	0.190	0.204	0.218	0.232	0.246	0.260	0.274	0.286	0.314	0.342
0.28	0.198	0.212	0.224	0.238	0.252	0.266	0.280	0.294	0.308	0.336	0.362

Axial tension forces of bolted joints

Scope of Table 5 for installed tension forces F_{sp}

Table 5 applies only to

- Bolts with standard thread and $\frac{\text{shank diameter } d_1}{\text{minor diameter } d_3} \geq 1.05$
- With respect to material strength, the correct mating of nut and bolt, or female thread in housings;
- Nut height at least 0.8 x thread diameter and sufficient depth of thread in housings and the like.

Correction factors for tension forces and coefficients of friction (selection)

Flat nuts with a height of $0.5 \cdot d$ and the same hardness as the bolts allow a tension force of only 80 % of the values given in the table. In the case of low-strength flat nuts (04) on 12.9 bolts, only 33 % of F_{sp} is permissible.

In the case of fastener heads and nuts with smaller or larger bearing faces as well as adjusting screws and locating screws which do not have contact at their heads, the μ_K-value must be reduced or increased by the same ratio as the ratio of the mean friction diameter of their bearing faces to the mean friction diameter of the above-mentioned standard hexagon etc. fasteners and nuts. It is this corrected hypothetical μ_K-value which is used to determine the K-value (for this reason, Table 4 gives μ_K-values beginning with 0.04).

In the case of countersunk-head screws, allowance must be additionally made for the conical effect of the countersink angle as well as the greater elasticity of the outer edge of the head; thus the μ_K-value hypothetically increases by a factor of 1.25 in the case of standard countersunk screws (countersink angle: 90°).

PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

Table 5. Tension forces F_{sp} in 10^3 N for standard threads and coefficients of friction μ_G in the threads (for 90 % utilization of $\sigma_{0.2}$ or σ_s)

Dimension (pitch)	Property class	F_{sp} (10^3 N) for coefficients of friction μ_G in the threads									
		0.06	0.08	0.10	0.12	0.14	0.16	0.18	0.20	0.24	0.28
	4.8	2.3	2.2	2.1	2.0	1.9	1.9	1.8	1.7	1.6	1.4
	5.8	2.8	2.7	2.6	2.5	2.4	2.3	2.2	2.1	2.0	1.8
M 4	8.8	4.5	4.4	4.2	4.1	3.9	3.7	3.6	3.4	3.2	2.9
	10.9	6.4	6.2	5.9	5.7	5.5	5.3	5.0	4.8	4.4	4.1
(0.7)	12.9	7.7	7.4	7.1	6.9	6.6	6.3	6.0	5.8	5.3	4.9
	4.8	3.7	3.6	3.5	3.3	3.2	3.1	2.9	2.8	2.6	2.4
	5.8	4.6	4.5	4.3	4.2	4.0	3.8	3.6	3.5	3.2	3.0
M 5	8.8	7.4	7.2	6.9	6.6	6.4	6.1	5.9	5.6	5.2	4.8
	10.9	10.4	10.1	9.7	9.4	9.0	8.6	8.3	7.9	7.3	6.7
(0.8)	12.9	12.5	12.1	11.7	11.2	10.8	10.3	9.9	9.5	8.7	8.1
	4.8	5.2	5.1	4.9	4.7	4.5	4.3	4.1	4.0	3.7	3.4
	5.8	6.6	6.3	6.1	5.9	5.6	5.4	5.2	5.0	4.6	4.2
M 6	8.8	10.5	10.1	9.8	9.4	9.0	8.6	8.3	7.9	7.3	6.7
	10.9	14.7	14.2	13.7	13.2	12.7	12.1	11.7	11.2	10.3	9.5
(1.0)	12.9	17.7	17.1	16.5	15.8	15.2	14.6	14.0	13.4	12.3	11.4
	4.8	9.6	9.3	8.9	8.6	8.3	7.9	7.6	7.3	6.7	6.2
	5.8	12.0	11.6	11.2	10.8	10.3	9.9	9.5	9.1	8.4	7.7
M 8	8.8	19.2	18.6	17.9	17.2	16.5	15.7	15.2	14.6	13.4	12.4
	10.9	27.0	26.1	25.2	24.2	23.2	22.3	21.4	20.5	18.9	17.4
(1.25)	12.9	32.4	31.3	30.2	29.0	27.9	26.8	25.7	24.6	22.6	20.9
	4.8	15.3	14.8	14.2	13.7	13.2	12.6	12.1	11.6	10.7	9.9
	5.8	19.0	18.5	17.8	17.1	16.5	15.9	15.1	14.5	13.4	12.3
M 10	8.8	30.5	29.5	28.5	27.4	26.3	25.3	24.2	23.2	21.4	19.7
	10.9	42.9	41.5	40.1	38.5	37.0	35.5	34.1	32.7	30.1	27.7
(1.5)	12.9	51.5	49.8	48.1	46.2	44.4	42.6	40.1	39.2	36.1	33.2
	4.8	22.2	21.5	20.8	20.0	19.2	18.4	17.7	16.9	15.6	14.4
	5.8	27.8	26.9	25.9	25.0	24.0	23.0	22.1	21.2	19.5	18.0
M 12	8.8	44.5	43.0	41.5	40.0	38.4	36.8	35.3	33.9	31.2	28.7
	10.9	62.5	60.5	58.4	56.2	54.0	51.8	49.7	47.7	43.8	40.4
(1.75)	12.9	75.0	72.6	70.0	67.4	64.8	62.2	59.6	57.2	52.6	48.5
	4.8	30.5	29.6	28.5	27.4	26.4	25.3	24.3	23.3	21.4	19.7
	5.8	38.1	36.9	35.6	34.3	32.9	31.6	30.3	29.1	26.8	24.7
M 14	8.8	61.0	59.1	57.0	54.9	52.7	50.6	48.5	46.5	42.8	39.5
	10.9	85.8	83.1	80.1	77.1	74.1	71.2	68.3	65.5	60.2	55.5
(2.0)	12.9	103	99.7	96.2	92.6	89.0	85.4	81.9	78.5	72.3	66.6
	4.8	41.8	40.5	39.2	37.7	36.3	34.8	33.4	32.1	29.5	27.2
	5.8	52.3	50.7	48.9	47.2	45.4	43.6	41.8	40.1	36.9	34.0
M 16	8.8	83.6	81.1	78.3	75.5	72.3	69.7	66.9	64.2	59.0	54.4
	10.9	118	114	110	106	102	98.0	94.1	90.2	83.0	76.5
(2.0)	12.9	141	137	132	127	122	118	113	108	99.6	91.9
	4.8	65.3	63.3	61.2	59.0	56.7	54.5	52.2	50.1	46.1	42.5
	5.8	81.7	79.2	76.5	73.7	70.9	68.1	65.3	62.7	57.7	53.2
M 20	8.8	131	127	122	118	113	109	105	100	92.3	85.1
	10.9	184	178	172	166	159	153	147	141	130	120
(2.5)	12.9	220	214	206	199	191	184	176	169	156	144

In contrast to 5.8 bolts, the tension forces F_{sp} for bolts in other (nonferrous) property classes must be converted based on their yield points or their yield strengths; e.g. for M6 Cu2 fastener with $\mu_G = 0.10$, $F_{sp} = (340/400) \cdot 6.1 \cdot 10^3 \approx 5.2 \cdot 10^3$N.

PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

Advantages of the K-method over previously used methods of determining tightening torque

- Easily takes into account the different coefficients of friction μ_K and μ_G.

The K-method therefore allows for more reliable design and greater minimum installed tension force (if the example below were calculated using $\mu_{tot} = \mu_K$, the maximum tension force would be impermissibly high, reaching a value just below the yield point. If the calculation were made using $\mu_{tot} = \mu_G$ instead, the tightening torques would be lower and the minimum tension force would be only 68 % of 14,800 N).

- In order to express the values listed in Table 3 as μ_{tot} values, more comprehensive tabular information would be required. Table 3 requires only a fraction of this information.
- The K-method is used to precisely determine the tightening torques when tightening threaded fasteners using a torque wrench, and is an advantageous replacement for the tables and somewhat complicated graphs used previously.
- Using yield point or angle-controlled tightening methods, better utilization factors can still be achieved at threaded-fastener connections (see "Tightening of threaded fasteners").

Determining tension forces and tightening torques

The maximum attainable installed tension force $F_{sp\ max}$ is determined using the smallest coefficient of friction in the threads μ_G and Table 5. (Most of the torque applied to the screw shank is used to overcome thread friction, while only the remainder is converted to tension force by the thread pitch; the maximum permissible tension force must not be exceeded even when this residual torque reaches its maximum value and μ_G its minimum value.)

The maximum permissible installed tightening torque $M_{sp\ max}$ is calculated using the K-value for the lowest values for μ_K and μ_G (because the maximum permissible tensioning force must not be exceeded even when maximum tightening torque coincides with the smallest coefficients of friction).

$$M_{sp\ max} = K_{min} \cdot F_{sp\ max} \cdot d$$

The lowest installed tightening torque $M_{sp\ min}$ is a result of the quality of the tightening process and the equipment used. If good power screwdrivers are used or the tightening process is carefully performed by hand using torque wrenches, the nominal torque can usually be achieved within a variation ±10 %. The lowest installed tightening torque is then

$$M_{sp\ min} = 0.8 \cdot M_{sp\ max}$$

The lowest installed clamping force $F_{sp\ min}$ which then may occur is calculated using the K-value for the maximum values of the two coefficients of friction μ_K and μ_G as follows:

$$F_{sp\ min} = M_{sp\ min}/(K_{max} \cdot d)$$

Example: *An M 10-8.8 bolt or screw (pressed and rolled, phosphated) is used to fasten a ground steel part to an AlMg housing with blind threaded holes; the parts are lightly oiled.*

From Table 3: $\mu_K = 0.16 \ldots 0.22$

$\mu_G = 0.08 \ldots 0.20$ → for $\mu_G = 0.08$ in accordance with Table 3: $F_{sp\ max} = 29{,}500$ N

From Table 4: $K_{min} = 0.176$ | $K_{max} = 0.280$

$M_{sp\ max} = 0.176 \cdot 29500 \cdot 0.010 = 51.9$ N · m

$M_{sp\ min} = 0.8 \cdot M_{sp\ max} = 41.5$ N · m

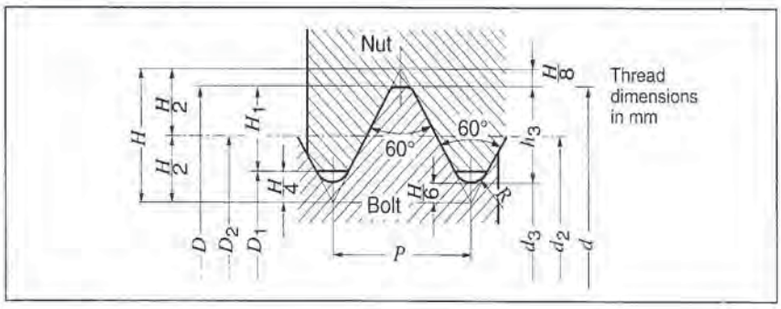
$F_{sp\ min} = 41.5/(0.280 \cdot 0.010) = 14800$ N (with d in m)

PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

Threads (Selection)

ISO metric screw threads

(DIN 13, ISO 965); Nominal dimensions



Metric standard thread

Example of designation: M8 (nominal thread diameter: 8 mm)

Nominal thread dia. $d = D$	Pitch P	Pitch diameter $d_2 = D_2$	Minor diameter d_3	Minor diameter D_1	Thread depth h_3	Thread depth H_1	Tensile stress area A_s in mm²
3	0.5	2.675	2.387	2.459	0.307	0.271	5.03
4	0.7	3.545	3.141	3.242	0.429	0.379	8.78
5	0.8	4.480	4.019	4.134	0.491	0.433	14.2
6	1	5.350	4.773	4.917	0.613	0.541	20.1
8	1.25	7.188	6.466	6.647	0.767	0.677	36.6
10	1.5	9.026	8.160	8.376	0.920	0.812	58.0
12	1.75	10.863	9.853	10.106	1.074	0.947	84.3
14	2	12.701	11.546	11.835	1.227	1.083	115
16	2	14.701	13.546	13.835	1.227	1.083	157
20	2.5	18.376	16.933	17.294	1.534	1.353	245
24	3	22.051	20.319	20.752	1.840	1.624	353

Metric fine thread

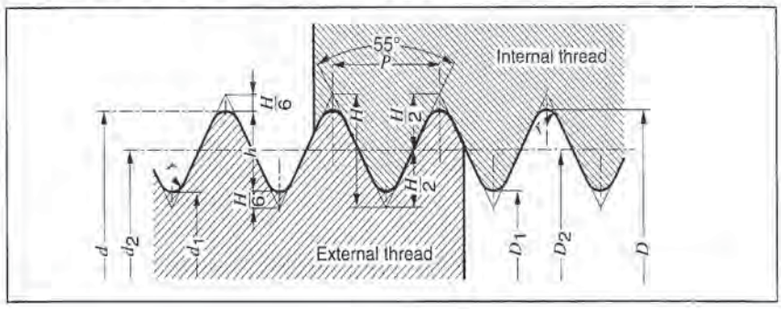
Example of designation: M8 x 1 (Nominal thread diameter: 8 mm; pitch: 1 mm)

Nominal thread dia. $d = D$	Pitch P	Pitch diameter $d_2 = D_2$	Minor diameter d_3	Minor diameter D_1	Thread depth h_3	Thread depth H_1	Tensile stress area A_s in mm²
8	1	7.350	6.773	6.917	0.613	0.541	39.2
10	1.25	9.188	8.466	8.647	0.767	0.677	61.2
10	1	9.350	8.773	8.917	0.613	0.541	64.5
12	1.5	11.026	10.160	10.376	0.920	0.812	88.1
12	1.25	11.188	10.466	10.647	0.767	0.677	92.1
16	1.5	15.026	14.160	14.376	0.920	0.812	167
18	1.5	17.026	16.160	16.376	0.920	0.812	216
20	2	18.701	17.546	17.835	1.227	1.083	258
20	1.5	19.026	18.160	18.376	0.920	0.812	272
22	1.5	21.026	20.160	20.376	0.920	0.812	333
24	2	22.701	21.546	21.835	1.227	1.083	384
24	1.5	23.026	22.160	22.376	0.920	0.812	401

PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

Pipe threads for non self-sealing joints

(DIN ISO 228-1); Parallel internal threads and external threads; nominal dimensions

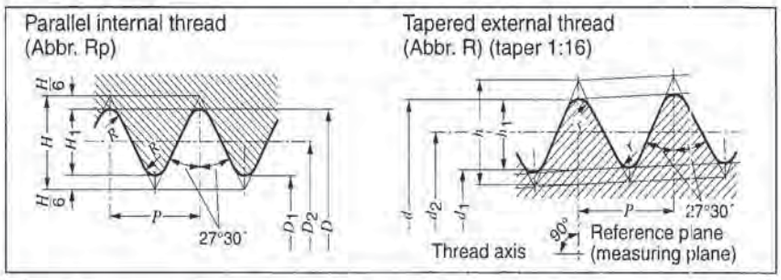


Example of designation: G 1/2 (nominal thread size 1/2 inch)

Nominal thread size	Number of threads per inch	Pitch P mm	Thread depth h mm	Major diameter $d = D$ mm	Pitch diameter $d_2 = D_2$ mm	Minor diameter $d_1 = D_1$ mm
1/4	19	1.337	0.856	13.157	12.301	11.445
3/8	19	1.337	0.856	16.662	15.806	14.950
1/2	14	1.814	1.162	20.955	19.793	18.631
3/4	14	1.814	1.162	26.441	25.279	24.117
1	11	2.309	1.479	33.249	31.770	30.291

Whitworth pipe threads for threaded pipes and fittings

(DIN 2999); Parallel internal threads and taper external threads; nominal dimensions (mm)



Designation		Major diameter	Pitch diameter	Minor diameter	Pitch	Number of threads per inch
External thread	Internal thread	$d = D$	$d_2 = D_2$	$d_1 = D_1$	P	Z
R 1/4	Rp 1/4	13.157	12.301	11.445	1.337	19
R 3/8	Rp 3/8	16.662	15.806	14.950	1.337	19
R 1/2	Rp 1/2	20.955	19.793	18.631	1.814	14
R 3/4	Rp 3/4	26.441	25.279	24.117	1.814	14
R 1	Rp 1	33.249	31.770	30.291	2.309	11

Areas of application: For joining pipes with parallel internal threads to valves, fittings, threaded flanges, etc. with tapered external threads.

PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

Joining and bonding techniques

Welding

Automotive components and subassemblies are joined using a wide and highly variegated range of welding and bonding techniques. Resistance-pressure and fusion welding are among the most commonly applied welding methods. This summary concentrates on the resistance-welding procedures in standard production use (processes and symbols based on DIN 1910, Part 5).

Resistance welding

Resistance spot-welding

In resistance spot-welding, locally applied electrical current is used to melt the contact surfaces of the parts to be joined into a soft or fluid state; the parts are then joined together under pressure. The contact-point electrodes which conduct the welding current also convey the electrode force to the parts. The amount of heat required to form the welding joint is determined according to $Q = I^2 \cdot R \cdot t$ (Joule's Law).

The precise amount of heat required is a function of current, resistance and time. The following factors should be coordinated in order to achieve a good weld and adequate spot diameter d_1:

- welding current I,
- electrode force F and
- welding time t.

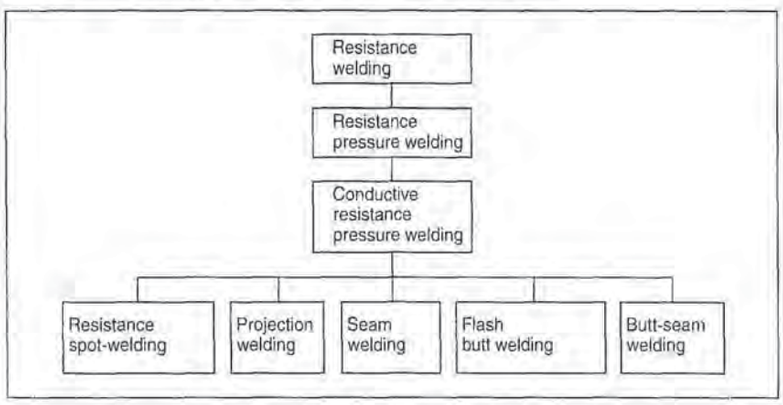
According to the manner in which the current is conducted, a distinction is drawn between:

- bilateral direct resistance spot-welding and
- unilateral indirect resistance spot-welding.

The electrode for a specific spot-welding operation is selected with reference to shape, outside diameter and point diameter. Before welding, the parts must always be completely free of scaling, oxides, paint, grease and oil; they therefore receive appropriate surface treatment prior to welding (where indicated).

Projection welding

Projection welding is a process in which electrodes with a large surface area are employed to conduct the welding current and the electrode force to the workpiece. The projections, which are generally incorporated into the thicker of the workpieces, cause the current to concentrate at the contact surfaces. The electrode force compresses the projections partially or completely during the welding process. A permanent, inseparable joint is produced at the contact points along the welding seam. One or more than one projection can be welded simultaneously, depending on the type of projection (round or annular) and the power available from the welding unit.



PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

Depending on the number of spots welded, a distinction is drawn between
- single-projection welding and
- multiple-projection welding.

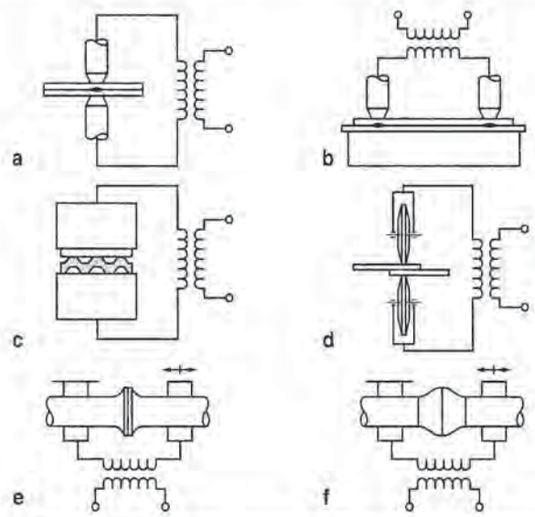
This technique requires high welding currents applied for short periods of time.
Applications:
- welding parts of different thicknesses,
- welding multiple projections in a single operation.

Seam welding
In this process, roller electrodes replace the spot-welding electrodes used in resistance spot-welding. Contact between the roller pair and the workpiece is limited to an extremely small surface area. The roller electrodes conduct the welding current and apply the electrode force. Their rotation is coordinated with the movement of the part.
Applications:
Production of sealed welds or seam spot welds (e.g. fuel tanks).

Flash-butt welding
In flash-butt welding, the butt ends of the workpieces are joined under moderate pressure while the flow of current at the contact surfaces produces localized heat and melting (high current density). The metal's vapor pressure drives molten material from the contact patches (burn-off) while force is applied to form an upset butt weld. The butt ends should be parallel to each other and at right angles to the direction in which the force is applied (or virtually so). Smooth surfaces are not required. A certain amount of extra length must be factored in to compensate for the losses incurred in the flash-welding process.
Result:
A weld with the characteristic projecting seam (burr).

Butt-seam welding
This process employs copper jaws to conduct the welding current to the workpieces being joined. When the welding temperature is reached, the current switches off. Constant pressure is maintained and the workpieces then weld to-

Welding techniques
a) Bilateral resistance spot-welding, b) Unilateral resistance spot-welding, c) Projection welding, d) Seam welding, e) Flash butt welding, f) Butt-seam welding.



PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

gether (requirement: properly machined butting faces). The result is a burr-free seam. The process does not completely displace contamination that may be present at the butt ends.

Fusion welding
The term "fusion welding" describes a process employing limited local application of heat to melt and join the parts; no pressure is applied.

Shielded (inert gas) arc welding is a type of fusion welding. The electrical arc extending between the electrode and the workpiece serves as the heat source. Meanwhile, a layer of inert gas shields the arc and the melted area from the atmosphere. The type of electrode is the factor which distinguishes between the various techniques:

Tungsten inert-gas welding
In this process, an arc is maintained between the workpiece and a stable, non-melting tungsten electrode. The shielding (inert) gas is argon or helium. Weld material is supplied from the side (as in gas fusion welding).

Gas-shielded metal-arc welding
In this process, an arc is maintained between the melting end of the wire electrode (material feed) and the workpiece. The welding current flows to the wire electrode via sliding contacts in the torch holder. Insert-gas metal-arc welding uses inert gases (slowly reacting and noble gases such as argon, helium, or combinations of the two) as protective gas. Active-gas metal-arc welding, on the other hand, employs reactive gases such as CO_2 and mixed gases containing CO_2, argon and sometimes oxygen, and is frequently referred to as CO_2 welding.

This process is employed for welding non-alloy and mild-alloy steels.
In addition, the following welding techniques are employed in the automobile industry:
- electron-beam welding,
- friction welding,
- arc pressure welding (stud welding),
- stored-energy welding (pulse welding).

Soldering

In soldering, a supplementary material (solder) is melted onto two or more parts of similar or varying metallic composition in order to produce a permanent connection between them. Flux and/or protective gas may also be used. The melting temperature of the solder is below that of the parts being joined. The solder is distributed along the join to produce the connection without the parts themselves being melted.

Soldering processes are classified by working temperature. This is defined as the lowest surface temperature at the connection between the workpieces to be joined at which the solder can be melted and distributed to form a bond.

Soft soldering
Soft soldering is employed to form permanent solder joints at melting temperatures below 450 °C (as with soldering tin). Soft solders which melt at temperatures of 200 °C and below are also known as quick solders.

Hard soldering
Hard soldering (brazing) is used to form permanent joins at melting temperatures above 450 °C (as with copper/zinc, combined copper/zinc and silver alloys, e.g. silver brazing filler). Further data on soldering materials is contained in DIN Sheets 1707, 8512, 8513 and 8516.

Fluxes (non-metallic substances) are applied to remove any film (oxidation) remaining on the surface of the parts after cleaning and to prevent a new film from forming: this makes it possible to apply a consistent coat of solder to the joint surfaces. Data on fluxes can be found in DIN 8511.

The strength of a soldered joint can be equal to that of the base material itself. This phenomenon is due to the fact that the more rigid adjoining materials limit the solder's deformation potential.

The method used for heating provides yet another criterion for the classification of soldering processes. The two standard types are: open-flame soldering and iron soldering.

PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

Flame soldering
A hand-held burner or a gas-fired unit provides the heat. Depending on the specific soldering operation, either oxy-acetylene burners (familiar from gas welding) or soldering lamps are employed.

Iron soldering
A hand-held or mechanically guided soldering iron provides the heat. Irons can also be used to solder pretinned surfaces.

Other processes include: oven, salt-bath, immersion, resistance, and induction soldering.

Adhesive technologies

Organic or anorganic adhesives are employed to form permanent rigid connections between two metallic or non-metallic materials. The adhesive bonds under pressure, either at room temperature or with moderate heat. The bond is adhesive, in other words physical and chemical bonding forces operate at the molecular level to bond the surfaces. A distinction is drawn between single-component and two-component adhesives, referring to their as-delivered condition.

Single-component adhesives
These are components which contain all the constituents necessary to form a bond.

Two-component adhesives
The second component in an adhesive of this type is a hardener that initiates the cross-linking process. An accelerator may be added to the hardener. Metal adhesives are generally of the two-component type. The hardening (cross-linking) process is polymerization, polycondensation or polyaddition under the influence of temperature and/or time. Spatially interlinked macro-modules are the result. Depending on the hardening temperature, a distinction is drawn between cold adhesives (harden at room temperature, relatively easy to apply, and hot adhesives harden at 100 ... 200 °C).

Adhesive connections should be designed to ensure that the bond is subjected exclusively to tensile shear loads. Overlapping connections are virtually the only application for adhes ves. Butt joins subjected to tensile or sliding forces should be avoided.

Metal adhesives can be employed in conjunction with spot welding. The adhesive prevents premature swelling of the sheets between the resistance-welded spots. This method is also suitable for reducing stress peaks at the edges of the weld spots and limited the number of spots required. Constructions of this nature evince enhanced structural integrity, rigidity, and damping when subjected to dynamic loads. Welding is carried out when the adhesive is still soft; otherwise the adhesive would act as an insulator.

The most significant metal adhesives include: epoxy, polyester and acrylic resins, vinyl acetate, and metal cement.

Automotive applications
Adhesive joining has become a standard technique in automotive engineering. The individual areas of application can be classified as follows.
- Body shell: Raised-seam and brace bonding for attached components.
- Painting line: Attachment of stiffeners.
- Assembly line: Attachment of insulating material, appliqués, moldings, mirror-support bracket to windshield.
- Component production: Bonding brake pads, laminated safety glass (LSG), rubber-to-metal connections to absorb vibration.

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US PATENT NO. 6,581,565

Riveting

Riveting is used to produce a permanent fixed connection between two or more components made of identical or dissimilar materials. Depending on the method and application, riveted connections are divided into the following categories:
- permanent rigid connections (interference fits, for example in mechanical and plant engineering),
- permanent, sealed connections (for example in boilers and pressure vessels) and
- extremely tight seals (for example in pipes, vacuum equipment, etc.).

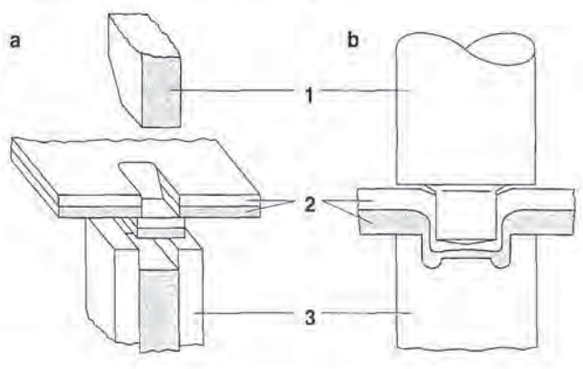
In some areas such as mechanical engineering in general and in tank manufacture, riveting has largely been displaced by welding. A distinction is drawn between cold and hot riveting, depending on the temperature used. Cold riveting is employed for rivet joints up to 10 mm in diameter in steel, copper, copper alloys, aluminum, etc. Rivets with diameters in excess of 10 mm are installed hot.

The most common types of rivet are mushroom head, countersunk, tallow-drop, hollow and tubular. There are also standardized rivets for specialized applications, for example explosive rivets and pop rivets. Pop rivets are hollow and are expanded by a drift or punch. The types of rivet and the corresponding materials are defined in DIN sheets; structural integrity and chemical composition are specified in DIN 17 111. In order to avoid the danger of electrochemical corrosion, the rivet material and the material of the parts being joined should be similar, if possible.

Advantages/disadvantages compared to other joining techniques
- Unlike welding, riveting exerts no effects such as hardening or molecular alteration on the material,
- no distortion of the components,
- suitable for joining dissimilar materials,
- riveting weakens the components,
- butt joints cannot be riveted and
- riveting is generally more expensive than welding when performed outside the factory.

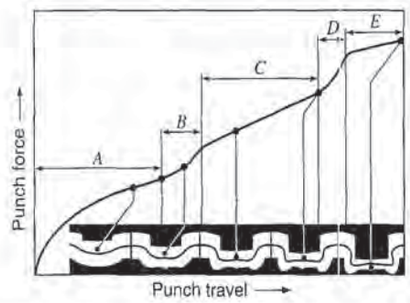
Automotive applications
- Riveting joint pins (window winders, windshield-wiper linkages),
- Riveting reinforcement plates (in the course of repairs).

Pressurized clinching
a) Clinching, b) Tox clinching; 1 Punch, 2 Work, 3 Die.



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US PATENT NO. 6,581,565

Punch force/punch travel curve
A Combined indentation and melding.
B Upsetting and expansion.
C Filling upper contour of mold.
D Filling annular canal.
E Cup extrusion.



Bonding and joining (pressurized clinching)

Pressurized clinching is a mechanical process for penetration assembly of layered materials. It combines cutting, penetration and cold upsetting in a single continuous joining operation without the application of additional heat. The basic principle is one of joining through reshaping. DIN 8593, Part 5, issued in September 1985 contains the first reference to this process as a means of joining sheet-metal panels by deformation.

The past few years have marked the advent of tox clinching as yet another joining process: it resembles penetration clinching, but does not include a cutting process. The tools employed for tox clinching are relatively small. The diameter can be varied to suit specific applications. At present, pressurized clinching can be used to join panels up to 3 mm thick, whereby the total thickness of the two sheets should not exceed 5 mm. The panels being joined can be of the same material (e.g. steel/steel) or dissimilar materials (e.g. steel/non-ferrous metal). In addition, pressurized clinching can be used to join coated sheets and painted parts, as well as components to which adhesives have been applied. Multiple clinching can be used to produce numerous pressure-assembly elements (up to 50) in a single process (one stroke of the press, for example). The typical curve for punch force relative to punch travel can be divided into five characteristic phases (A...E).

Advantages/disadvantages of pressurized clinching
- No need for noise-abatement encapsulation,
- tox clinching does not impair corrosion protection,
- when combined with a cutting operation there is a partial loss of corrosion protection,
- no heat distortion,
- painted, protected (oil, wax) and glued sheets can be clinched,
- different materials can be joined (e.g. steel/plastic)
- energy savings, no power supply required for welding and no cooling water,
- one side of the workpiece evinces a projection similar to that produced by a rivet head, while the other side has a corresponding depression.

Automotive applications
- Windshield wiper brackets,
- Fastening door interior panels,
- Positioning individual components.

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US PATENT NO. 6,581,565

Punch riveting

The punch riveting technique joins solid materials employing stamping and riveting elements (solid or semi-hollow rivets) in a combined cutting and joining operation.

The parts to be joined do not have to be predrilled and stamped, as is the case with other riveting techniques.

Punch riveting with solid rivets

The first stage in punch riveting with solid rivets is to position the work on the die plate.

The top section of the rivet unit, including the blank holder, descends and the rivet die presses the rivet through the parts to be joined in a single stamping operation.

Punch riveting with semi-hollow rivets

The first stage in punch riveting with semi-hollow rivets is to position the work on the (bottom) die plate. The rivet die descends and presses the semi-hollow rivet through the upper sheet into the lower sheet in a single stamping operation. The rivet deforms and the bottom spreads to form a securing element, usually without fully penetrating the lower sheet.

Equipment

Highly rigid hydraulic tools are used in punch riveting. Rivets can be singled from a batch feed or carried in belts with individual holders to the riveting tool.

Materials

The rivets must be harder than the materials to be joined. The most common materials are steel, stainless steel, copper and aluminum with various coatings.

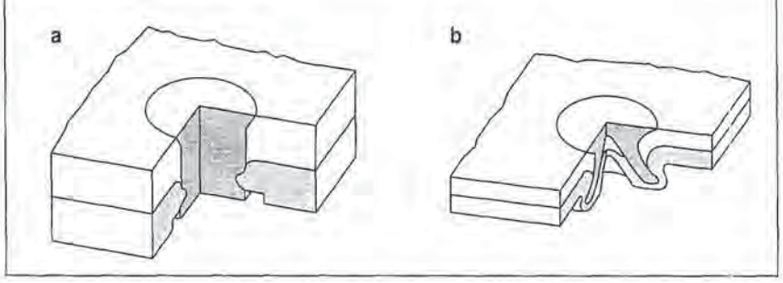
Characteristics

- Used to join similar and dissimilar materials (e.g. steel, plastic or aluminum), parts of various thicknesses and strengths, and painted sheets.
- No preliminary stamping or drilling, no electrical current or vacuum extraction required.
- Approved overall material thicknesses: steel 6.5 mm; aluminum 11 mm.
- Low-heat, low-noise joining process.
- Tools have a long service life (approx. 300,000 rivet applications) with consistent quality.
- Process dependability with monitored process parameters.
- High forces have to be applied.
- Restricted jaw reach.

Applications

Punch riveting with solid rivets: joining metal sheets in automotive engineering, e.g. window winders in passenger cars. Punch riveting with semi-hollow rivets: joining materials at the body-in-white stage in automotive engineering, household appliances, joining metals to composites (heat shields).

Punch-riveted joints
a) Solid rivet, b) Semi-hollow rivet.



PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

Sheet-metal processing

Deep-drawing technology

Units and symbols

Symbols		Units
D	Section diameter	mm
d_B	Support diameter of hold-down device	mm
d_1	Stamping diameter, 1st draw	mm
d_2	Stamping diameter, 2nd draw	mm
F	Total deep-drawing force	kN
F_B	Hold-down force	kN
F_Z	Deep-draw force	kN
p	Pressure of hold-down device	N/mm²
R_m	Tensile strength	N/mm²
s	Sheet thickness	mm
β_1, β_2	Individual drawing ratio	
β_{max}	Maximum possible drawing ratio	

Drawing methods

The drawing procedure employs a drawing die, punch and sheet-metal retaining device to reshape flat sections, blanks and round plates.

The variables exercising an influence on the deep-drawing of three-dimensional bodywork components are exceedingly complex; rough calculations thus do not provide a suitable basis for their definition. The finite-element method (FEM), in which powerful computers employ numerical procedures from continuum mechanics, is used to calculate deep-drawing processes. Software development has reflected the significance of specific parameters affecting the deep-drawing process, including the influence of friction, of bilateral contact, of material values which vary according to rolling direction, and additional factors.

Deep-draw simulation is of major assistance in designing deep-drawing tools, helping to optimize production times for this equipment.

Deep-drawing process

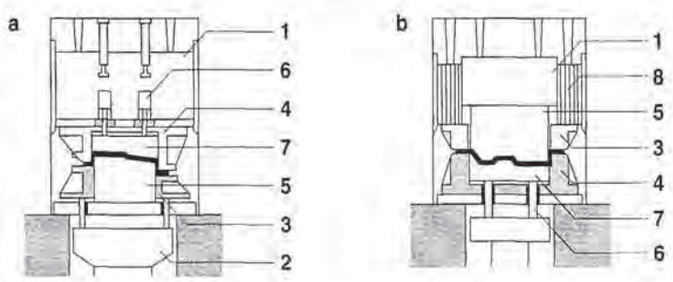
The drawing process reshapes a flat blank into a concave section. A punch, surrounded by a retaining mechanism (holding-down clamp, blank holder), pulls the sheet metal into the matrix. The clamp applies the defined force to the sheet, prevents creases from forming, and permits application of tensile force.

The maximum possible drawing ratio is determined by various factors:
- material strength,
- tool dimensions and sheet thickness,
- retention force,
- friction,
- lubrication,
- material and surface of the workpiece.

Drawing with single-action presses

This category includes all crank presses in which the ram represents the only moving component.

Deep drawing
a) With single-action press, b) With double-action press; 1 Ram, 2 Air cushion, 3 Hold-down device, 4 Matrix, 5 Punch, 6 Air pins, 7 Matrix insert, 8 Hold-down ram.



PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

Applications: for deep-drawing flat workpieces in simple reshaping operations to produce trimmers, arches, supports.

Drawing on double-action presses

Double-action drawing presses employ separate rams to retain the metal and to pull it. The hold-down ram is actuated mechanically via disk or toggle, or hydraulically.

Applications: for deep-drawn components of complex geometry, including fenders, wheel arches, decklids.

Calculations for the drawing process

$$F_Z = \pi \cdot (d_1 + s) \cdot s \cdot R_m$$

$$F_B = \frac{\pi}{4} \cdot (D^2 - d_B^2) \cdot p$$

$$F = F_Z + F_B$$

$$\beta_1 = D/d_1;\ \beta_2 = d_1/d_2.$$

Example

D = 210 mm; d_B = 160 mm; d_1 = 140 mm; p = 2.5 N/mm²; R_m = 380 N/mm²; s = 1 mm; β_{max} = 1.9.

Result

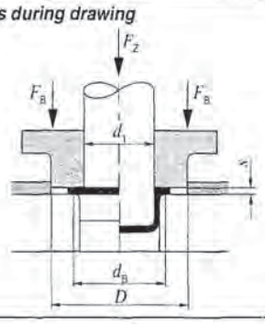
Drawing ratio β_1 = 1.5; Deep-drawing force F_Z = 112.2 kN; Sheet retention force F_B = 36.3 kN; Total deep-drawing force F = 148.5 kN.

The diagram shows the motions of the drawing ram and the hold-down ram in the course of one cycle. The rams have differing strokes. The drawing ram, with its longer stroke, is set in motion first, while the shorter-stroke retention ram starts second, but is first to achieve contact.

Material	β_1 max.	β_2 max.	Pressure of hold-down device p N/mm²
St 12	1.8	1.2	2.5
St 13	1.9	1.25	
St 14	2.0	1.3	
Copper	2.1	1.3	
CuZn 37 w	2.1	1.4	2.0...2.4
CuZn 37 h	1.9	1.2	
CuSn 6 w	1.5	–	
Al 99.5 w	2.1	1.6	
AlMg 1 w	1.85	1.3	1.2...1.5
AlCuMg 1 pl w	2.0	1.5	
AlCuMg 1 pl ka	1.8	1.3	

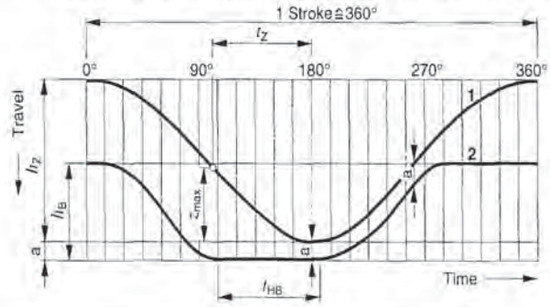
1) Valid up to d_1/s = 300; specified for d_1 = 100 mm, s = 1 mm. Different sheet thicknesses and punch diameters result in minor deviations from the figures given. Basic qualities are also used with galvanized surfaces. The figures can vary slightly.

Forces during drawing



Time-distance diagram for a double-action drawing press

1 Curve for drawing ram, 2 Curve for hold-down ram;
h_z Stroke of drawing ram, h_B Stroke of hold-down ram, z_{max} Max. drawing depth,
t_z Duration of drawing process, t_{HB} Holding time of hold-down ram, a Approach of drawing ram



PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

Laser technology

Deviation mirrors are used to deflect the coherent light beam generated by the laser for concentration in a focusing device. At the focal point the laser beam has a diameter of approx. 0.3...0.5 mm, allowing intensities in excess of 10^6...10^8 W/cm^2 to be achieved. The workpiece melts and vaporizes in milliseconds if the focal point is in its immediate vicinity, an effect which can be exploited in laser-based processing operations. According to the physical condition of the active medium, the device is classified as a solid-state, gas, semiconductor or liquid laser. The following are applied in industrial metal processing:

Solid-state laser (Nd:YAG)
Neodynium (Nd) is an element of the rare-earth group, while YAG stands for yttrium-aluminum-garnet ($Y_5Al_5O_{12}$). The crystal exhibits mechanical and thermal stability equal to that of the ruby.

The Nd:YAG laser emits light in the infrared range of the spectrum, with a wavelength of 1.06 µm. The essential advantage of the Nd:YAG laser lies in the fact that the generated beam can be transmitted through optical fibers, making it possible to dispense with complicated beam-relaying systems. Present laser power ratings are in the range of 400... 1200 W; pulse-mode operation is generally employed. The foremost application for solid-state lasers is in welding parts with high requirements for precision, e.g. in precision mechanics. Welding depths depend on power and welding speed, and are in the range of tenths of a millimeter.

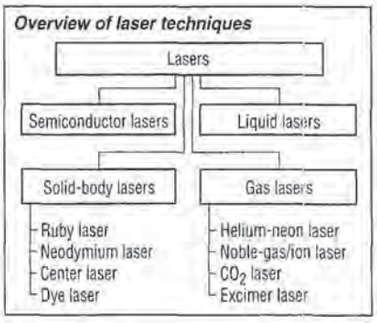
Gas laser (CO_2)
The CO_2 laser is among the most important gas-laser devices. Molecular gas serves as the active medium. Radiation emissions are in the medium infrared band, with wavelengths spread across a number of spectral ranges between 9.2 and 10.9 µm; the mean is 10.6 µm. This type of laser is generally used in continuous operation, with standard laser power ratings of between 2 and 5 kW. Laboratory units with up to 20 kW have already been installed.

Characteristics of the laser cutter
- Clean, burr-free edges,
- high dimensional/manufacturing precision (no subsequent processing necessary),
- minimal thermal and mechanical stresses on the workpiece during the cutting process,
- uncomplicated clamping devices,
- sheet metal of up to 10 mm can be cut (1 kW), rapid cutting (10 m/min with metal 1 mm thick).

Characteristics of laser welding
- High depth/width ratio (e.g. approx. 1 mm seam width at 5 mm seam depth),
- minimal base-material thermal stress results in narrow heat-affected zone (HAZ),
- minimal distortion,
- welding in protective-gas atmosphere,
- joint gaps must be virtually zero,
- welding with filler metals is possible (for bridging gaps and achieving specific effects at the weld),
- specialized clamping and equipment technology is required.

Applications in machine tool and automotive engineering
- Cutting and welding bodywork sheet metal,
- welding (joining) blanks of various thicknesses,
- welding (joining) of rotationally symmetrical parts (transmission components, tappets, automotive components),
- boring and perforation,
- surface hardening, e.g. in treating valve seats, etc.,
- providing surfaces with upgraded finishes, e.g. structural transformation around cylinder liners.

Overview of laser techniques



PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

Tribology

Purpose and goals

Tribology is defined as "The study of the phenomena and mechanisms of friction, lubrication, and wear of surfaces in relative motion". Industrial application is directed toward gathering information for use in extending the service lives of products and maximizing utilization of resources. The specific activities are as follows:

- Analysis of friction and wear,
- Analysis and evaluation of tribological damage,
- Provision of technical recommendations on materials, lubricants and design (for damage control and in the design of new components and products),
- Quality assurance,
- Assistance in providing optimal performance,
- Service-life assessment,
- Development and selection of new materials and lubricants.

The complexity of the task renders a synergetic approach essential. The disciplines involved include materials science, physics, chemistry and mechanical engineering.

Definitions

Friction (DIN 50281)
Friction is physical resistance to relative motion at two or more surfaces in a state of mutual contact. The most significant physical parameters for describing friction are:
Friction force: The amount of force with which a motion is resisted.
Coefficient of friction (or friction factor): Friction force relative to normal force.
Friction power: Friction force x sliding speed.
Classification of friction:

1. According to condition of friction (type of contact),
2. According to type of motion.

Condition of friction (type of contact)
Gas friction: A gas layer completely separates the base object and the opposed body from one another, thereby assuming the entire load.
Fluid friction (hydrodynamic and hydrostatic friction): A fluid layer completely separates the base object and opposed body from one another, thereby assuming the entire load.
Combination friction: The base object and the opposed body are in mutual contact at surface peaks. The load is shared by the fluid/lubricating film and by the objects in contact.
Dry or boundary friction: The lubricating film no longer performs any support function, but residue from previously absorbed lubricants continues to exercise a tribological effect.
Solid-body friction: Direct contact between the opposed surfaces.

Types of motion
Possible types of motion contact include sliding friction, rolling friction, and combinations of the two.

In many cases the effects which friction produces in machine components are undesirable, as there are negative consequences for energy consumption and/or associated temperature increases and/or changes to the material. In other cases, friction can make a necessary contribution to proper operation. This is the case with self-locking transmission devices, where the lubricant must furnish a specific coefficient of friction, and in some types of clutches in which defined levels of friction are also required.

Tribological concepts (DIN 50323)
Tribology embraces the science and associated technology devoted to the interaction of surfaces in mutually opposed states of motion. It focuses on the entire range of friction, wear and lubrication, and also includes the effects at the contact surfaces of solids as well as those between solids and gases.

Tribo-technology (DIN 50323)
This branch is devoted to the actual technical application of tribology.

Tribological stress (DIN 50320)
The stress which results at a solid body

PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

from contact with, and the relative motion of, an opposed body in solid, fluid or gaseous form.

Tribological damage
Damage resulting from tribological stress. This contrasts with the concepts of wear and wear damage as defined in DIN 50320, which always include material erosion. Here, the term "tribological damage" extends to embrace both numerous tribologically induced changes to the material's surface, as well as reductions in operational efficacy which according to DIN 50320 are not regarded as wear.

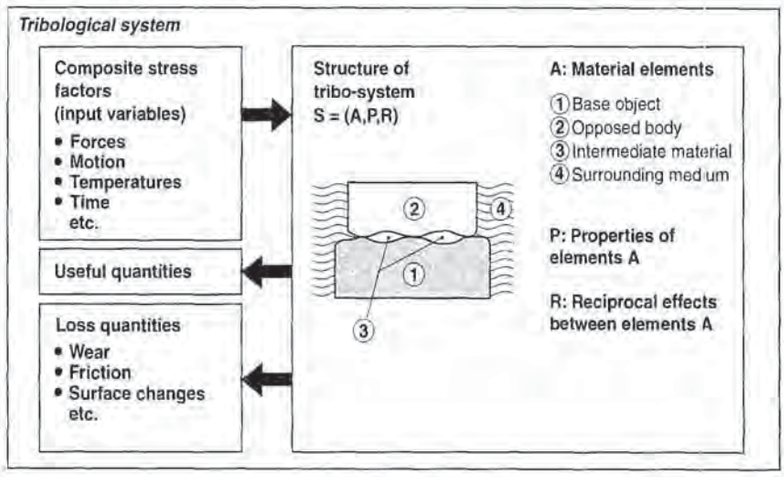
Wear (DIN 50320)
Progressive loss of material from the surface of a solid body, caused by tribological stress. Wear is characterized by the presence of abraded particles, as well as changes in both the material and structure of the surface which is exposed to tribological stress. This type of wear is generally undesirable, and impairs functionality (exception: "running-in" processes). Thus any such procedures which enhance the object's operational value are not considered as wear.

Tribological system

Wear can be regarded as a system characteristic. There exists no specific material value for "wear-resistance" as such, corresponding to, say, tensile strength in materials science. The tribological system for the contact surfaces which are the focus of the present discussion consists of:
- The material "A elements": base object, opposed body, intermediate material and surrounding medium (see illustration),
- The properties P of the elements A,
- The reciprocal influence R between the elements A.

The elements' properties and their reciprocal effects form the structure of the tribological system. As the composite stress factor (comprising forces, motion, temperature) acts upon this structure, it is transformed into useful quantities and loss quantities. The latter include friction and wear.

Because tribological stresses are surface stresses, the previously mentioned P properties must also be viewed as surface characteristics. In technical applications, there frequently exists a substantial discrepancy between the material values measured for the base material and those actually found upon examination of the surfaces. Manufacturing procedures, cleaning processes, and the operating environment can all cause changes in the surface layer, provoking attendant variations in the tribological response of the material.



PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

Types of wear

Wear processes can be classified according to the type of stress and the materials involved (system structure), a classification from which the following types of wear emerge (DIN 50320):

System structure
Solid body/solid body:
Sliding → Sliding wear,
Rolling → Rolling wear,
Oscillation → Vibration wear.

System structure
Solid body/liquid:
Flow/oscillation → cavitation erosion

Manifestations of wear

The manifestations of wear as defined by DIN 50 320 apply to the surface changes resulting from abrasion, as well as to the type and shape of the resulting particles.

Wear mechanisms

Adhesion
Formation and separation of (atomic) surface bonds.

There is a transfer of material when the separation deviates from the original boundary between the base object and the opposed body.

The adhesion process starts at the molecular level, but can expand until damage on a massive scale occurs (seizing).

Abrasion
Scraping stress and microscopic scraping action, performed by the base object, the opposed body, reaction products or solid particles in the intermediate medium.

Surface fatigue
Alternating tribological loads (e.g. impact, rolling, sliding stress, cavitation) cause mechanical stresses. The resulting fissures lead to material separation (separation of wear particles).

Tribo-chemical reactions
Reaction of the base and/or opposed object with the intermediate material (and surrounding medium), caused by tribological stress.

Wear quantities

Rates of wear are defined by the so-called wear quantites. These provide direct or indirect indices of the variations in the shape or in the mass of a body which are traceable to wear (definitions based on DIN 50 321).

Coefficient of wear
The quantities by which wear is defined can only be indicated as system-specific quantities.

The wear coefficient k facilitates comparison of wear rates at varying surface pressures and speeds:

$k = W_v/(F \cdot s)$ in $mm^3/(N \cdot m)$

F Force,
W_v Volumetric wear,
s Sliding distance.

Tribological damage analysis

In cases where operation of a component results in damage or changes of a kind which could be expected to affect the component's functional integrity, DIN 50320 indicates that all stresses and characteristics occurring at the points of tribological contact should be investigated (DIN 50320, Appendix A).

The topographical and material analysis of tribological contact (base object, opposed body, intermediate material) provides information on the mechanisms causing the damage and/or wear (adhesion, abrasion, surface fatigue, tribochemical reaction). These investigations also provide data on accompanying phenomena and on changes in the lubricant.

The analysis must be based on the following information:
– surface structure,
– material composition of the surface,
– microstructure,

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US PATENT NO. 6,581,565

- microhardness,
- intrinsic tension,
- material composition and chemical/physical changes in the lubricant.

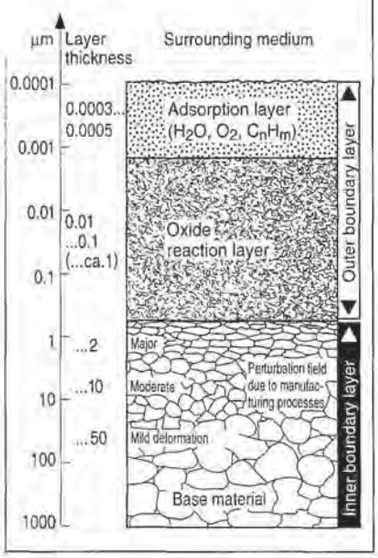
Tribological test procedures

DIN 50 322 divides wear testing into 6 categories:
Category I: Operational testing.
Category II: Test-stand procedure with complete machine.
Category III: Test-stand procedure with assembly or entire unit.
Category IV: Testing with unmodified component or reduced assembly.
Category V: Simulated operational stress with test specimens.
Category VI: Model test with simple test specimens.
The original production assembly is employed for testing according to Categories I through III, while the structure of the system is modified substantially from Category IV onward. It is highly advisable to integrate the tribological examinations of essential and safety-related products into a tribological test series:
- Model testing (e.g. pin and sphere, plate for sliding and vibration wear),
- Component testing,
- Product testing.

Model tests (with simple test specimens), although frequently employed in basic research, are no longer used for examining complex tribological systems. Of the various elements in the sequence of tribological tests, it is examination of the actual product itself that provides the most reliable information. Modern research procedures (such as radio-nuclide testing) make it possible to garner precise data on wear as a function of composite stress.

For radio-nuclide testing (RNT), the parts to be examined are marked radioactively. The abraded particles can then be detected "on-line" by gamma-ray measurement.

Material layers subject to tribological stress at surface



Inhibiting wear

Variations in design, materials and lubricants can all be employed to improve tribological properties.

Design
The design options include:
- Improving the surface topography,
- Reduction of surface pressures by raising the contact-surface ratio,
- Surface reinforcement,
- Improving lubrication efficiency.

Material
The materials options include:
- Skin-layer hardening,
- Skin-layer remelting,
- Surface-layer remelt alloying,
- Thermochemical processes,
- Electrical and chemical plating,
- Resurface welding,
- Thermal spraying,
- Metal coating,
- PVD/CVD layers (physical/chemical vapor deposition, e.g., TiN, TiC),
- Ion-beam treatment (e.g. ion implantation, ion-beam mixing, transition-layer mixing and ion-beam supported coating),

PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

- Diamond-like carbon coatings (metallic),
- Anti-friction paints,
- Penetration coatings (soft metals prevent high local surface pressures).

There exist a multiplicity of surface treatment and coating processes. The process parameters can frequently be adjusted to focus on specific kinds of wear. The result is a wide potential field of options for applying tribology to assist products in meeting numerous requirements.

Lubricants

In many cases, selection of the appropriate lubricant (P. 230) and optimal lubrication-system design will exercise a dramatic effect on the tribological properties – the potential is frequently greater than that represented by a change in materials. The choice of lubricant is the initial response when dealing with unsatisfactory operating characteristics.

Lubricity

Lubricity testing is based on specially defined wear parameters. The test results provide a relative index of the efficiency of fluids as separating media in defined tribological systems. In evaluations under otherwise identical conditions, a fluid displays a higher lubricity than another when it produces relative reductions in wear between the opposed elements in the tribological system and/or lower generation of friction energy (DIN/ISO 12156).

It is important to distinguish between lubricity and viscosity.

Descriptions and specific numerical data on lubricity are derived from product, component and model wear testing with specifically defined stress and load parameters. The results are indicated in specific test data for wear.

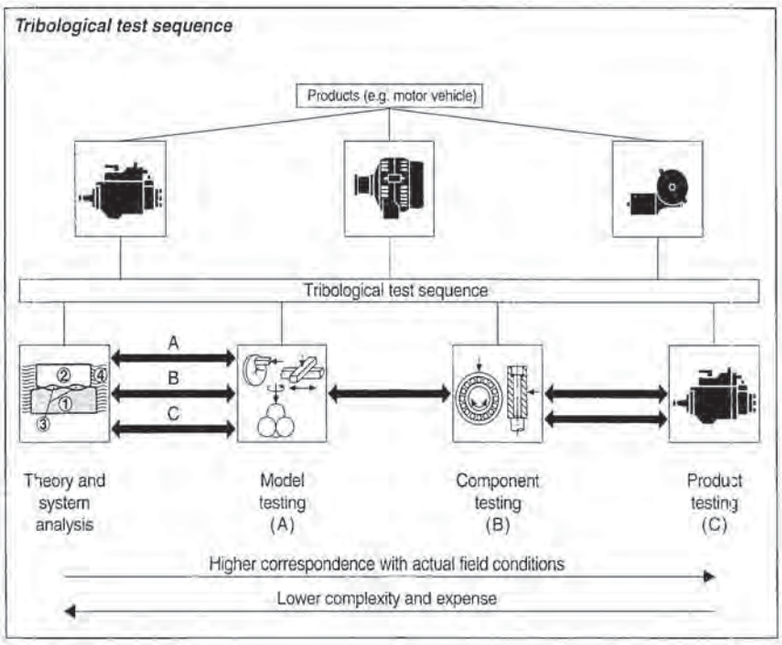
The results of this type of testing are valid only for the object tribological system.

In many products the separating element in the tribological system is also a flow medium, as in hydraulic and fuel-injection systems.

The lubricity level of such a fluid must suffice to ensure reliable operation and a long service life.

Special additives can be employed to enhance lubricity.

Tribological test sequence



PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

Road-going vehicle requirements

Driveability

The vehicle must be capable of effecting the transition from a stationary to a mobile state. Once in motion, it must be able to ascend gradients and accelerate to the desired cruising speed with a reasonable degree of alacrity. With a given power P, ideal correspondence with these requirements is achieved when force M and the engine speed n can be varied according to the formula $P = M \cdot n$. Under these conditions, the limits on the potential field of operation are defined by $P_{Nominal}$, resulting in an inverse relationship between available force and vehicle speed (limit curve defined by tractive-force hyperbola).

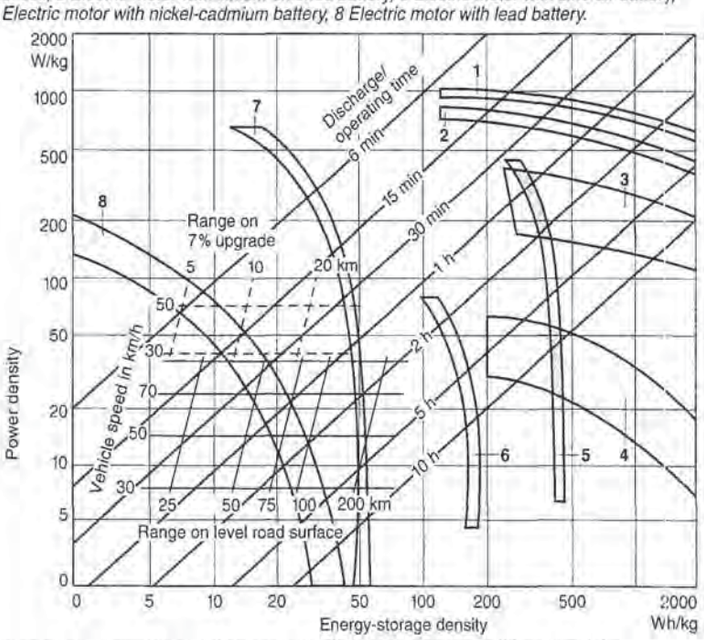
Power density and storage density

Both the power density (W/kg) and the energy-storage density (Wh/kg) of the combined engine/energy-storage system must be high if the size of the vehicle – and with it the mass to be accelerated – are to be maintained at a modest level. Low density factors for power and energy storage would increase vehicle size and mass, with attendant escalations in the power and energy consumption which would be needed to achieve the desired performance (acceleration, speed).

Discharge or operating time of the energy-storage device

The vehicle's operating time – and thus the operating range which can be covered before the energy-storage device must be replenished (renewed/refilled/recharged) – is a function of the energy-storage density, the power requirement, and the vehicle weight; the latter, in turn, is influenced by the power-to-weight ratio of the drive system.

Power and energy-storage densities for various propulsion concepts *(Engine/motor and storage unit).*[1)]
1 Gas turbine, 2 Internal-combustion engine, 3 External-combustion engine, 4 Electric motor with fuel cell, 5 Electric motor with lithium-chloride battery, 6 Electric motor with zinc-air battery, 7 Electric motor with nickel-cadmium battery, 8 Electric motor with lead battery.

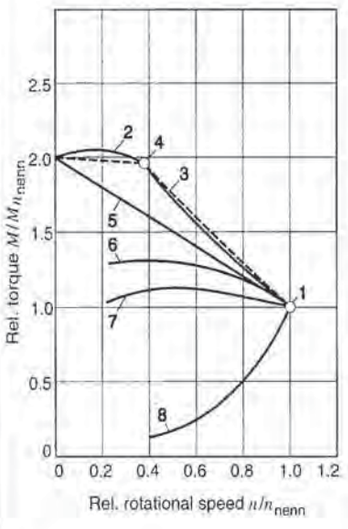


[1)] From "The Automobile and Air Pollution", US Dep. of Commerce (Morsebericht); Mahle "Kolben-Handbuch" (Piston Handbook).

PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

The drive and energy-storage configuration employed for a specific application determines the relationship between power density/energy-storage density and operating time; it also exerts a decisive influence on the shape of the torque curve. Favorable power-to-weight ratios combine with high energy-storage densities (period of operation per tank of fuel) to make internal-combustion engines particularly suitable for vehicular applications. However, the torque curves provided by the standard piston powerplants (diesel and spark-ignition engines) are less satisfactory. They thus need some form of transmission unit for both the transfer and the conversion of torque. The unit must be capable of transmitting torque in the slip range (for starting off) while incorporating various torque-conversion ratios (for ascending gradients and selecting different speed ranges).

Relative torques for various power units
1 Reference point: Gas-turbine base point, Piston engine n_{max}, 2 Steam engine, 3 Electric motor, 4 Limit curve for max. pressure/max. current, 5 Dual-shaft gas turbine, 6 Gasoline (SI) engine, 7 Diesel engine, 8 Single-shaft gas turbine.



Electric and steam-driven powerplants also need a transmission due to the limitations imposed by the respective maxima for current and steam pressure.

In addition to fulfilling the basic requirements enumerated above, the powerplant and energy-storage system must also meet the following demands:

Economic efficiency, characterized by minimal fuel consumption, low manufacturing and maintenance costs, long service life;

Environmental compatibility, with low emissions levels for both pollutants and noise, sparing use of raw materials;

Flexibility in operation, including good starting from −30 °C to +50 °C, operation unaffected by climate and altitude, good drive-off, acceleration and braking characteristics.

Internal-combustion (IC) engines are thus the most favorable option for independent self-contained vehicles, whereby the priority assigned to the individual factors in the above list will vary according to application. Examples:

Passenger cars: High power density, low exhaust and noise emissions, low manufacturing costs,

Trucks and buses: Maximum economy, long service life, and conformity with all emissions requirements.

Special drive systems, such as those relying exclusively on electric motors, or hybrids (dual-system buses, P. 533 ff.) can represent the best, or indeed only, option for special applications and/or under certain operating conditions.

Basic requirements for fuels

Engines featuring external combustion place the least exacting demands on fuel quality, as their combustion and working gases are not identical and remain isolated from each other. In contrast, the fuel used in internal-combustion engines must burn rapidly and virtually without residue.

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US PATENT NO. 6,581,565

Determining fuel consumption

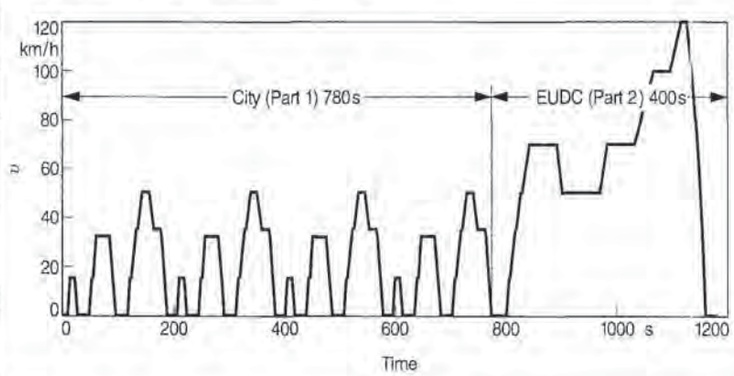
The New European Driving Cycle (NEDC, 93/116/EEC) is a cycle run on a dynamometer to ascertain fuel consumption. It consists of four similarly weighted urban cycles each lasting 195 s and an extra-urban cycle lasting 400 s. The exhaust gas is collected in a sample bag and its components subsequently analyzed. CO, HC and CO_2 are factored into the calculations in accordance with the carbon analysis.

The CO_2 content of the exhaust gas is proportional to the fuel consumption. It can therefore be used as an indicator to gauge the vehicle's fuel consumption (diesel or gasoline, as appropriate).

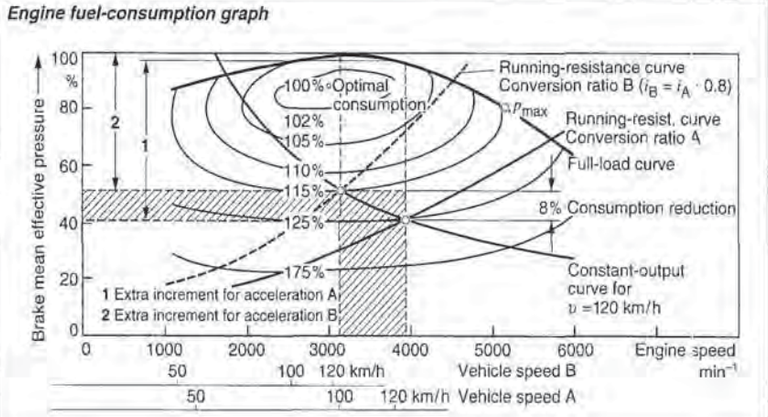
The test mass specified for the vehicle equals the vehicle's empty mass plus a payload of 100 kg. The vehicle mass has to be simulated by finite balanced inertia masses on the dynamometer, so an inertia-mass class is assigned to the vehicle test mass.

Current legislation (EU 2) permits the engine to be started 40 s before testing commences $t = t_0$. This lead-in time will not be allowed under the more stringent EU 3 requirements.

New European Driving Cycle NEDC (93/116/EEC)

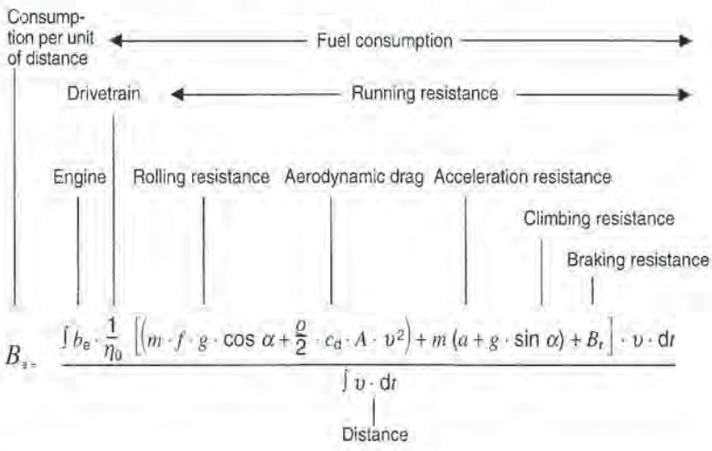


Engine fuel-consumption graph



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US PATENT NO. 6,581,565

Effect of vehicle design on fuel consumption



Symbol		Unit
B_s	Consumption per unit of distance	g/m
η_0	Transmission efficiency of drivetrain	–
m	Vehicle mass	kg
f	Coefficient of rolling resistance	–
g	Gravitational acceleration	m/s²
α	Angle of ascent	°
ϱ	Air density	kg/m³
c_d	Drag coefficient	–
A	Frontal area	m²
v	Vehicle speed	m/s
a	Acceleration	m/s²
B_t	Braking resistance	N
t	Time	s
b_e	Specific fuel consumption	g/kWh

The consumption equation distinguishes between three distinct groups of factors:
– Engine,
– Transmission, and
– External resistance factors.

Transmission (gearbox)

The influence of the gearbox is determined by both the power-transmission losses and the transmission (gear) ratios selected for the application. The former should be kept as low as possible; the latter determine the correspondence between vehicle speed and specific points on the engine's fuel-consumption curve.

"Long" (wide-ratio) gearing will generally move a specific operating state onto a point on the curve corresponding to lower fuel consumption (see illustration), albeit at the expense of reduced acceleration.

External resistance factors

Outside resistance to motion can be counteracted by measures such as weight reduction, improved aerodynamics and lower rolling resistance.

On an average production vehicle, 10 % reductions in weight, drag and rolling resistance result in fuel-economy improvements of roughly 6 %, 3 %, and 2 %, respectively.

The cited formula distinguishes between acceleration resistance and braking resistance, thereby illustrating with particular clarity the fact that the increased fuel consumption associated with acceleration largely stems from subsequent application of the brakes. If the brakes are not applied, then the vehicle's kinetic energy is exploited to impel it forward (retardation = negative acceleration), with attendant reductions in overall fuel consumption.

PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

Motor-vehicle dynamics

Dynamics of linear motion

Symbol		Unit
A	Largest cross-section of vehicle[1])	m^2
a	Acceleration, braking (deceleration)	m/s^2
c_w	Drag coefficient	–
F	Motive force	N
F_{cf}	Centrifugal force	N
F_L	Aerodynamic drag	N
F_{Ro}	Rolling resistance	N
F_{St}	Climbing resistance	N
F_W	Running resistance	N
f	Coefficient of rolling resistance	–
G	Weight = $m \cdot g$	N
G_a	Sum of wheel forces on driven or braked wheels	N
g	Gravitational acceleration = 9.81 $m/s^2 \approx 10 m/s^2$	m/s^2

Symbol		Unit
i	Gear or transmission ratio between engine and drive wheels	–
M	Engine torque	N · m
m	Vehicle mass (weight)	kg
n	Engine speed	min^{-1}
P	Power	W
P_w	Motive power	W
p	Gradient (= 100 tan α)	%
r	Dynamic radius of tire	m
s	Distance traveled	m
t	Time	s
v	Vehicle speed	m/s
v_0	Headwind speed	m/s
W	Work	J
α	Gradient angle	°
μ_r	Coefficient of static friction	–

Additional symbols and units in text.

Total running resistance

The running resistance is calculated as:

$$F_W = F_{Ro} + F_L + F_{St}$$

Running-resistance power

The power which must be transmitted through the drive wheels to overcome running resistance is:

$$P_W = F_W \cdot v$$

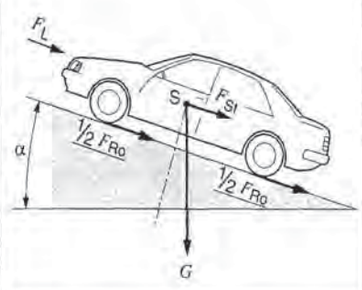
or

$$P_W = \frac{F_W \cdot v}{3600}$$

with P_W in kW, F_W in N, v in km/h.

[1]) On passenger cars A = 0.9 × Track × Height.

Running resistance



Rolling resistance

The rolling resistance F_{Ro} is the product of deformation processes which occur at the contact patch between tire and road surface.

$$F_{Ro} = f \cdot G = f \cdot m \cdot g$$

An approximate calculation of the rolling resistance can be made using the coefficients provided in the following table and in the diagram on page 339.

The increase in the coefficient of rolling resistance f is directly proportional to the level of deformation, and inversely proportional to the radius of the tire. The coefficient will thus increase in response to

Road surface	Coefficient of rolling resistance f
Pneumatic car tires on	
Large sett pavement	0.015
Small sett pavement	0.015
Concrete, asphalt	0.013
Rolled gravel	0.02
Tarmacadam	0.025
Unpaved road	0.05
Field	0.1 ..0.35
Pneumatic truck tires on concrete, asphalt	0.006 ..0.01
Strake wheels in field	0.14 ..0.24
Track-type tractor in field	0.07 ..0.12
Wheel on rail	0.001 ..0.002

PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

greater loads, higher speeds and lower tire pressure.

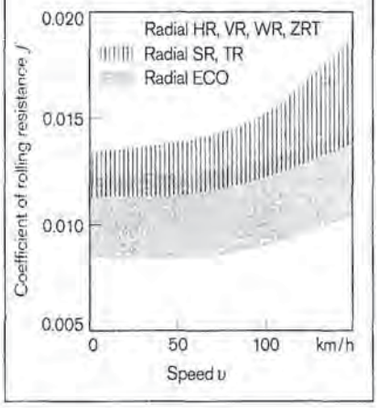
During cornering, the rolling resistance is augmented by the

cornering resistance

$$F_K = f_K \cdot G$$

The coefficient of cornering resistance f_K is a function of vehicle speed, curve radius, suspension geometry, tires, tire pressure, and the vehicle's response under lateral acceleration.

Rolling resistance of radial car tires on smooth, level road surfaces under normal load and at correct tire pressure



Aerodynamic drag

Aerodynamic drag is calculated as:

$$F_L = 0.5 \cdot \varrho \cdot c_d \cdot A \, (v + v_0)^2$$

With v in km/h:

$$F_L = 0.0386 \cdot \varrho \cdot c_d \cdot A \cdot (v + v_0)^2$$

Air density ϱ
(at 200 m altitude ϱ = 1.202 kg/m³),
Drag coefficient c_d PP. 37 and 340.

Aerodynamic drag in kW

$$P_L = F_L \cdot v = 0.5 \cdot \varrho \cdot c_d \cdot A \cdot v \cdot (v + v_0)^2$$

or

$$P_L = 12.9 \cdot 10^{-6} \cdot c_d \cdot A \cdot v \cdot (v + v_0)^2$$

with P_L in kW, F_L in N, v and v_0 in km/h, A in m², ϱ = 1.202 kg/m³.

Empirical determination of coefficients for aerodynamic drag and rolling resistance

Allow vehicle to coast down in neutral under windless conditions on a level road surface. The time that elapses while the vehicle coasts down by a specific increment of speed is measured from two initial velocities, v_1 (high speed) and v_2 (low speed). This information is used to calculate the mean deceleration rates a_1 and a_2. See following table for formulas and example.

The example is based on a vehicle weighing m = 1450 kg with a cross section A = 2.2 m².

The method is suitable for application at vehicle speeds of less than 100 km/h.

	1st Trial (high speed)	2nd Trial (low speed)
Initial velocity Terminal velocity Interval between v_a and v_b	v_{a1} = 60 km/h v_{b1} = 55 km/h t_1 = 6.5 s	v_{a2} = 15 km/h v_{b2} = 10 km/h t_2 = 10.5 s
Mean velocity	$v_1 = \frac{v_{a1} + v_{b1}}{2} = 57.5$ km/h	$v_2 = \frac{v_{a2} + v_{b2}}{2} = 12.5$ km/h
Mean deceleration	$a_1 = \frac{v_{a1} - v_{b1}}{t_1} = 0.77 \frac{\text{km/h}}{\text{s}}$	$a_2 = \frac{v_{a2} - v_{b2}}{t_2} = 0.48 \frac{\text{km/h}}{\text{s}}$
Drag coefficient	$c_d = \frac{6\,m \cdot (a_1 - a_2)}{A \cdot (v_1^2 - v_2^2)} = 0.36$	
Coefficient of rolling resistance	$f = \frac{28.2\,(a_2 \cdot v_1^2 - a_1 \cdot v_2^2)}{10^3 \cdot (v_1^2 - v_2^2)} = 0.013$	

PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

Drag coefficient and associated power requirements for various body configurations

		Drag coefficient	Drag power in kW, average values for A = 2 m² at various speeds[1])			
		c_d	40 km/h	80 km/h	120 km/h	160 km/h
	Open convertible	0.5...0.7	1	7.9	27	63
	Station wagon (2-box)	0.5...0.6	0.91	7.2	24	58
	Conventional form (3-box)	0.4...0.55	0.78	6.3	21	50
	Wedge shape, headlamps and bumpers integrated into body, wheels covered, underbody covered, optimized flow of cooling air.	0.3...0.4	0.58	4.6	16	37
	Headlamps and all wheels enclosed within body, underbody covered	0.2...0.25	0.37	3.0	10	24
	Reversed wedge shape (minimal cross-section at tail)	0.23	0.38	3.0	10	24
	Optimum streamlining	0.15...0.20	0.29	2.3	7.8	18
Trucks, truck-trailer combinations		0.8...1.5	–	–	–	–
Motorcycles		0.6...0.7	–	–	–	–
Buses		0.6...0.7	–	–	–	–
Streamlined buses		0.3...0.4	–	–	–	–

1) No headwind ($v_0 = 0$).

PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

Climbing resistance and downgrade force

Climbing resistance (F_{St} with positive operational sign) and downgrade force (F_{St} with negative operational sign) are calculated as:

$$F_{St} = G \cdot \sin \alpha$$
$$= m \cdot g \cdot \sin \alpha$$

or, for a working approximation:

$$F_{St} \approx 0.01 \cdot m \cdot g \cdot p$$

valid for gradients up to $p \leq 20\%$, as $\sin \alpha \approx \tan \alpha$ at small angles (less than 2% error).

Climbing power is calculated as:

$$P_{St} = F_{St} \cdot v$$

with P_{St} in kW, F_{St} in N, v in km/h:

$$P_{St} = \frac{F_{St} \cdot v}{3600} = \frac{m \cdot g \cdot v \cdot \sin \alpha}{3600}$$

or, for a working approximation:

$$P_{St} \approx \frac{m \cdot g \cdot p \cdot v}{360{,}000}$$

The gradient is:

$$p = (h/l) \cdot 100\ \% \text{ or}$$
$$p = (\tan \alpha) \cdot 100\ \%$$

with h as the height of the projected distance l. In English-speaking countries, the *gradient* is calculated as follows:

Conversion:

Gradient 1 in $100/p$

Example: "1 in 2".

Gradient angle α	Incline p %	Incline *Gradient*	Climbing resistance at m = 1000 kg N
45°	100	1 in 1	6500
40°	90		6000
35°	80		5500
30°	70	1 in 1.5	5000
25°	60		4500
20°	50	1 in 2	4000
15°	40	1 in 2.5	3500
10°	30	1 in 3	3000
5°	20	1 in 4	2500
0°	10	1 in 5	2000
	0	1 in 10	1500
		1 in 20	1000
		1 in 50	500
		1 in 100	0

Values at m = 1000 kg					
Climbing resistance F_{St} N	Climbing power P_{St} in kW at various speeds				
	20 km/h	30 km/h	40 km/h	50 km/h	60 km/h
6500	36	54	72	–	–
6000	33	50	67	–	–
5500	31	46	61	–	–
5000	28	42	56	69	–
4500	25	37	50	62	–
4000	22	33	44	56	67
3500	19	29	39	49	58
3000	17	25	33	42	50
2500	14	21	28	35	42
2000	11	17	22	28	33
1500	8.3	12	17	21	25
1000	5.6	8.3	11	14	17
500	2.3	4.2	5.6	6.9	8.3
0	0	0	0	0	0

Example: To climb a hill with a gradient of $p = 18\%$, a vehicle weighing 1500 kg will require approximately $1.5 \cdot 1700$ N = 2550 N motive force and, at v = 40 km/h, roughly $1.5 \cdot 19$ kW = 28.5 kW climbing power.

PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

Motive force

The higher the engine torque M and overall transmission ratio i between engine and driven wheels, and the lower the power-transmission losses, the higher is the motive force F available at the drive wheels.

$$F = \frac{M \cdot i}{r} \cdot \eta \quad \text{or} \quad F = \frac{P \cdot \eta}{v}$$

η Drivetrain efficiency level
(longitudinally installed engine
$\eta \approx 0.88...0.92$)
(transverse engine $\eta \approx 0.91...0.95$)

The motive force F is partially consumed in overcoming the running resistance F_W. Numerically higher transmission ratios are applied to deal with the substantially increased running resistance encountered on gradients (gearbox).

Vehicle and engine speeds

$$n = \frac{60 \cdot v \cdot i}{2 \cdot \pi \cdot r}$$

or with v in km/h:

$$n = \frac{1000 \cdot v \cdot i}{2 \cdot \pi \cdot 60 \cdot r}$$

Acceleration

The surplus force $F - F_W$ accelerates the vehicle (or retards it when F_W exceeds F).

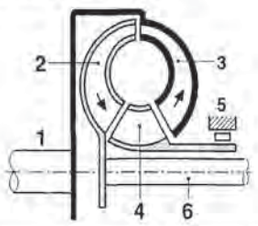
$$a = \frac{F - F_W}{k_m \cdot m} \quad \text{or} \quad a = \frac{P \cdot \eta - P_W}{v \cdot k_m \cdot m}$$

The rotational inertia coefficient k_m, compensates for the apparent increase in vehicle mass due to the rotating masses (wheels, flywheel, crankshaft, etc.).

Motive force and road speed on vehicles with automatic transmissions

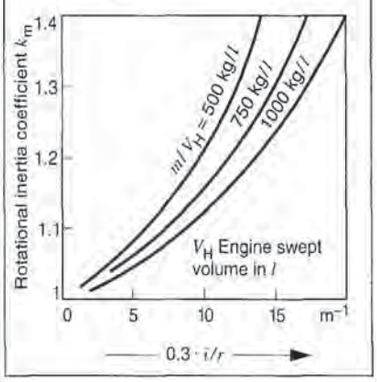
When the formula for motive force is applied to automatic transmissions with hydrodynamic torque converters or hydrodynamic clutches, the engine torque M is replaced by the torque at the converter turbine, while the rotational speed of the converter turbine is used in the formula for engine speed.

Hydrodynamic converter
1 Power, 2 Turbine, 3 Pump, 4 Stator, 5 One-way clutch, 6 Output

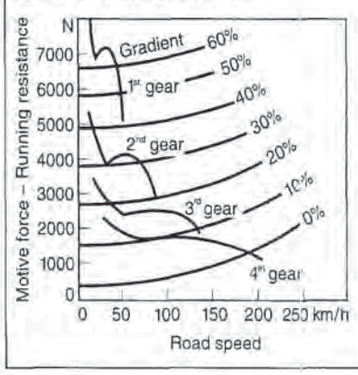


The relationship between $M_{Turb} = f(n_{Turb})$ and the engine characteristic $M_{Mot} = f(n_{Mot})$ is determined using the characteristics of the hydrodynamic converter (P. 635).

Determining the rotational inertia coefficient k_m



Running diagram for car with automatic transmission and hydrodynamic trilok converter under full throttle



PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

Adhesion to road surface

Coefficients of static friction for pneumatic tires on various surfaces

Vehicle speed km/h	Tire condition	Road condition Dry	Wet: water approx. 0.2 mm	Deep rainfall: water approx. 1 mm deep	Puddles: water approx. 2 mm deep	Iced (black ice)
		Coefficient of static friction μ_{HF}				
50	new	0.85	0.65	0.55	0.5	0.1 and less
	worn[1])	1	0.5	0.4	0.25	
90	new	0.8	0.6	0.3	0.05	
	worn[1])	0.95	0.2	0.1	0.05	
130	new	0.75	0.55	0.2	0	
	worn[1])	0.9	0.2	0.1	0	

The static coefficient of friction (between the tires and the road surface), also known as the tire-road-interface friction coefficient, is determined by the vehicle's speed, the condition of the tires and the state of the road surface (see table above). The figures cited apply for concrete and tarmacadam road surfaces in good condition. The coefficients of sliding friction (with wheel locked) are usually lower than the coefficients of static friction.

Special rubber compounds providing friction coefficients of up to 1.8 are employed in racing tires.

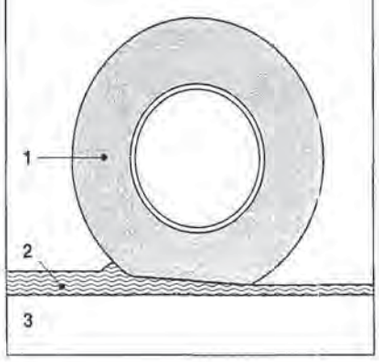
The maxima for acceleration and uphill driving, and for retardation and downhill braking, are provided on page 345.

Aquaplaning

Aquaplaning, has a particularly dramatic influence on the contact between tire and road surface. It describes the state in which a layer of water separates the tire and the (wet) road surface. The phenomenon occurs when a wedge of water forces its way underneath the tire's contact patch and lifts it from the road. The tendency to aquaplan is dependent upon such factors as the depth of the water on the road surface, the vehicle's speed, the tread pattern, the tread wear, and the load pressing the tire against the road surface. Wide tires are particularly susceptible to aquaplaning. It is not possible to steer or brake an aquaplaning vehicle, as its front wheels have ceased to rotate, meaning that neither steering inputs nor braking forces can be transmitted to the road surface.

[1]) Worn to tread depth of ≥1.6 mm (legal minimum in Germany, as per para. 36.2, StVZO (FMVSS/CUR)).

Aquaplaning
1 Tire, 2 Wedge of water, 3 Road surface.



PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

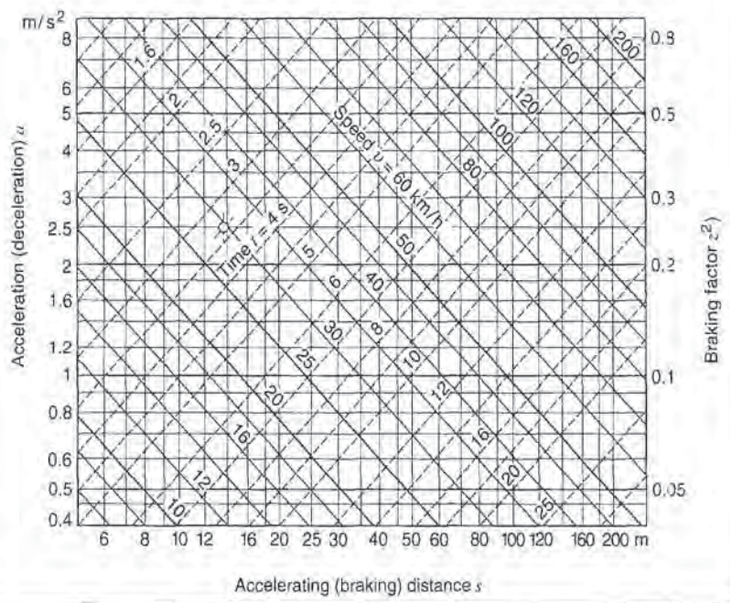
Accelerating and braking

The vehicle is regarded as accelerating or braking (decelerating) at a constant rate when a remains constant. The following equations apply to an initial or final speed of 0:

	Equations for v in m/s	Equations for v in km/h
Acceleration or braking (deceleration) in m/s²	$a = \frac{v^2}{2 \cdot s} = \frac{v}{t} = \frac{2 \cdot s}{t^2}$	$a = \frac{v^2}{26 \cdot s} = \frac{v}{3.6 \cdot t} = \frac{2 \cdot s}{t^2}$
Accelerating or braking time in s	$t = \frac{v}{a} = \frac{2 \cdot s}{v} = \sqrt{\frac{2 \cdot s}{a}}$	$t = \frac{v}{3.6 \cdot a} = \frac{7.2 \cdot s}{v} = \sqrt{\frac{2 \cdot s}{a}}$
Accelerating or braking distance[1] in m	$s = \frac{v^2}{2 \cdot a} = \frac{v \cdot t}{2} = \frac{a \cdot t^2}{2}$	$s = \frac{v^2}{26 \cdot a} = \frac{v \cdot t}{7.2} = \frac{a \cdot t^2}{2}$

Stopping distance, P. 346.
Symbols and units, P. 15.

Constant rates of acceleration and deceleration



Each point on the graph represents a particular relationship between v, a or z^2), s and t. Two values must be available for all values to be determined.
Given: Vehicle speed v = 30 km/h, braking distance s = 13.5 m.
Determined: Mean deceleration a = 2.5 m/s², retardation z = 0.25, Braking time t = 3.3 s.

[1]) If final speed v_2 is not 0, braking distance $s = v_1 \cdot t - at^2/2$ at v_1 in m/s.
[2]) Deceleration rate relative to 1 g.

PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

Maxima for acceleration and braking (deceleration)

When the motive or braking forces exerted at the vehicle's wheels reach such a magnitude that the tires are just still within their limit of adhesion (maximum adhesion is still present), the relationships between the gradient angle α, coefficient of static friction μ_{HF}[1]) and maximum acceleration or deceleration are defined as follows. The real-world figures are always somewhat lower, as all the vehicle's tires do not simultaneously exploit their maximum adhesion during each acceleration (deceleration). Electronic traction control and antilock braking systems (TCS, ABS, ESP)[2]) maintain the traction level in the vicinity of the coefficient of static friction.

k = Ratio between the load on driven or braked wheels and the total weight. All wheels driven or braked: $k = 1$. At 50 % weight distribution $k = 0.5$.
Example: $k = 0.5$; $g = 10$ m/s²,
$\mu_{HF} = 0.6$; $p = 15\,\%$
$a_{max} = 0.5 \cdot 10 \cdot (0.6 \pm 0.15)$
Braking on upgrade (+): $a_{max} = 3.75$ m/s²,
downgrade braking (–): $a_{max} = 2.25$ m/s²

Work and power

The power required to maintain a consistent rate of acceleration (deceleration) varies according to vehicle speed. Power available for acceleration:

$$P_a = P \cdot \eta - P_W$$

where P = engine output, η = efficiency, and P_W = motive power.

Acceleration and braking (deceleration)

	Level road surface	Inclined road surface α°; $p = 100 \cdot \tan\alpha\,\%$	
Limit acceleration or deceleration a_{max} in m/s²	$a_{max} \le k \cdot g \cdot \mu_r$	$a_{max} \le k \cdot g\,(\mu_r \cos\alpha \pm \sin\alpha)$ approximation[3]): $a_{max} \le k \cdot g\,(\mu_r \pm 0.01 \cdot p)$	+ Upgrade braking or downgrade acceleration – Upgrade acceleration or downgrade braking

Achievable acceleration a_e (P_a in kW, v in km/h, m in kg)

Level road surface	Inclined road surface	
$a_e = \dfrac{3600 \cdot P_a}{k_m \cdot m \cdot v}$	$a_e = \dfrac{3600 \cdot P_a}{k_m \cdot m \cdot v} \pm g \cdot \sin\alpha$	+ Downgrade acceleration – Upgrade acceleration for $g \cdot \sin\alpha$ the approximation[3]) is $g \cdot p/100$

Work and power

	Level road surface	Inclined road surface α°; $p = 100 \cdot \tan\alpha\,\%$	
Acceleration or braking work W in J[4])	$W = k_m \cdot m \cdot a \cdot s$	$W = m \cdot s\,(k_m \cdot a \pm g \cdot \sin\alpha)$ approximation[3]): $W = m \cdot s\,(k_m \cdot a \pm g \cdot p/100)$	+ Downgrade braking or upgrade acceleration – Downgrade acceleration or upgrade braking
Acceleration or braking power at velocity v in W	$P_a = k_m \cdot m \cdot a \cdot v$	$P_a = m \cdot v\,(k_m \cdot a \pm g \cdot \sin\alpha)$ approximation[3]): $P_a = m \cdot v\,(k_m \cdot a \pm g \cdot p/100)$	v in m/s. For v in km/h, use $v/3.6$.

[1]) See P. 343 for numerical values.
[2]) ABS: antilock braking system; P. 659, 693.
TCS: traction control system; P. 606.
ESP: electronic stability program; P. 704.
[3]) Valid to approx. $p = 20\,\%$ (under 2 % error).
[4]) J = N · m = W · s; conversions, P. 16.

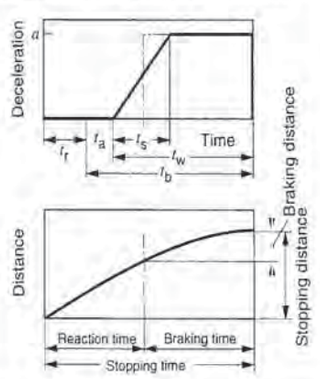
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US PATENT NO. 6,581,565

Stopping distance

This is the distance covered between the moment when a hazard is recognized and the time when the vehicle comes to a complete stop. It is the sum of the distances traveled during the reaction time t_r, the brake response time t_a (at constant vehicle speed v), and the active braking time t_w. Maximum retardation a is obtained in the pressure-build-up period t_s. Alternately, half of the pressure-buildup period may be considered to be fully decelerated. The periods in which no braking occurs are combined to form the so-called lost time t_{vz}:

$$t_{vz} = t_r + t_a + t_s/2.$$

Deceleration and stopping distances



The upper limits on retardation are determined by the static coefficient of friction between tires and road surface (P. 343), while the lower extremes are defined by the legally required minima (P. 650).

The difference between the stopping time/distance and the braking time/distance is defined by t_{vz} or $v \cdot t_{vz}$. The brake actuation time given on page 647 includes the times $t_a + t_s/2$.

Reaction time

The reaction time is the period which elapses between recognition of the object, the decision to brake and the time it takes for the foot to hit the brake pedal. The reaction time is not a fixed value: it ranges from 0.3 to 1.7 s, depend ng upon the driver and on external factors (P. 347).

Special tests are necessary to determine individual reaction patterns (such as those performed by the Medical and Psychological Institute for Traffic Safety of the German Inspection, Testing, and Certification Authorities, Stuttgart [TÜV Stuttgart, e.V.]).

	Equations for v in m/s	Equations for v in km/h
Stopping time t_h in s	$t_h = t_{vz} + \frac{v}{a}$	$t_h = t_{vz} + \frac{v}{3.6 \cdot a}$
Stopping distance s_h in m	$s_h = v \cdot t_{vz} + \frac{v^2}{2 \cdot a}$	$s_h = \frac{v}{3.6} t_{vz} + \frac{v^2}{26 \cdot a}$

Personal factors influencing reaction time

	Psycho-physical reaction ←			→ Muscular reaction	
Perceived object (i.e. road sign) →	Perceive	Perceive	Decide	Incipient response	Move → Actuated control element (brake pedal)
	Acuity of vision	Perception and understanding	Mental processing	Locomotor system	Personal response time

PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

Effect of personal and extraneous factors on reaction time

	Reduction down to 0.3 s ◄	► **Extension up** to 1.7 s
A) Personal factors	Trained, reflexive response	Inexperienced, uncoordinated response
	Good frame of mind; optimum performance potential	Poor frame of mind, e.g. fatigue
	Highly skilled driver	Lower level of driving skill
	Youth	Advanced age
	Anticipation	Inattentiveness, distraction
	Physical and mental health	Health disorders in response groups
		Panic, alcohol
B) Extraneous factors Traffic situation	Uncomplicated, easily comprehended, predictable, familiar	Complicated, difficult to comprehend, unpredictable, rarely encountered
Type of perceived object	Explicit, conspicuous	Equivocal, inconspicuous
Location of perceived object	Within field of vision	At edge of field of vision
Nature of control element	Logical layout of mechanic controls	Poor layout of mechanical controls

Brake response and pressure build-up times

The condition of the brakes at the moment at which they are applied (e.g. wet brake disks) is one factor influencing brake response time t_a and pressure build-up time t_s; others are the designs of the actuation and force-transmission mechanism and linkage. Consult ECE R13 for legal regulations concerning t_a and t_s on tightly adjusted brakes. The applicable regulation[1]) for testing braking-system effectiveness uses figures of 0.36 s (vehicle class M_1) and 0.54 s (vehicle classes M_2, M_3, N_1...N_3) in the braking-distance equation $t_a + t_s/2$, see P. 717). The response and pressure build-up times are longer if the brake system is in poor condition. A response delay of 1 s results in a stopping distance of (see table):

[1]) EC Directive 71J320/EEC.

Deceleration a in m/s²	**Vehicle speed prior to braking in km/h**												
	10	30	50	60	70	80	90	100	120	140	160	180	200
	Distance during delay of 1 sec (no braking) in m												
	2.8	8.3	14	17	19	22	25	28	33	39	44	50	56
	Stopping distance in m												
4.4	3.7	16	36	48	62	78	96	115	160	210	270	335	405
5	3.5	15	33	44	57	71	87	105	145	190	240	300	365
5.8	3.4	14	30	40	52	65	79	94	130	170	215	265	320
7	3.3	13	28	36	46	57	70	83	110	145	185	230	275
8	3.3	13	26	34	43	53	64	76	105	135	170	205	250
9	3.2	12	25	32	40	50	60	71	95	125	155	190	225

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US PATENT NO. 6,581,565

Dynamics of lateral motion

Response to crosswinds

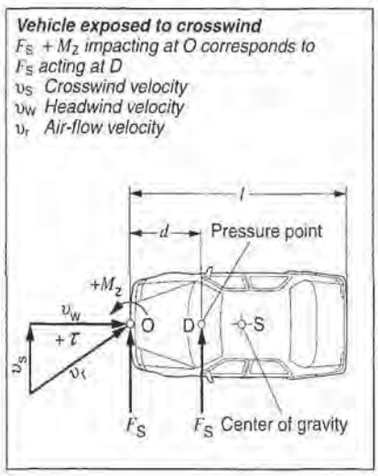
Strong crosswinds will deflect a vehicle from its lane, particularly at high speeds. Sudden crosswinds, of the kind encountered when embankments give way to open, unsheltered roads, can evoke yaw-angle changes as well as causing the vehicle to drift off course. On vehicles which are sensitive to crosswind (for reaction time see page 346), these effects are substantial and can occur before the driver has had time to react and take preventive action. It is thus essential that the relevant factors be considered in the vehicle design stage.

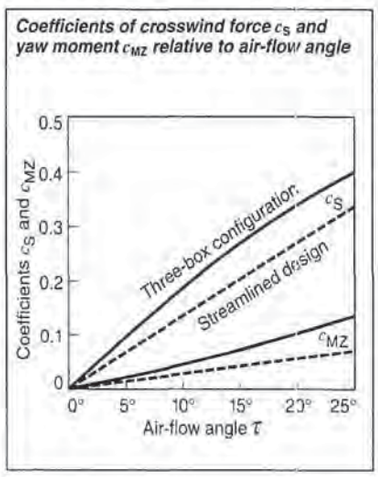
When wind impacts upon the vehicle from an oblique angle, aerodynamic drag is supplemented by a lateral component of the aerodynamic force. This force, which is distributed across the entire body, can be considered as a single force – the crosswind force – directed toward what is known as the pressure point. The actual location of the pressure point is determined by body shape and angle of impact.

The pressure point will usually be found on the front half of the vehicle, whereby it is closer to the center on conventional notchback ("three-box") bodies than on vehicles with a sloping tail section ("fast-backs" and aerodynamic designs). With the latter, the pressure point may even lie forward of the front axle in extreme cases. The pressure point on notchback bodies generally remains in a specific location; on streamlined vehicles it tends to vary according to the angle of impact. All else being equal, the greater lateral forces will occur with the notchback.

As the above implies, the pressure point will hardly provide a suitable reference point for use in illustrating the effects of crosswinds, because it has no fixed location on the vehicle. At the same time, the center of gravity varies with vehicle load. It is thus useful to select a central location on the forward section of the body as a reference point, a practice which also ensures that the data remain unaffected by the variations in the relative positions of bodywork and suspension.

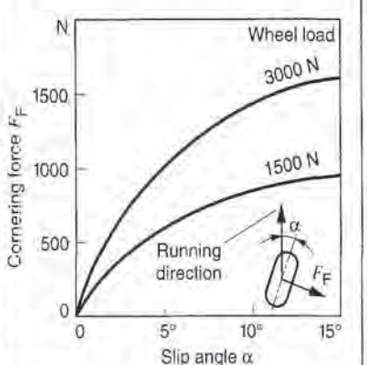
When the crosswind force is specified for a reference point other than the pressure point, it must be supplemented by the crosswind moment at the pressure point in question. The standard practice in aerodynamics is to replace the forces and moments with dimensionless coefficients which remain independent of air-flow velocity. The graph shows the curves for crosswind force and the yaw coefficients for both a notchback vehicle and a fastback body style. Due to the multiplicity of possible body designs, the present investigation can only provide general refer-



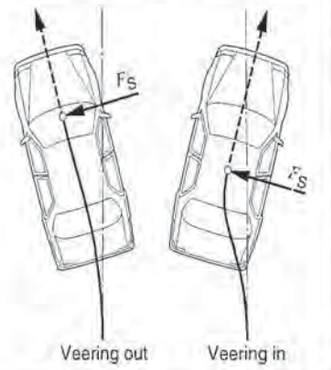


PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

Cornering force as a function of wheel-slip angle



Crosswind impacting against center of gravity. Vehicle veers in or veers out



ence data; wind-tunnel measurements are essential for obtaining precise information on specific body shapes.

The following equations are applied to derive forces and moments from the coefficients:

Crosswind force $F_S = c_S \cdot \varrho \cdot (v_r^2/2) \cdot A_S$

Yaw moment $M_Z = c_{MZ} \cdot \varrho \cdot (v_r^2/2) \cdot A_S \cdot l$

ϱ Atmospheric density, v_r Resulting air-impact velocity, A_s Projected lateral surface of the vehicle, l Total length of the vehicle (in coherent units), c_S Coefficient of crosswind force, c_{MZ} Coefficient for crosswind-induced yaw.

The distance between the pressure point and the leading edge of the vehicle is calculated using

$$d = M_Z/F_S = l \cdot c_{MZ}/c_S$$

The crosswind force is resisted by the lateral cornering forces at the wheels. The degree of lateral cornering force which a pneumatic tire can provide depends upon numerous factors, such as wheel slip angle, wheel load, tire design and dimensions, tire pressure and the amount of traction afforded by the road surface.

The pneumatic tire's specific response characteristics mean that, at a constant slip angle, higher wheel loads will not evoke proportional increases in cornering forces. In the example (diagram), doubling the wheel load increases the cornering force by a factor of only 1.5...1.7. The slip angle must also be increased if the cornering force is to be doubled. This explains why the axle supporting the greater load assumes a larger slip angle than its less heavily loaded counterpart, assuming that identical ratios of lateral (side) force to tire contact force act on both axles.

When the crosswind force impacts against the center of gravity, the distribution ratio for the required cornering forces between front and rear axles is a function of the wheel-load factors, i.e. the ratio of contact force to lateral force is the same at both axles. When the center of gravity is in the forward section of the vehicle, the crosswind force causes the vehicle to "veer-out" due to the front axle's necessarily greater wheel slip angle; when the center of gravity is at the rear, the vehicle "veers-in". In both cases, the vehicle's center of gravity is deflected downwind from the original path of travel. When the crosswind ceases, the "veering-out" vehicle will continue to deviate from its original path, while the "veering-in" vehicle will return to and indeed go beyond the original line.

The crosswind's impact point, however, will rarely coincide with the center of gravity; the distance forward from the center of the vehicle depends upon the individual body shape. Thus a vehicle which responds to impact at the center of gravity

PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

by veering inwards, will veer out when exposed to actual crosswinds. Due to the forward displacement of the pressure point, vehicles with a sloping rear section display a greater tendency to veer-out; they are thus more difficult to control than notchback vehicles.

Rear-mounted vertical stabilizer fins can be installed to move the pressure point rearward, toward the center of gravity. However, except in competition vehicles and other special applications, this option has to date rarely been applied.

Placing the pressure point in the immediate vicinity of the vehicle's center of gravity results in improved directional stability. Oversteer vehicles have a minimum tendency to deviate from their path of travel when the pressure point is forward of the center of gravity. On understeer vehicles, the best pressure-point location is just to the rear of the center of gravity.

Crosswind force is generally minimal relative to contact force, and the influence of non-linearity in the ratio of contact force to lateral force is limited when small lateral forces are applied – a vehicle's response to crosswinds is therefore not governed by tire properties alone. Other important factors include compliance in suspension and steering. Thus the suspension designer can also exercise a major influence on the vehicle's crosswind response characteristics.

Oversteer and understeer

Cornering forces can only be generated between a rubber tire and the road surface when the tire rolls at an angle to its longitudinal plane; that is, a certain wheel slip angle is required.

A vehicle is said to understeer when, as lateral acceleration increases, the slip angle at the front axle increases more than it does at the rear axle. The opposite applies for a vehicle which oversteers.

A vehicle will not necessarily display the same self-steering effect at all possible rates of lateral acceleration. Some vehicles always understeer or oversteer, and some display a transition from understeer to oversteer as lateral acceleration increases, while other vehicles respond in precisely the opposite manner.

References:
Barth, R.: "Windkanalmessungen an Fahrzeugmodellen und rechteckigen Körpern mit verschiedenem Seitenverhältnis bei unsymmetrischer Anströmung" (Wind Tunnel Measurements on Vehicle Models and Rectangular Bodies with Different Side Ratios under Asymmetrical Flow). Dissertation, Stuttgart Technical College, 1958.
Barth, R.: "Luftkräfte am Kraftfahrzeug" (Aerodynamic Forces Acting on Motor Vehicles). Deutsche Kraftfahrtforschung und Straßenverkehrstechnik (German Motor Vehicle Research and Traffic Engineering), Vol. 184, Düsseldorf, VDI-Verlag, 1966.
Mitschke, M.: "Dynamik der Kraftfahrzeuge" (Dynamics of Motor Vehicles). Springer-Verlag, 1972.

Cornering behavior

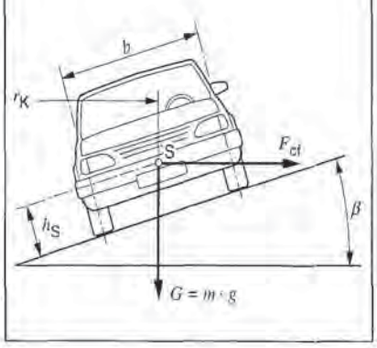
Centrifugal force in curves

$$F_{cf} = \frac{m \cdot v^2}{r_K} \text{ (P. 35)}$$

or, with v specified in km/h and r_K as the radius of the curve in m

$$F_{cf} = \frac{m \cdot v^2}{12.96 \cdot r_K}$$

Centrifugal force when cornering
b Track width in m,
h_S Height of center of gravity in m.



PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

Critical speeds

The concrete example provided in the table below would apply to a vehicle with a track $b = 1.5$ m, height of center of gravity $h_S = 0.6$ m; max. adhesion coefficient $\mu_r = 0.8$, curve radius $r_K = 90$ m, curve banking $\beta = 20°$.

	Flat curve	Banked curve
Speed at which the vehicle exceeds the limit of adhesion (skid)	$v \leq 11.28 \sqrt{\mu_r \cdot r_K}$ km/h	$v \leq 11.28 \sqrt{\frac{(\mu_r + \tan\beta)\, r_K}{1 - \mu_r \cdot \tan\beta}}$ km/h
Example:	≤ 96 km/h	≤ 137 km/h
Speed at which the vehicle tips	$v \geq 11.28 \sqrt{\frac{b \cdot r_K}{2 \cdot h_S}}$ km/h	$v \geq 11.28 \sqrt{\frac{\left(\frac{b}{2 \cdot h_S} + \tan\beta\right) \cdot r_K}{1 - \frac{b}{2 \cdot h_S} \cdot \tan\beta}}$ km/h
Example:	≥ 120 km/h ($\mu_r \geq 1.25$)	≥ 184 km/h ($\mu_r \geq 1.25$)

Body roll in curves

Roll axis

When cornering, the centrifugal force which concentrates around the center of gravity causes the vehicle to tilt away from the path of travel. The magnitude of this rolling motion depends upon the rates of the springs and their response to alternating compression, and upon the lever arm of the centrifugal force (distance between the roll axis and the center of gravity). The roll axis is the body's instantaneous axis of rotation relative to the road surface. Like all rigid bodies, the vehicle body consistently executes a screwing or rotating motion; this motion is supplemented by a lateral displacement along the instantaneous axis.

The higher the roll axis, i.e. the closer it is to a parallel axis through the center of gravity, the greater will be the transverse stability and the less the roll during cornering. However, this generally implies a corresponding upward displacement of the wheels, resulting in a change in track width (with negative effects on operating safety). For this reason, designs are sought which combine a high instantaneous roll center with minimal track change. The goal therefore is to place the instantaneous axes of the wheels as high as possible relative to the body, while simultaneously keeping them as far from the body as possible.

Determining the roll axis

A frequently-applied means for finding the approximate roll axis is based on determining the centers of rotation of an equivalent body motion. This body motion takes place in those two planes through the front and rear axles which are vertical relative to the road. The centers of rotation are those (hypothetical) points in the body which remain stationary during the rotation. The roll axis, in turn, is the line which connects these centers (instantaneous centers). Graphic portrayals of the instantaneous centers are based on a rule according to which the instantaneous centers of rotation of three systems in a state of relative motion lie on a common pole line.

The complexity of the operations required for a more precise definition of the spatial relationships involved in wheel motion makes it advisable to employ computers capable of carrying out calculations in matrix notation (vector algebra, linear geometry, etc.).

References:

E. v. d. Osten-Sacken: "Berechnung allgemein räumlicher, vielgliedriger Gelenkgetriebe"; Dissertation, Aachen Technical College, 1970.

G. Prigge: "Grundsätzliches der Ermittlung der Rollachse von Kraftfahrzeugen mit räumlichen Radführungen"; Dissertation, Braunschweig Technical University, 1972.

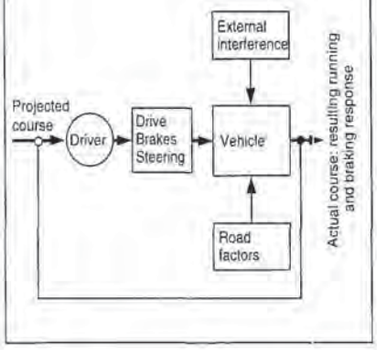
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US PATENT NO. 6,581,565

ISO procedures for evaluating vehicle handling

The science devoted to studying the dynamics of vehicle handling generally defines its subject as the overall behavior of the entire system represented by "driver + vehicle + environment". As the first link in the chain, the driver makes judgements on the vehicle's handling qualities based on the sum of diverse subjective impressions. On the other hand, handling data derived from specific driving maneuvers executed without driver input (open-loop operation) provide an objective description of the vehicle's handling qualities. The driver, who up to the present day still cannot be defined accurately in terms of behavior, is replaced in these tests by a specific, objectively quantifiable interference factor. The resulting vehicular response can then be analyzed and discussed.

Standardized versions of the driving maneuvers in the list (performed on a dry road surface) below have either already been defined by the ISO or are under consideration; they serve as recognized standard procedures for vehicular evaluation [1], [2]:
- Steady-state skidpad [3],
- Transient response [4], [5], [6],
- Braking during cornering [7],
- Crosswind sensitivity,
- Straight-running stability, and
- Reaction to throttle change on skidpad.

"Driver-vehicle-environment" synergism represented as a closed-loop control system

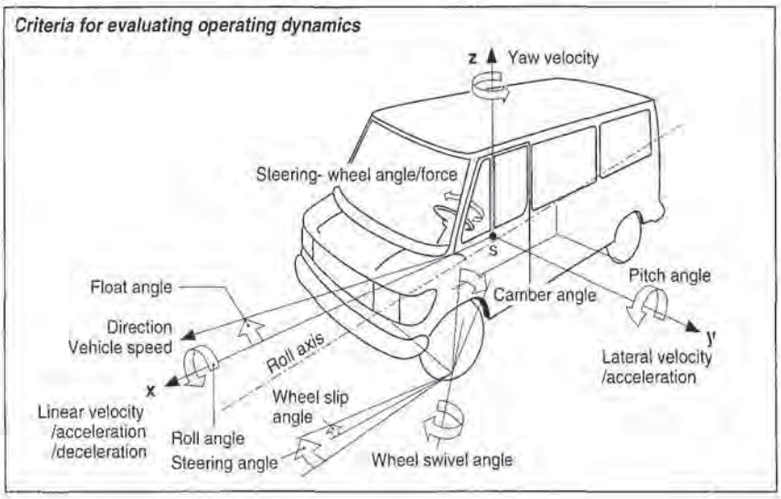


To date, it has still not been possible to arrive at comprehensive objective definitions for the dynamic characteristics associated with closed-loop operation, as adequate data on the precise control characteristics of the human element are still unavailable.

Test quantities

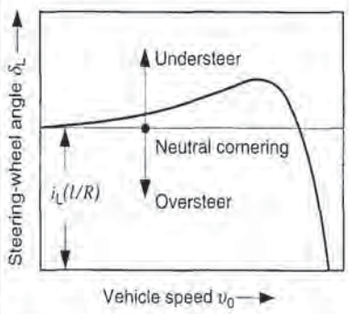
The main criteria employed in evaluating vehicle dynamics are:
- Steering-wheel angle,
- Lateral acceleration,
- Longitudinal acceleration and deceleration,
- Yaw speed,
- Float and roll angles.

Criteria for evaluating operating dynamics



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US PATENT NO. 6,581,565

Definitions of attitude in steady-state cornering (according to Mitschke [10])
i_L Steering ratio, l Wheelbase, R (= const.) Cornering radius, $i_L (l/R)$ Ackermann effect.



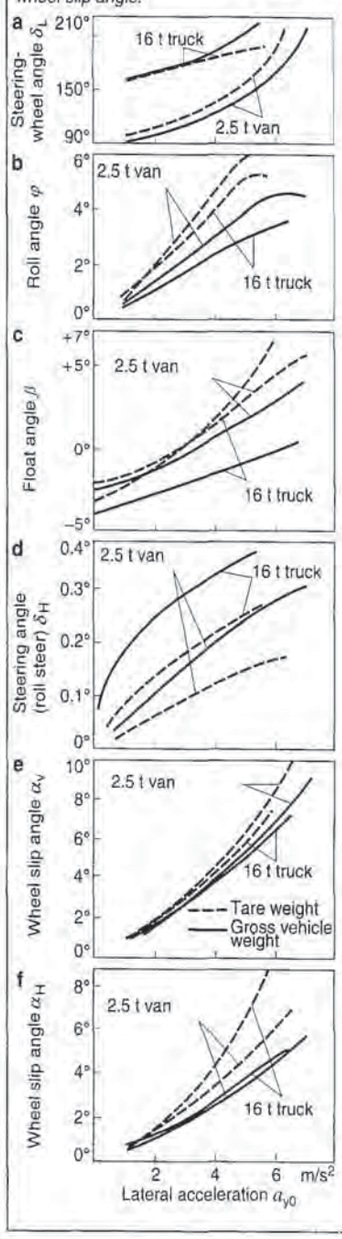
Additional data are employed to verify and confirm the previously derived information on specific points of vehicle handling:
- Linear and lateral velocity,
- Steering angles at front and rear wheels,
- Slip angles at all wheels,
- Camber and pitch angles,
- Steering-wheel force.

Steady-state skidpad

The most important data to be derived from steady-state skidpad testing are the maximum lateral acceleration, and the manner in which individual dynamic parameters respond to the variations in lateral acceleration which occur up to the limit. This information is employed in evaluating the vehicle's self-steering response [2], [3]. Compliance in both the steering system and the suspension is represented in the current standard definition of steering response, which employs the terms "oversteer", "understeer" and "neutral steering". Several dynamic factors and their derivatives are considered in conjunction with lateral acceleration in describing vehicle handling, examples being steering-wheel angle, roll angle and float angle. Other significant vehicle parameters are steering angle and slip angle.

In the following, a light utility van (similar to a passenger car) and a heavy truck are used to provide examples of the results gathered on a dry surface.

Steady-state skidpad *(42 m radius)*
a) Steering-wheel angle, b) Roll angle, c) Float angle, d) Rear-axle steering angle (roll steer), e) Front-wheel slip angle, f) Rear-wheel slip angle.



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US PATENT NO. 6,581,565

Steering-wheel angle
As small utility vehicles are fitted with tires similar to those used in passenger cars, and are also equipped with relatively high-power engines, they achieve high rates of lateral acceleration. Both vehicle types understeer.

Roll angle
The degree of self-steering which prevails at the axles is largely determined by the roll angle. Higher loads result in more pronounced roll angles due to the greater vehicle mass and the attendant increase in effective centrifugal force.

Float angle
The float angle encountered at high rates of lateral acceleration is regarded as an index of controllability, in other words the vehicle's response to driver input. High absolute figures or fluctuations in float angle are regarded as particularly undesirable [8].

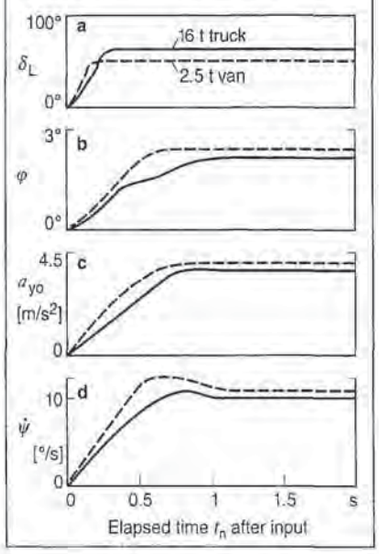
At low rates of lateral acceleration, the float angle is a function of the radius of the driven circle, and the vehicle's center of gravity which varies along with changes in vehicle load.

Rear-axle roll steer
The relationship between the steering angle at the rear axle (roll-steer angle) and lateral acceleration illustrates how the roll-steer angle at the rear axle decreases in response to higher vehicle loads.

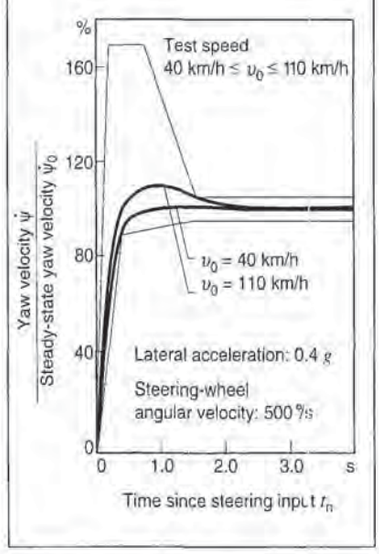
Slip angle
The wheel slip angles at the individual wheels provide information on the vehicle's self-steering characteristics. The wheel slip angles increase in response to higher vehicle loads, a consequence of the tires requiring greater slip angles as their loading increases [2], [9].

Transient response
In addition to self-steering in steady-state testing, yet another significant factor is the vehicle's response during directional changes (e.g. for rapid evasive maneuvers) [2]. Two test methods have become accepted internationally. These are defined according to the type of input stimulus, and illustrate both the time and frequency ranges of the vehicle's response:
– Step-input (sharp change in steering

Sudden steering-angle change *(unloaded, v_0 = 60 km/h)*
a) Steering-wheel angle δ_L, b) Roll angle φ, c) Lateral acceleration a_{yo}, d) Yaw speed $\dot{\psi}$.



ESV limit curves for steady-state yaw speed in "steering-angle jump" test *(2.5 ton van, empty)*



PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

angle),
– Sinusoidal input (sinusoidal steering-angle input).

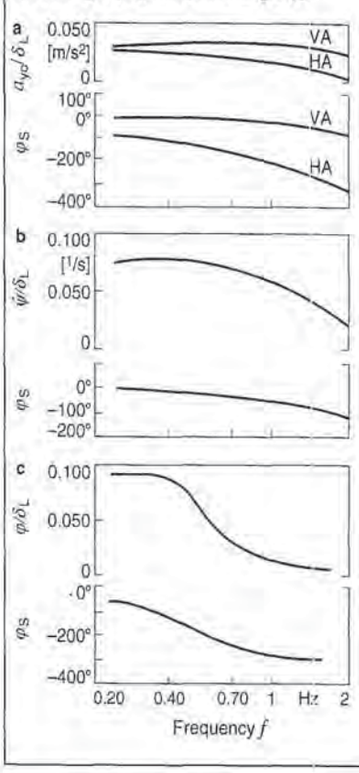
Step input (time response)
Starting with the vehicle traveling in a straight line, the steering wheel is abruptly "pulled" to a specified angle; the vehicle's response serves as the basis for evaluation. The most important quantities to be measured are [5]:
– Steering-wheel angle,
– Yaw speed,
– Vehicle speed, and
– Lateral acceleration.
The light utility van responds to the step input with a more rapid change in lateral acceleration – and thus in yaw – than the heavy truck.

US authorities have defined a transient yaw requirement for Experimental Safety Vehicles (ESV) [11]. Those vehicles which are classified as being similar to passenger cars may exhibit relatively pronounced overswing in the initial input phase (whereby this effect must cease after a certain period).

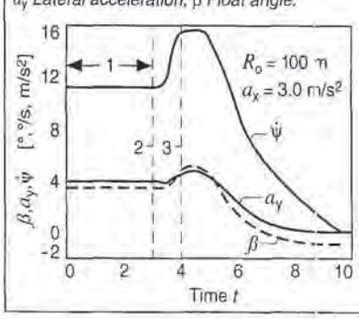
Sinusoidal steering-angle input (frequency curve)
Permanent sinusoidal steering inputs at the steering wheel are also used to measure frequency-response characteristics. This provides yet another basis for evaluating a vehicle's transitional handling response, the intensity and phase of which varies according to the steering frequency. The most important factors for this evaluation are [6]:
– Steering-wheel angle,
– Lateral acceleration,
– Yaw speed, and
– Roll angle.

Braking in a curve
Of all the maneuvers encountered in everyday driving situations, one of the most critical – and thus one of the most important with regard to vehicle design – is braking during cornering. The vehicle's concept must be optimized so that its reaction to this maneuver is characterized by the best possible compromise between steerability, stability and retardation [2], [7]. Testing starts from a specified initial rate of lateral acceleration, and focuses upon

Sinusoidal steering-angle input
(16-ton truck, v_0 = 60 km/h, δ_L = 60°).
Relative to steering-wheel angle:
a) Lateral acceleration a_{y0}/δ_L,
b) Yaw velocity $\dot{\psi}/\delta_L$ and
c) Roll angle φ/δ_L; with phase angle φ_s.



Time curve for braking during cornering
(7.5 ton truck, loaded)
1 Steady-state skidpad, 2 Initial braking, 3 Evaluation point. $\dot{\psi}$ Yaw speed, a_y Lateral acceleration, β Float angle.



PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

float angle and yaw speed relative to lateral deceleration as the significant factors.

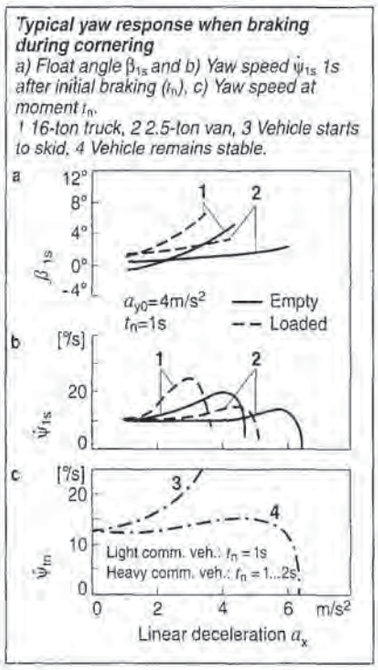
The vehicle, initially in steady-state circulation at a stipulated lateral acceleration, is braked with the rate of deceleration being increased incrementally. In every test, using the time functions, the measurements are taken "1 s after initial braking" for vehicles with hydraulic brakes (a delay of 1.5...2 s after initial braking is applied for heavy-duty commercial vehicles with compressed-air brake systems).

Float angle

Due to the weight transfer away from the rear axle and the tires' response to this phenomenon, higher rates of retardation result in greater float angles. At high rates of deceleration, in the vicinity of the traction limit, the distribution of brake force is the decisive factor for the float angle. The sequence in which the wheels lock exercises a decisive influence on the vehicle's stability.

Yaw speed

The yaw speed serves as a reference in determining whether braking performance during cornering is stable or unstable. In the illustration, both vehicles' yaw-speed curves move toward zero as the vehicles progress through phases of increasing deceleration up to full lockup at both axles. This indicates acceptable braking response: The vehicle remains stable.



Typical yaw response when braking during cornering

a) Float angle β_{1s} and b) Yaw speed $\dot{\psi}_{1s}$ 1s after initial braking (t_n), c) Yaw speed at moment t_n.

1 16-ton truck, 2 2.5-ton van, 3 Vehicle starts to skid, 4 Vehicle remains stable.

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[3] ISO. Road Vehicles – Steady-State Circular Test Procedure. ISO, 1982, No. 4138.

[4] ISO. Road Vehicles – Double Lane Change. ISO, 1975, TR 3888.

[5] ISO. Draft Proposal for an International Standard, Road Vehicles – Transient Response Test Procedure (Step/Ramp Input). ISO/TC22/SC9/N 185.

[6] ISO. Draft Proposal for an International Standard, Road Vehicles – Transient Response Test Procedure (Sinusoidal Input). ISO/TC 22/SC9/N219.

[7] ISO. Road Vehicles – Braking in a Turn. Open-Loop Test Procedure. ISO/DIS 7975.

[8] Zomotor A.; Braess H.H.; Rönitz R. Doppelter Fahrspurwechsel, eine Möglichkeit zur Beurteilung des Fahrverhaltens von Kfz? (Double Lane Change, a Method for Evaluating a Vehicle's Driveability?) ATZ 76, 1974, Volume 8.

[9] Mitschke M. Dynamik der Kraftfahrzeuge. (Dynamics of the Motor Vehicle), Springer Verlag, 1st Edition 1972, 2nd Edition 1982 and 1984, and subsequent Editions.

[10] Mitschke M. Fahrtrichtungshaltung – Analyse der Theorien. (Maintaining Direction of Travel – Analysis of the Theories). ATZ 70, 1968, Volume 5.

[11] Mischke A.; Göhring E.; Wolsdorf P.; von Glasner E.C. Contribution to the Development of a Concept of Driving Mechanics for Commercial Vehicles. SAE 83 0643.

PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

Special operating dynamics for commercial vehicles

Units and symbols

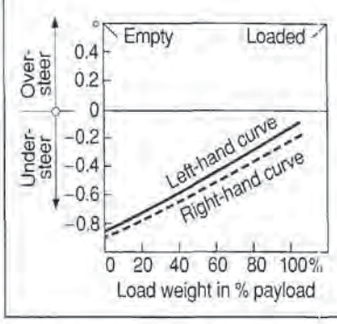
G_V	N	Front-axle load
G_H	N	Rear-axle load
G_G	N	Total weight
G_F	N	Sprung weight
U_V	N	Unsprung weight, front
U_H	N	Unsprung weight, rear
C_{DST}	N · m/wheel	Torsional spring rate for all stabilizers
$C_{FV,H}$	N/m	Spring rates for axle springs
$C_{RV,H}$	N/m	Spring rates for tires
$S_{FV,H}$	m	Spring track
$S_{RV,H}$	m	Tire track
$m_{V,H}$	m	Instantaneous center height
l_{IF}	m	Height of center of gravity, sprung weight
l_{IG}	m	Height of center of gravity, total vehicle
$C_{QV,H}$	N/m	Lateral stiffness rate of tires
r	m	Radius of curve

Self-steering properties

In the development phase in which the parameters affecting a vehicle's self-steering properties are determined, empirical and test-stand measurements and computer simulations are employed in an optimization process. The major determinants are the geometry and compliance rates of the steering, the frame and the suspension.

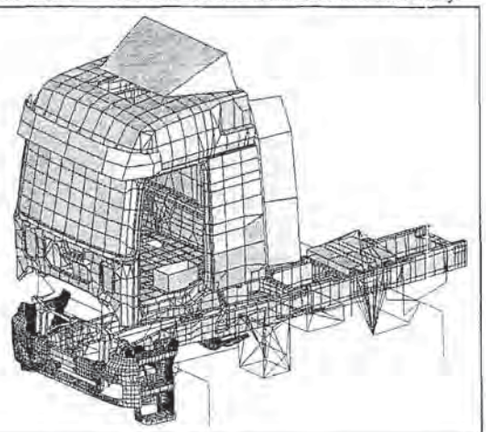
The objects of analysis are those interference factors influencing straight-running stability and cornering behavior which can be traced to the interaction between steering and suspension and which do not stem from driver inputs. The self-steering effect is examined at both the front and rear axles, in steady-state driving on a circular course, during braking and under unilateral spring compression.

Self-steering angle of a 18-ton truck cornering at 3 m/s² lateral acceleration
The self-steering angle is plotted along Y.



When the springs are compressed on one side, a solid axle supported on leaf springs will tend to rotate about the vehicle's vertical axis. The degree of spring tilt exercises a major effect on this type of roll-steer effect. Neutral behavior or mild understeer, desirable from the safety point of view, are enhanced by tilting the front spring upward at the front and downward toward the rear, while the rear spring is mounted in the opposite manner, with the lowest end at the forward extremity.

Finite element model of a semitrailer unit for determining self-steering properties
Includes kinematic patterns for steering and suspension as well as the entire frame of the semitrailer tractor.



PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

The wheel loads at the truck's rear axle vary dramatically, depending upon whether it is empty or loaded. This leads to the vehicle responding to reductions in load with more pronounced understeer.

On three-axle 6 x 4 vehicles, the non-steered tandem axle unit represents a constraining force around the vehicle's vertical axis, thus enhancing straight-line stability. For low speeds, the additional cornering force required at the front and rear axles is determined as follows:

Cornering forces resulting from constraint

$F_{S1} = F_{S2} - F_{S3}$ with

$F_{S2} = c_{p2} \cdot n_2 \cdot \alpha_2$

$F_{S3} = c_{p3} \cdot n_3 \cdot \alpha_3$

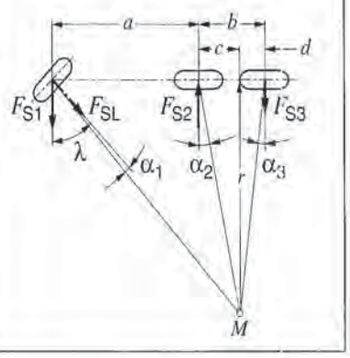
Slip angle

$$\alpha_2 = \frac{1}{r} \cdot \frac{c_{p3} \cdot n_3 \cdot b \cdot (a+b)}{c_{p3} \cdot n_3 \cdot (a+b) + c_{p2} \cdot n_2 \cdot a}$$

$$\alpha_3 = (b/r) - \alpha_2$$

c_p Coefficient of cornering stability from tire performance curve,
n Number of tires per axle, other designations as in illustration.

Cornering forces F_s and slip angle α on a 3-axle vehicle with non-steered tandem axle



Tipping resistance

As the vehicle's total height increases, there is also an increasing tendency in a curve for it to tip over to the side before it starts to slide. Simulations of the tipping process for precise determination of the tip limit take into account both the elasticities and displacements of the center of gravity.

Achievable rates of lateral acceleration at the tip limit:

Van	$b = 6$ m/s²,
Truck	$b = 5$ m/s²,
Double-decker bus	$b = 3$ m/s².

Width requirement

The width requirement of motor vehicles and truck-trailer combinations is greater during cornering than when the vehicle moves in a straight line. With respect to selected driving maneuvers, it is necessary to determine the radius described by the vehicle's outer extremities during cornering, both in order to ascertain its suitability for certain applications (e.g. narrow transit routes

Tipping resistance

Approximation formula for critical velocities v_{tip} on a 2-axle truck (in km/h):

$$v_{tip} = 7.98 \cdot \sqrt{\frac{r \cdot (G_V \cdot S_{RV} + G_H \cdot S_{RH})}{G_G \cdot h_G + \frac{G_V^2}{C_{QV}} + \frac{G_H^2}{C_{QH}} + \frac{G_F^2 \cdot h_m^2}{C_D - G_F \cdot h_m}}}$$

with $C_D = \frac{C_{DF} \cdot C_{DR} \cdot i^2}{C_{DF} + C_{DR} \cdot i^2}$; $i = \frac{h_m}{h_m + m}$;

$$h_m = h_m - m; \quad m = \frac{(G_V - U_V) \cdot m_V + (G_H - U_H) m_H}{G_F}$$

$$C_{DF} = 1/2 \cdot (C_{FV} \cdot S_{FV}^2 + C_{FH} \cdot S_{FH}^2) + C_{DSt}$$

$$C_{DR} = 1/2 \cdot (C_{RV} \cdot S_{RV}^2 + C_{RH} \cdot S_{RH}^2)$$

PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

through constricted areas) and to confirm compliance with legal regulations. Evaluation is conducted with reference to the tractrix principle using electronic programs.

Handling characteristics

Objective analyses of vehicle handling are based on various maneuvers such as steady-state circulation on the skidpad, steering-angle jumps, wag frequency response, and braking in a curve.

The dynamic lateral response of truck-trailer combinations generally differs from that of rigid vehicles. Particularly significant are the distribution of loads between truck and trailer, and the design and geometry of the mechanical coupling device within a given combination.

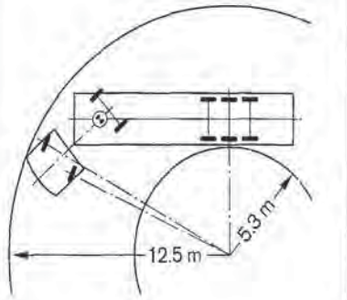
Rotary oscillation, with the vehicle's masses turning against the tires (as compliance element) around a vertical axis, impairs straight-running stability. This phenomenon is induced by

- Rapid steering corrections associated with evasive maneuvers,
- Crosswinds, and
- Pronounced slope to the side of the road, obstacles on one side, lane grooves.

Oscillation amplitudes associated with this pendulum motion must subside rapidly if vehicle stability is to be maintained.

Below is a graphic depiction of the yaw-speed progressions for various types of truck-trailer combination. The worst case is represented by a combination, in which the towing vehicle is empty, while the center-axle trailer is loaded. Here, the curve indicates an excessive increase in resonance. This type of combination demands a high degree of driver skill and circumspection.

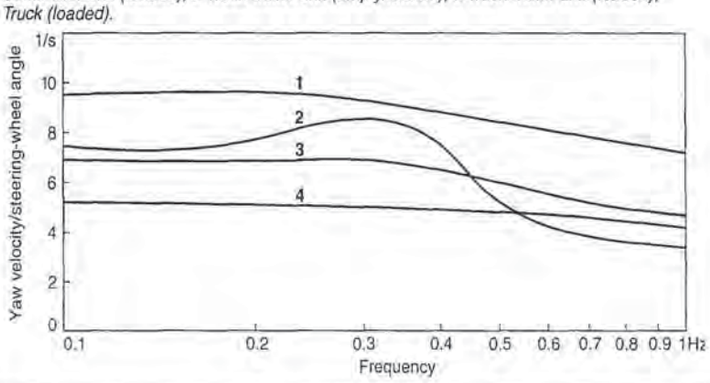
Tractor-semitrailer combination in the circular area as stipulated by the German Road Traffic Regulations (StVZO)



With semitrailer units, braking maneuvers undertaken under extreme conditions can induce jackknifing.

This process is initiated when, on a slippery road surface, loss of lateral-force is induced by excess braking force being applied at the tractor's rear axle, or due to excess yaw moment under μ-split conditions. Installation of an antilock braking system (ABS) represents the most effective means of preventing jackknifing. This measure is mandatory for towing vehicles >18 metric tons since 1 October, 1991.

Yaw-speed frequency progression
1 Semitrailer unit (loaded), 2 Truck-trailer unit (empty/loaded), 3 Truck-trailer unit (loaded), 4 Truck (loaded).



PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

Requirements for agricultural tractors

Units and symbols

Symbol		Unit
F	Weight (wheel oad) of a wheel	N
F_R	Rolling resistance	N
F_{Rh}	Rolling resistance, rear axle	N
F_{Rv}	Rolling resistance, front axle	N
F_{St}	Climbing resistance	N
F_T	Traction (motive) force of a wheel	N
F_{Th}	Traction (motive) force, rear	N
F_{Tv}	Traction (motive) force, front	N
F_w	Soil (ground) resistance	N
F_Z	Drawbar pull of tractor	N
$F_{Zerf.}$	Drawbar-pull requirement of implement	N
P_e	Net engine power	kW
$P_{Getr.}$	Transmission power losses	kW
P_N	Rated engine power	kW
P_R	Auto-motive power requirement	kW
P_S	Slip power losses	kW
P_{St}	Hill-climbing power requirement	kW
P_Z	Drawbar powe·	kW
v	Vehicle speed	km/h
v_0	Peripheral velocity of a driving wheel	km/h
η_{Getr}	Transmission/gearbox efficiency	–
η_L	Tractive efficiency at tractor wheels	–
η_T	Tractive efficiency of a single wheel	–
η_Z	Tractive efficiency of an agricultural tractor	–
λ	Engine utilization ratio	–
$\varkappa$	Coefficient of traction force	–
ϱ	Rolling resistance	–
σ	Wheel slip	%

Applications

Agricultural tractors are employed for field work and for general transport and farmyard duties. Depending upon the type of unit, power from the engine can be transmitted through an auxiliary PTO shaft or hydraulic lines, as well as via the drive wheels. The engine outputs for farm tractors used in the Federal Republic of Germany range up to approximately 250 kW, with weights of over 120 kN.

Higher engine outputs exaggerate the problems associated with supporting the weight at the ground on large-volume tires of adequate capacity, as well as the difficulties encountered in transforming the engine's power into tractive power at acceptable tractor speeds.

Essential requirements of a tractor

- High drawbar pull, high tractive efficiency.
- The engine must combine high torque increase and low specific fuel consumption with as constant a power characteristic as possible.
- Depending upon application and the distances involved, vehicle speeds (rated speeds) up to 25, 32, 40, 50 km/h, with > 60 km/h for special-purpose tractors; multiple conversion ratios with appropriate gear spacing (especially important up to 12 km/h), suitable for shifting under load if possible.
- Power take-off (PTO) shaft and hydraulic connections for powering auxiliary equipment. Option of installing and/or powering equipment at the front of the tractor.
- Facilities for monitoring and operating auxiliary equipment from the driver's seat, e.g. with hydraulic control levers (P. 824).
- Clear and logical layout of control levers in ergonomically correct arrangement.
- On field tractors, provision for adjusting track to suit crop-row spacing.
- Driver protection against vibration, dust, noise, climatic influences and accident.
- Universal applicability.

Drawbar pull and drawbar power of a tractor in field work

The effective drawbar pull is essentially determined by the tractor's weight, the type of drive (rear-wheel or 4-wheel drive) and the operating characteristics of its tires. The operational response of the tractor's drive tires is determined by such factors as type of soil and ground conditions (moisture and porosity), tire dimensions, carcass and tread design, and tire pressure. Due to these particular operating characteristics, the farm tractor in field work develops its maximum drawbar pull only when tire slip is high, whereas the maximum drawbar power is achieved at relatively low levels of slip and drawbar pull. With the engine developing 90% of its maximum output, the drawbar pull of an AWD tractor v ceed 60% of the rated engine pc under extremely favorable condi The effective engine output is:

$$P_e = P_Z + P_R + P_S + P_{Getr.}$$
$$(+ P_{St} \text{ on gradients})$$

The drawbar power is calculate

$$P_Z = F_Z \cdot v$$

With rear-wheel drive, the powe to propel the tractor itself is:

$$P_R = F_{Rv} \cdot v + F_{Rh} \cdot v_0$$

The slip power losses are defin

$$P_S = F_T \cdot (v_0 - v) = F_T \cdot \sigma \cdot v_0$$

The power losses in the transm are determined with the equatic

$$P_{Getr.} = P_e \cdot (1 - \eta_{Getr.})$$

Efficiency levels:
With rear-wheel drive:

$$\eta_L = \frac{F_{Th} - F_{Rv}}{F_{Th} + F_{Rh}} \cdot (1 - \sigma)$$

With all-wheel drive (AWD):

$$\eta_L = \frac{F_{Th} + F_{Tv}}{F_{Th} + F_{Tv} + F_{Rh} + F_{Rv}} \cdot (1 -$$

For single wheel:

$$\eta_T = \frac{F_T}{F_T + F_R} \cdot (1 - \sigma) = \frac{\varkappa}{\varkappa + \varrho} ($$

For the tractor:

$$\eta_Z = \eta_{Getr.} \cdot \eta_L = P_Z / P_e$$

The coefficients are calculated a

$$\varkappa = F_T / F$$
$$\varrho = F_R / F$$
$$\lambda = P_e / P_N$$
$$\sigma = (v_0 - v) / v_0$$

Drawbar-pull requirements of auxiliary equipment and traile

At a constant speed on a flat su drawbar-pull requirement depe on the rolling resistance F_R (equipment) or on soil resistanc the force needed to move a to

Examples of the power requi 1 meter swath on moderate gro

Implement	
Tiller on loose soil	
Vibrating harrow	
Circular harrow	

PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

drawbar pull of an AWD tractor will not exceed 60 % of the rated engine power, even under extremely favorable conditions.
The effective engine output is:

$$P_e = P_Z + P_R + P_S + P_{Getr.}$$
$$(+ P_{St} \text{ on gradients})$$

The drawbar power is calculated as:

$$P_Z = F_Z \cdot v$$

With rear-wheel drive, the power required to propel the tractor itself is:

$$P_R = F_{Rv} \cdot v + F_{Rh} \cdot v_o$$

The slip power losses are defined as:

$$P_S = F_T \cdot (v_o - v) = F_T \cdot \sigma \cdot v_o$$

The power losses in the transmission unit are determined with the equation:

$$P_{Getr.} = P_e \cdot (1 - \eta_{Getr.})$$

Efficiency levels:
With rear-wheel drive:

$$\eta_L = \frac{F_{Th} - F_{Rv}}{F_{Th} + F_{Rh}} \cdot (1 - \sigma)$$

With all-wheel drive (AWD):

$$\eta_L = \frac{F_{Th} + F_{Tv}}{F_{Th} + F_{Tv} + F_{Rh} + F_{Rv}} \cdot (1 - \sigma)$$

For single wheel:

$$\eta_T = \frac{F_T}{F_T + F_R} \cdot (1 - \sigma) = \frac{\varkappa}{\varkappa + \varrho} (1 - \sigma)$$

For the tractor:

$$\eta_Z = \eta_{Getr.} \cdot \eta_L = P_Z / P_e$$

The coefficients are calculated as follows:

$$\varkappa = F_T / F$$
$$\varrho = F_R / F$$
$$\lambda = P_e / P_N$$
$$\sigma = (v_o - v) / v_o$$

Drawbar-pull requirements of auxiliary equipment and trailers
At a constant speed on a flat surface, the drawbar-pull requirement depends either on the rolling resistance F_R (e.g. farm equipment) or on soil resistance F_W (e.g. the force needed to move a tool through the soil) or on both at the same time (e.g. beet lifter). Rolling resistance is calculated using the coefficient of rolling resistance and the sum of the weights supported at the wheels, giving:

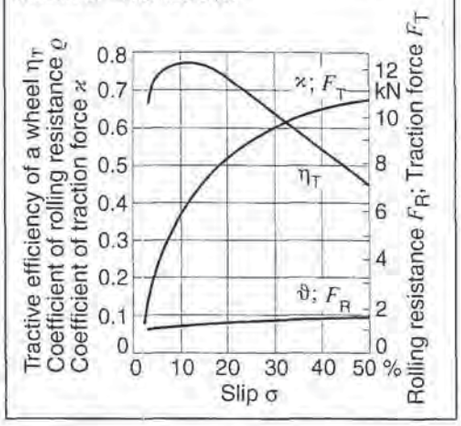
$$F_R = \varrho \cdot \Sigma F$$

For pneumatic tires on asphalt: $\varrho \leq 0.03$
For pneumatic tires on field:

$$\varrho = 0.04 \ldots 0.35$$

Soil resistance is determined by the type and condition of the soil, number and type of implements, working depth and vehicle speed. General reference figures for plowing would be a specific ground resistance of 400...600 N/dm² on moderate soils, with 600...1000 N/dm² on hard (clay) soils. On moderate ground at speeds of between 6 and 9 km/h, the soil resistance per meter of working width of a cultivator is 5500...7800 N for a working depth of 13...15 cm, and 11,000...12,500 N for a depth of 22...25 cm.

Operating characteristics of a tractor drive wheel
Tire: 16.9/14-30 AS; Wheel load: 1582 daN; Tire pressure: 1.1 bar; Ground: loamy clay, wheat stubble, treated with disk harrow, moisture: 17.3...20.8 %.



Examples of the power required by PTO-driven agricultural equipment working a 1 meter swath on moderate ground.

Implement	Required engine power kW	Working depth cm	Vehicle speed km/h
Tiller on loose soil	10.5...25	8	3...7
Vibrating harrow	8...22	8	3.5...6.5
Circular harrow	0...15	8	3.5...6.5

PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

Environmental stresses on automotive equipment

Climatic factors

Climatic stress factors acting upon automotive components encompass the effects of the natural environment, i.e. the macroclimate, and influences stemming from the vehicle itself (such as fuel vapor) and the microclimate within a component (such as the heat generated in electrical devices).

Temperature and temperature variations
The range extends from extremely low temperatures (storage, transport) all the way to the high temperatures associated with operation of the internal-combustion engine.

Atmospheric humidity and variations
This range embraces everything from arid desert climates to tropical environments, and can even extend beyond these under certain conditions (as occur for instance when water is sprayed against a hot engine block). Humid heat (high temperatures combined with high atmospheric humidity) is especially demanding. Alternating humidity results in surface condensation, which causes atmospheric corrosion.

Corrosive atmospheres
Salt spray encountered when the vehicle is operated on salt-treated roads and in coastal areas promotes electrochemical and atmospheric corrosion. Industrial atmospheres in concentrated manufacturing regions lead to acid corrosion on metallic surfaces. When they are present in sufficient concentrations, today's increasing amounts of atmospheric pollutants (SO_2, H_2S, Cl_2 and NO_x) promote the formation of contaminant layers on contact surfaces, with the result that resistance increases.

Water
Stresses of varying intensities result from rain, spray, splash, and hose water as encountered when driving in rain, during car and engine washes, and – in exceptional cases – during submersion.

Aggressive chemical fluids
The product in question must be able to resist the chemical fluids encountered in the course of normal operation and maintenance at its particular operating location. Within the engine compartment, such chemicals include fuel (and fuel vapor), engine oil and engine detergents. Certain components are confronted by additional substances, for example, brake-system components and the brake fluid used to operate them.

Sand and dust
Malfunctions result from the friction due to sand and dust on adjacent moving surfaces. In addition, under the influence of moisture, certain types of dust layers can cause current tracking in electrical circuits.

Solar radiation
The sun's rays cause plastics and elastomers to age (a factor to be taken into account in the design of external, exposed components).

Atmospheric pressure
Fluctuations in atmospheric pressure affect the operation and reliability of differential-pressure components, such as diaphragms, etc.

Laboratory simulation of stress

Climatic and environmental conditions are simulated both according to standardized test procedures (DIN IEC 68 – Environmental testing procedures for electronic components and equipment) and in special field-testing programs designed specifically for individual cases. The goal is to achieve the greatest possible approximation of the stresses encountered in actual practice ("test tailoring").

Temperature, temperature variation and atmospheric humidity
Simulation is carried out in temperature and climate chambers as well as in climate-controlled rooms which afford access to test personnel.

The dry heat test allows evaluation of a component's suitability for storage and operation at high temperatures. Testing is not restricted to ascertaining the effects of

PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

heat upon operation; it also monitors influences on material characteristics. Depending upon the particular application (component mounted on body, engine, or exhaust system), the degree of heat can cover an extremely wide range. The stress time can be up to several hundred hours.

Testing the product's operation under cold conditions devotes particular attention to starting behavior and changes in materials characteristics at low temperatures. The testing range extends down to – 40 °C for operation, and to – 55 °C for storage. At less than 100 hours, the actual testing times are shorter than those employed for dry heat.

A further test simulates temperature fluctuation between the extremes encountered in actual operation; the temperature gradient and the dwell time also contribute to determining the degree of stress. The dwell time must be at least long enough to ensure that the sample achieves thermal equilibrium. The different levels of thermal expansion mean that the temperature variations induce both material aging and mechanical stresses within the component. The selection of appropriate test parameters makes it possible to achieve substantial time-compression factors.

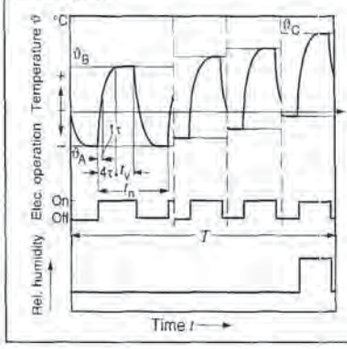
Atmospheric humidity testing under steady-state damp heat (e.g., +40 °C/93 % relative humidity) is employed in the evaluation of a product's suitability for operation and storage at relatively high humidity levels (tropical climates).

Corrosive atmospheres

Salt fog is produced by diffusing a 5 % NaCl solution at a room temperature of 35 °C. Depending upon the intended installation location, the test times can extend to several hundred hours. Cyclic salt fog is a combination test comprising the following: "salt fog, dry heat and damp heat". It yields a closer correlation with field results. The industrial-climate test comprises up to 6 cyclical alternations between an 8-hour dwell period at 40 °C/100 % relative humidity at 0.67 % SO_2 and 16 hours at room temperature. The pollutant test with SO_2, H_2S, NO_X and CL_2 is performed either for single gases or as a multisubstance test. Testing is carried out at 25 °C/75 % relative humidity

Test schedule for simulating combined stresses

t_v Dwell time, t_n Temperature-variation cycle, T Test cycle.



with concentrations in the ppm and ppb ranges, and lasts up to 21 days.

Water spray

A pivoting sprayer is used to simulate water spray. Water pressure, spray angle and the pivot angle can all be adjusted for different stress severity levels. The water-spray test employs high-pressure jets and standard steam-cleaners of the type used for cleaning engines.

Aggressive chemical fluids

The sample is wetted with the fluid in question for a defined period. This is followed by 24-hour storage at elevated temperature. This test can be repeated numerous times, according to the particular application.

Sand and dust

Dust simulation is carried using a device which maintains a dust density of 5 g per m^3 in moving air. A mixture of lime and fly ash is one of the substances employed.

Combined tests

Combined temperature, temperature variation and humidity tests on an operating electrical product ensure a high degree of convergence with the aging effects to be anticipated under extreme operating conditions. The advantage of this test is its high level of conformity with actual practice. The disadvantage is the test duration, which is generally well in excess of that required for the corresponding individual investigations.

PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

Internal-combustion engines

Operating concepts and classifications

The internal-combustion (IC) engine is the most frequently employed power source for motor vehicles. Internal-combustion engines generate power by converting chemical energy bound in the fuel into heat, and the heat thus produced into mechanical work. The conversion of chemical energy into heat is accomplished through combustion, while the subsequent conversion of this thermal energy into mechanical work is performed by allowing the heat energy to increase the pressure within a medium which then performs work as it expands.

Liquids, which supply an increase in working pressure via a change of phase (vaporization), or gases, whose working pressure can be increased through compression, are used as working media.

The fuels – largely hydrocarbons – require oxygen in order to burn; the required oxygen is usually supplied as a constituent of the intake air. If fuel combustion occurs in the cylinder itself, the process is called internal combustion. Here the combustion gas itself is used as the working medium. If combustion takes place outside the cylinder, the process is called external combustion.

Continuous mechanical work is possible only in a cyclic process (piston engine) or a continuous process (gas turbine) of heat absorption, expansion (production of work) and return of the working medium to its initial condition (combustion cycle).

If the working medium is altered as it absorbs heat, e.g. when a portion of its constituents serve as an oxidant, restoration of its initial condition is possible only through replacement.

This is called an open cycle, and is characterized by cyclic gas exchange (expulsion of the combustion gases and induction of the fresh charge). Internal combustion therefore always requires an open cycle.

In external combustion, the actual working medium remains chemically unchanged, and can thus be returned to its initial condition by suitable measures (cooling, condensation). This enables the

Table 1. Classification of the internal-combustion engine

Type of process		**Open process**				**Closed process**	
		Internal combustion				External combustion	
		Combustion gas ≙ working medium				Combustion gas ≠ working medium	
						Phase change in working medium	
						No	Yes
Type of combustion		Cyclic combustion			Continuous combustion		
Type of ignition		Auto-ignition	Externally supplied ignition				
Type of machine	Engine ≙ machine enclosing a working chamber	Diesel	Hybrid	Otto	Rohs	Stirling	Steam
	Turbine ≙ gas turbine	–	–	–	Gas	Hot steam	Steam
Type of mixture		Heterogeneous (in the combustion chamber)		Homogenous	Heterogeneous (in a continuous flame)		

PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

use of a closed process.

In addition to the main process characteristics (open/closed) and the type of combustion (cyclic/continuous), the various combustion processes for internal-combustion engines can also be defined according to their air-fuel mixture formation and ignition arrangements.

In external air-fuel mixture formation, the mixture is formed outside the combustion chamber. In this type of mixture formation a largely homogenous air-fuel mixture is present when combustion is initiated, so it is also referred to as homogeneous mixture formation.

In internal air-fuel mixture formation the fuel is introduced directly into the combustion chamber. The later internal combustion occurs, the more heterogeneous the air-fuel mixture will be at the time combustion is initiated. Internal mixture formation is therefore also called heterogeneous mixture formation. External ignition designs rely on an electric spark or a glow plug to initiate combustion. In auto-ignition, the mixture ignites as it warms to or beyond its ignition temperature during compression, or when fuel is injected into air whose boundary conditions permit evaporation and ignition.

Cycles

The p-V diagram

A basic precondition for continuous conversion of thermal energy into kinetic energy is a modification in the condition of the working medium; it is also desirable that as much of the working medium as possible be returned to its initial condition.

For technical applications the focus can rest on changes in pressure and the corresponding volumetric variations which can be plotted on a pressure vs. volume work diagram, or p-V diagram for short.

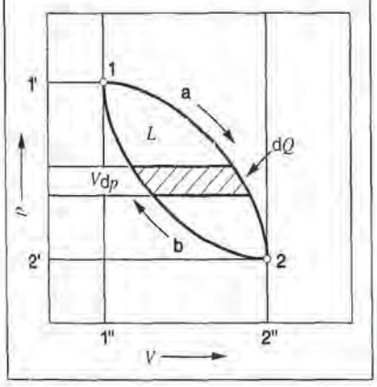
As the figure shows, the addition of heat and the change in condition of the working medium that accompany the progress of the process in the 1→2 phase must consume less energy than that required for the 2→1 phase. Once this condition is satisfied the result is an area corresponding to the process work potential: $L = \oint V dp$.

The T-S diagram

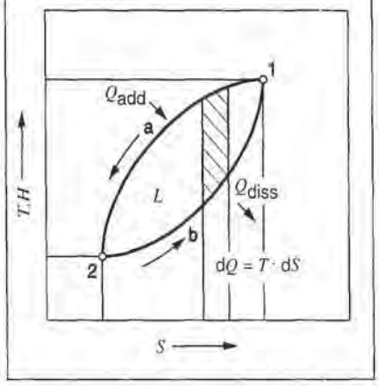
The temperature entropy, or T-S diagram, is used to provide a similar graphic representation of the bidirectional thermal energy transfers in this cyclic process.

In the T-S diagram heat quantities can be represented as areas in the same manner that work is represented as an area in the p-V diagram. With known specific working-medium heats, the T-S diagram can be transformed into the H-S dia-

A thermodynamic cycle illustrated using the p-V diagram



A thermodynamic cycle illustrated using the T-S or H-S diagram



PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

gram, known as the enthalpy-entropy diagram, in accordance with the equation $dH = c_p \cdot dT$.

The cycle illustrated in the p-V diagram on page 367 shows the amount of heat added along "a"

$$Q_{add} = \int_2^1 T_a \, dS$$

and the amount of heat dissipated along "b"

$$Q_{diss} = \int_2^1 T_b \, dS, \text{ where}$$

$$Q_{add} - Q_{diss} = L = \oint V dp$$

(difference between the amount of heat supplied and the amount of heat discharged) which corresponds to the available amount of mechanical work. The diagram also shows that a thermal efficiency $\eta_{th} = (Q_{add} - Q_{diss})/Q_{add}$ can be defined based on the equality of mechanical work and the difference between the heat quantities. It also illustrates the theoretical cycle providing the maximum amount of technical work, as found in the area between two specified temperatures for the working medium.

The Carnot cycle

This cycle, described in 1824 by Carnot, consists of two isothermal [1]) and two isentropic [2]) changes in condition, which yield the maximum area in the T-S diagram between T_{max} and T_{min}. Because the Carnot cycle represents maximum process efficiency between the defined temperature limits, it is the theoretical optimum for converting heat into work:

$$\eta_{thCarnot} = (T_{max} - T_{min})/T_{max}$$

Internal-combustion engines operate according to other cycles, however, because isothermal compression, i.e. a pressure increase in the working medium without an increase in temperature, and isothermal expansion are not technically feasible.

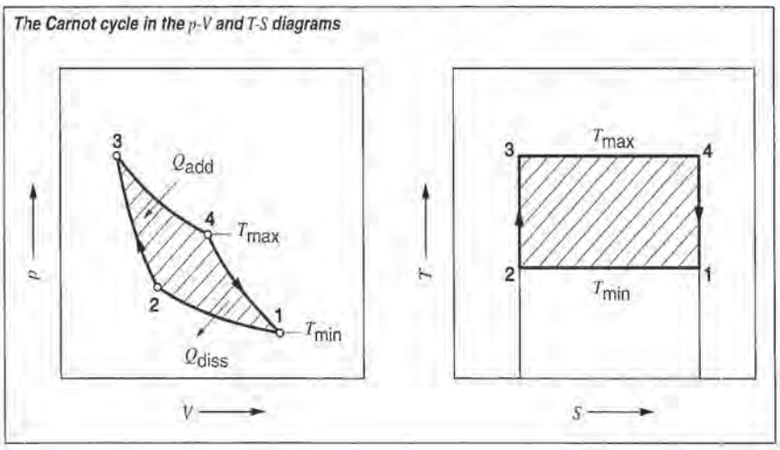
Theoretical treatment today involves the following ideal combustion cycles:
the constant-volume cycle for all piston engines with periodic combustion and generation of work, and
the constant-pressure cycle for all turbine engines with continuous combustion and generation of work.

Both cycles will be dealt with in more detail in the discussion of the corresponding machines.

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[1]) Isothermal change in condition: temperature does not change.
[2]) Isentropic change in condition: adiabatic (heat is neither added nor dissipated) and frictionless (reversible).
[3]) Isochoric change in condition: volume does not change; see P. 369.

The Carnot cycle in the p-V and T-S diagrams



PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

Reciprocating-piston engines with internal combustion

Operating concept

All reciprocating-piston engines operate by compressing air or an air-fuel mixture in the working cylinder prior to igniting the mixture, or by injecting fuel into the hot compressed air to initiate combustion. The crankshaft assembly converts the work generated in this process into torque available at the end of the crankshaft.

The p-V diagram reflects the actual power-generation process in the engine as a function of piston travel. It shows the mean effective pressures p_{mi} within the cylinder during a complete working cycle. Easier to produce are other diagrams such as the pressure vs. time (p-t) and the pressure vs. crankshaft angle (p-α) diagrams. The surfaces defined in these two diagrams do not directly indicate the amount of work generated, but they do provide a clear picture of essential data such as firing point and peak combustion pressure. The product of the mean effective pressure in the cylinder and the displacement yields the piston work, and the number of working cycles per unit of time indicates the piston power or the internal power (power index) for the engine. Here it will be noted that the power generated by a reciprocating-piston internal-combustion engine increases as min^{-1} rises (see equations on P. 416 to P. 423).

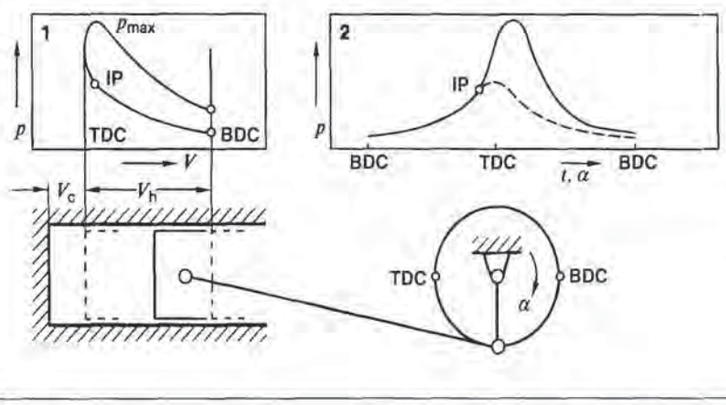
Ideal combustion cycle for piston engines with internal combustion

For reciprocating-piston engines with internal combustion, the ideal thermodynamic combustion process is the "constant-volume process", consisting of isentropic compression, isochoric[3]) heat supply, isentropic expansion and isochoric reversion of the ideal working gas to its initial condition. This cycle is only possible if the following conditions are met:

- No heat or gas losses, no residual gas,
- Ideal gas with constant specific heats c_p, c_v and $\varkappa = c_p/c_v = 1.4$,
- Infinitely rapid heat supply and discharge,
- No flow losses.

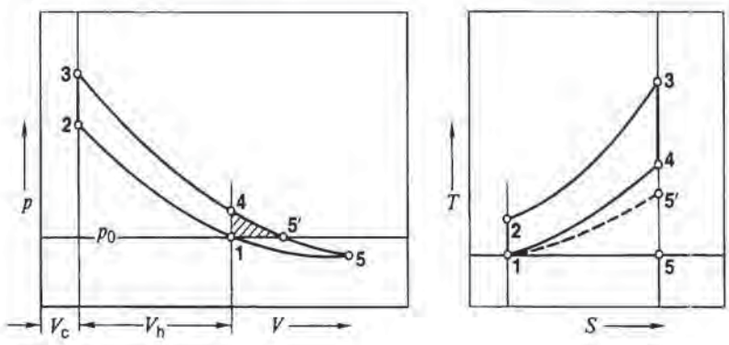
Because the crankshaft assembly restricts expansion to finite levels, the

The engine power cycle
1 In the p-V diagram, 2 in the p-t and p-α diagrams.



PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

Ideal constant-volume combustion cycle as shown in the p-V and T-S diagrams



4–5–1 surface in the diagrams is not directly available for use. Section 4–5'–1, lying above the atmospheric pressure line, becomes available when an exhaust-gas turbine is connected downstream.

The efficiency of the ideal constant-volume combustion cycle is calculated in the same manner as all thermal efficiencies:

$$\eta_{th} = \eta_v = (Q_{add} - Q_{diss})/Q_{add}$$

with $Q_{add} = Q_{23} = m \cdot c_v \cdot (T_3 - T_2)$ and
$Q_{diss} = Q_{41} = m \cdot c_v \cdot (T_4 - T_1)$

Using the same $\varkappa$ for compression and expansion:

$$\eta_{th} = 1 - Q_{diss}/Q_{add} = 1 - \frac{T_4 - T_1}{T_3 - T_2} = 1 - T_1/T_2$$

if $T_1/T_2 = \varepsilon^{\varkappa-1}$ then
$\eta_{th} = 1 - \varepsilon^{1-\varkappa}$,

where the compression ratio is defined as $\varepsilon = (V_c + V_h)/V_c$ with a displacement of V_h and a compression volume of V_c.

Real internal-combustion engines do not operate according to ideal cycles, but rather with real gas, and are therefore subject to fluid, thermodynamic and mechanical losses.

Efficiency sequence (DIN 1940)
The overall efficiency η_e includes the sum of all losses, and can thus be defined as the ratio of effective mechanical work to the mechanical work equivalent of the supplied fuel:

$\eta_e = W_e/W_B$ where

W_e is the effective work available at the clutch and
W_B is the work equivalent of the supplied fuel.

In order to better distinguish among the different losses, a further distinction can be made:

the fuel conversion factor η_B provides an index of combustion quality:

$\eta_B = (W_B - W_{Bo})/W_B$, where

W_B is the work equivalent of the supplied fuel,
W_{Bo} is the work equivalent of the unburned fuel.

There are no operating conditions in which complete combustion takes place.

A portion of the supplied fuel does not burn (hydrocarbon constituents in the exhaust gas), or fails to combust completely (CO in exhaust).

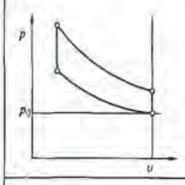
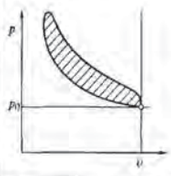
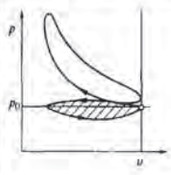
η_B is often defined as "1" for small diesel engines at operating temperature and for comparisons.

The efficiency index η_i is the ratio of indicated high-pressure work to the calorific content of the supplied fuel $\eta_i = W_i/W_B$.

The efficiency of cycle factor η_g includes all internal losses occurring in both high-pressure and low-pressure processes. These stem from:

PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

Table 2. Graphic representations and definitions of the individual and overall efficiencies of the reciprocating-piston engine

Pressure vs. volume diagram	Designation	Conditions	Definition	Efficiencies
	Theoretical reference constant-volume cycle	Ideal gas, constant specific heats, infinitely rapid heat addition and dissipation, etc.	$\eta_{th} = 1-\varepsilon^{1-\kappa}$ Theoretical or thermal efficiency	η_{th}
	Real high-pressure working cycle	Wall heat losses, real gas, finitely rapid heat addition and dissipation, variable specific heats	η_{gHP} Efficiency factor of the high-pressure cycle	η_g
	Real charge cycle (4-stroke)	Flow losses, heating of the mixture or the air, etc.	η_{gLW} Charge exchange efficiency	η_i
Mechanical losses	Losses due to friction, cooling, auxiliary units	Real engine	η_m	η_m η_m η_e

Real working gas, residual gas, wall heat losses, gas losses and pumping losses. For this reason, η_g is more appropriately broken down into η_{gHD} for the high-pressure portion and η_{gLW} for gas-exchange processes. The efficiency of cycle factor therefore indicates how closely engine performance approaches the theoretical ideal combustion cycle:

$$\eta_g = \eta_{gHD} \cdot \eta_{gLW} = W_i/W_{th} \text{ where}$$

W_i is the indicated work and
W_{th} is the work generated in the ideal-combustion cycle.

Mechanical efficiency η_m defines the relationship between mechanical losses – especially friction losses in the crankshaft assembly and induction/exhaust systems, and in oil and water pumps, fuel pump, alternator, etc. – and the work index:

$$\eta_m = W_e/W_i \text{ where}$$

W_e is the effective work available at the clutch and
W_i is the work index.

The efficiency chain therefore appears as follows:

$$\eta_e = \eta_B \cdot \eta_{th} \cdot \eta_{gHD} \cdot \eta_{gLW} \cdot \eta_m$$

(see Table 2).

PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

The spark-ignition (Otto) engine

The spark-ignition engine (or SI engine) is a piston engine with external or internal air-fuel mixture formation. External mixture formation generally produces homogenous mixtures, whereas an internally formed mixture is largely heterogeneous at the instant of ignition. The time of mixture formation is a major factor influencing the degree of homogenization achievable by internal mixture formation.

In both cases, the mixture is compressed to approximately 20...30 bar (ε = 8...12) on the compression stroke, to generate a final compression temperature of 400...500 °C. This is still below the auto-ignition threshold of the mixture, which then has to be ignited by a spark shortly before the piston reaches TDC.

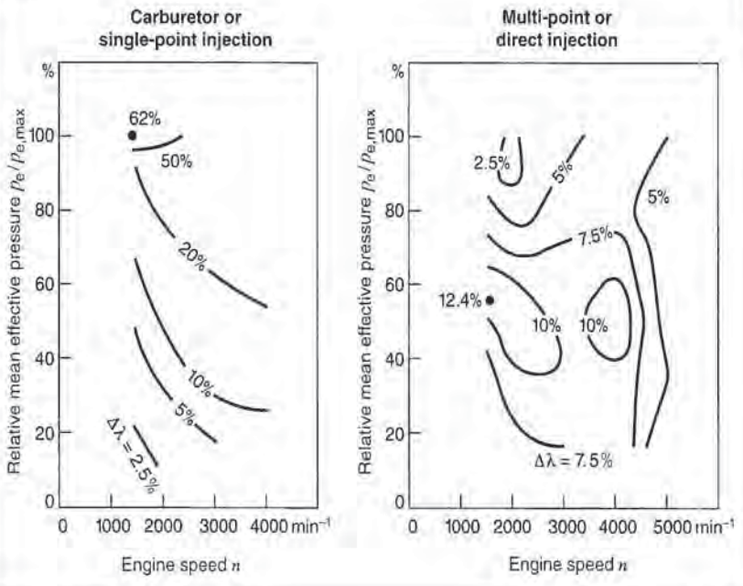
Since reliable ignition of homogenous air-fuel mixtures is only possible within a narrowly defined window of the air-fuel ratio (excess-air factor λ = 0.6...1.6), and flame velocity drops steeply as the excess-air factor λ increases, SI engines with homogenous mixture formation have to operate in a λ range 0.8...1.4 (best overall efficiency is achieved at $1.2 < \lambda < 1.3$). The λ range is further restricted to 0.98...1.02 for engines with three-way catalytic converters.

On account of this narrow λ range, load has to be controlled by the quantity of mixture entering the cylinders (quantity control): this is achieved by throttling the amount of air-fuel mixture entering the cylinders under part-load operating conditions (throttle control).

Optimization of the overall efficiency of SI engines has given rise to increasing development effort directed at engines with internal heterogeneous mixture formation.

Homogenous and heterogeneous mixture formation are alike in that economic efficiency and untreated emissions depend on the combustion process which takes place after ignition. Combustion, in turn, can be influenced to a very large extent by the flows and turbulences that can

Mixture-preparation
Differences in the air-fuel (A/F) ratio in the individual cylinders as a function of load and engine speed.



PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

be produced in the combustion chamber by the geometry of the intake duct and the combustion chamber.

Homogenous mixture formation

The homogenous mixtures present at the time when ignition commences cause the fuel to vaporize fully, because only gas or gas/vapor mixtures can achieve homogeneity.

If some factor (such as low temperature during a cold start) inhibits complete vaporization of the fuel, sufficient additional fuel must be provided to ensure that the volatile, vaporizable constituent can produce an adequately rich – and therefore combustible – air-fuel mixture (cold-start enrichment).

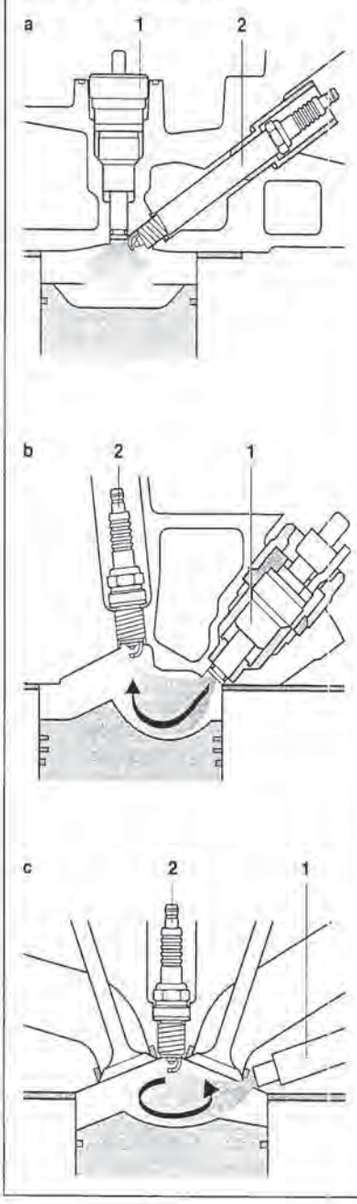
In addition to mixture homogenization, the mixture-formation system is also responsible for load regulation (throttle regulation) and for ensuring the minimization of deviations in the A/F ratio from cylinder to cylinder and from working cycle to working cycle.

Heterogeneous mixture formation

The aim pursued in heterogeneous internal mixture formation is that of operating the engine without throttle control across the entire operating map. Internal cooling is a side-effect of direct injection, so engines of this type can operate at higher compression ratios. The conjunction of these two factors, no throttle control and higher compression, means that the degree of efficiency is higher than that attainable with a homogenous mixture. Load is controlled by means of the mass of injected fuel. Development in mixture-formation systems gave fresh impetus to the "hybrid" or "stratified-charge" techniques that were the subject of much research from about 1970 onward. The definitive breakthrough came with the high-speed fuel injectors that allowed flexibility in injection timing and could achieve the high injection pressures required.

GDI (Gasoline Direct Injection) was the generic term applied to worldwide development in "jet-directed", "wall-directed" or "air-directed" mixture-formation systems (see illustration). The positions of spark plug and injector have a major influence on mixture formation, but flows in the

Mixture-formation systems for gasoline direct injection GDI *(assisted by swirl or tumble in each case)*
a) Jet-directed, b) Wall-directed, c) Air-directed.
1 Fuel injector, 2 Spark plug.



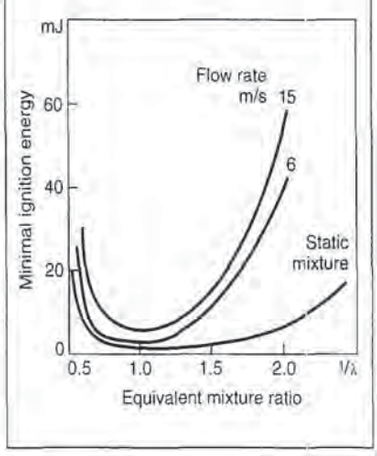
PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

combustion chamber are another, supporting factor. Swirl (induced by spiral or tangential channels) is primarily rotation about an axis paralleling that of the cylinder, whereas the axis of tumble, which is induced by fill channels, is normal to the cylinder's axis.

Precision positioning of the spark plug and the jet from the fuel injector is essential for jet-directed spray injection. The spark plug is under severe strain, because it is struck directly by the jet of liquid fuel. Wall-directed and air-directed configurations direct the mixture to the plug by means of the motion of the charge, so requirements in this respect are not as high.

Heterogeneous mixture formation entails excess air (unthrottled operation), so lean-burn catalytic converters have to be developed in order to reduce nitrogen-oxide emissions.

Minimal ignition energy for propane/air mixtures



Ignition

The ignition system must reliably ignite the compressed mixture at a precisely defined instant, even under dynamic operating conditions with the attendant substantial fluctuations in mixture flow patterns and air-fuel ratios. Reliable ignition can be promoted by selecting spark-plug locations with good mixture access in combination with efficient mixture swirl patterns; these are especially important considerations for lean operation and at very low throttle apertures. Similar improvements can also be achieved by positioning the spark plug in small auxiliary "ignition chambers".

Ignition-energy requirements depend on the mixture's air-fuel (A/F) ratio. An ignition energy of 0.2 mJ is required for gasoline/air mixtures in the stoichiometric range, while up to 3 mJ may be required to ignite richer or leaner mixtures.

The ignition voltage required increases with the gas pressure at the instant of ignition. Increasing the electrode gap is one way of improving ignition reliability, but at the expense of higher ignition voltage and accelerated electrode wear.

The energy content of the mixture ignited by the spark must be sufficient to ignite the neighboring mixture. This defines the leanest possible mixture and the earliest possible instant of ignition. In engines with a compression ratio $\varepsilon \approx 8...12$, this range is approximately 40...50 °crankshaft before TDC.

Combustion process

The initial thermal reaction which occurs between the provision of the ignition energy by the spark and the exothermal reaction of the air-fuel mixture is the ignition phase. This phase is roughly constant over time, with mixture composition as the only influencing factor. As a result, increasing engine speeds are accompanied by proportionately higher ignition delays – referred to piston travel (°crankshaft) – which change together with excess-air factor (lambda) λ.

The moment of ignition, therefore, has to be advanced as engine speed increases and excess-air factor λ rises. Ignition advance, however, is limited by the fall-off in the mixture's energy density in the vicinity of the electrodes (see above). When this physical limit is reached, designers can resort to twin spark-plug configurations or pre-chamber ignition to improve the situation.

The heat-release transient is determined by the rate of combustion, which in turn is defined by the speed of flame propagation and the area of the flame front. The speed of flame propagation depends

PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

on diffusion processes at the flame front and reaches a peak of approximately 20...40 $m \cdot s^{-1}$ in gasoline-air mixtures with approx. 10% air deficiency ($\lambda = 0.9$). It is influenced by the excess-air factor λ and the temperature of the mixture.

The area of the flame front can be influenced by the geometry of the combustion chamber and the position of the spark plug. Folding of the flame front due to turbulence and induced flows (such as swirl and tumble) is a significant factor in this respect. The flows induced primarily by the induction process and to a lesser extent by combustion-chamber geometry in conjunction with the compression squish fold the flame front and thus accelerate the process of energy conversion. Tumble, swirl and squish increase with engine speed and consequently, folding of the flame front also becomes more pronounced. This explains why the rate of heat release increases with speed despite the fact that by definition, the rate of flame propagation must remain constant.

Although it can factor in ultra-low-turbulence processes or in tests in low-flow pressure chambers, the turbulence created by flame propagation itself is of no significance in the combustion process as it takes place in modern SI engines.

The rising pressure due to local flame propagation causes an increase in temperature throughout the mixture, including that not yet reached by the flame and known as the "end gas". On account of local heat radiation and heat conduction, however, the temperature in the flame front is higher than in the rest of the mixture. This ensures regular flame propagation. The anomaly known as combustion knock or pre-ignition, due to simultaneous combustion of the end gas, occurs when the increase in pressure causes the temperature of the end gas to exceed its ignition limits.

Low fuel consumption and high efficiency are promoted by high combustion speeds (brief duration), combined with the optimal thermal release pattern relative to piston travel. Maximum heat release should occur shortly (approx. 5...10 °crankshaft) after top dead center. If most of the heat is released too early, wall heat losses and mechanical losses (high peak pressure) are increased. Late heat release leads to sacrifices in thermal efficiency (efficiency of cycle factor) and high exhaust-gas temperatures.

The ignition timing must be selected for optimum thermal generation curves in accordance with the:
- Air-fuel mixture ratio (λ, T),
- Effects of engine parameters (particularly load and speed) on combustion-chamber turbulence,
- Constant-duration ignition and flame propagation processes, meaning that variations in ignition timing are required as engine speed increases.

Problems and limits of combustion

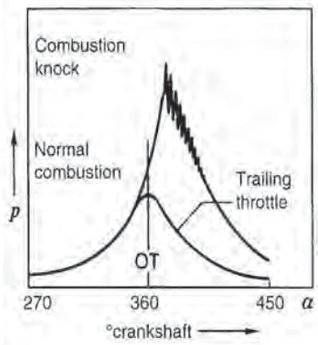
In actual practice, reliable flame initiation and propagation in engines with external mixture formation and spark ignition prohibit the use of mixtures leaner than $\lambda > 1.3$, although these would be desirable for improving the levels of theoretical (polytropic exponent) and gas-exchange (low throttle losses) efficiency, along with useful reductions in wall-heat and dissociation losses (reduction in combustion temperature). Appropriate tests are undertaken with GDI engines (Gasoline Direct Injection).

Although higher compression ratios provide enhanced part-load efficiency, they also increase the risk of combustion knock (pre-ignition) under full load. Pre-ignition occurs when the entire charge of end gas reaches ignition temperature and burns instantaneously without regular flame propagation. The end gas is highly compressed and its energy density is therefore very high, so pre-ignition suddenly releases very large amounts of heat. The high local temperatures caused in this way place extreme loads on the engine components and can also damage them. The high-energy cycles also result in extreme pressure peaks. Within the combustion chamber these pressure peaks propagate at the speed of sound and can cause damage to the piston, cylinder head, and cylinder-head gasket at critical points.

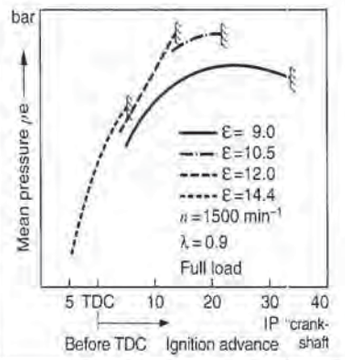
The risk of combustion knock can be reduced by using fuel additives or by richer mixtures (supplementary internal cooling).

PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

Combustion knock with normal combustion and trailing throttle in the p-α diagram



Potential mean effective pressure at pre-ignition limit as a function of compression ratio and timing



The current expedient of avoiding pre-ignition by retarding the ignition timing raises problems of its own, especially when used on high-compression engines. Because the ignition curve (mean pressure relative to ignition point) becomes increasingly steeper as compression increases, the resulting sacrifices in mean effective pressure are accompanied by extreme exhaust-gas temperatures. Reliable detection and avoidance of pre-ignition are thus vitally important in the ε = 11...13 compression range.

Load control

In unthrottled GDI engines with heterogeneous mixture, load is controlled by means of the quantity of fuel injected. Spark-ignition engines with homogenous mixture formation, on the other hand, afford little latitude for operation with lean mixtures, so load control has to be implemented by adjusting the mass flow of mixture. In carburetor engines, which have lost virtually all their significance in automotive engineering, this can be achieved by throttling the mixture mass flow. In engines with intake-manifold injection, throttle control to reduce the density of the intake air is the conventional approach. This arrangement, however, increases charge-cycle losses, so development is concentrating on alternative methods of load control. Mass flow can be influenced, for example, by prematurely closing the intake valves and thus shortening the effective intake periods. This complicated means of load control, however, requires fully variable valve timing and can cause fuel condensation as the result of expansion when the intake valves are closed. This drawback can be countered with "feedback control", an arrangement in which the intake valves are not closed until the requisite mass of mixture has just had time to fill the cylinder.

Another way of reducing or even eliminating throttle losses is exhaust-gas recirculation with the intake valves open. Load can be varied across a wide range by modulating the exhaust-gas recirculation rate.

Regulating charge-air pressure is a method of accomplishing load control over wide regions of the characteristic map with supercharged spark-ignition engines.

PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

Power output and economy

The efficiency index for engines with external mixture formation and spark ignition falls primarily in the lower portion of the map (see illustration). This is owing to combustion inefficiency (insufficient turbulence, inadequate charge density) along with an inefficient gas-exchange process.

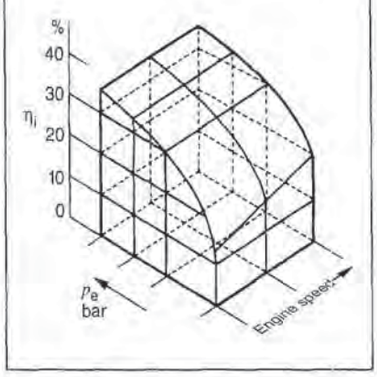
Effective efficiency is further reduced by the low mechanical efficiency characteristic of this region of the map.

All measures for avoiding these lower sections of the map therefore contribute to improving the engine's overall efficiency.

Selective interruption of the fuel supply to individual cylinders allows the remaining cylinders to operate at higher efficiency levels with improved combustion and gas exchange. Valve deactivation provides further reductions in power loss by allowing the intake and exhaust valves for the deactivated cylinders to remain closed. Cylinder shut-off entails immobilizing the mechanical power-transmission components of the idle cylinders for further increases in mechanical efficiency.

The measures cited above vary in terms of sophistication, but engine speed reduction also enhances general efficiency while promoting effective gas exchange; simultaneous reductions in mean frictional pressure also improve mechanical efficiency.

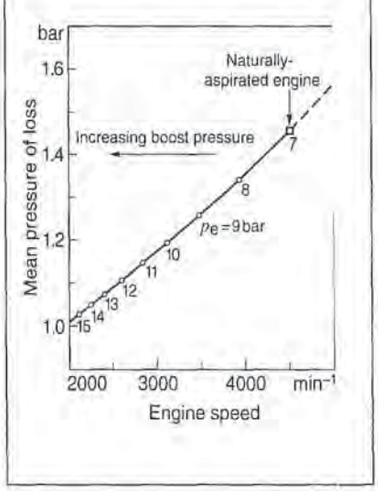
Curves of indicated efficiency vs. load factor and engine speed for a spark-ignition engine with throttle control



The diesel engine

A diesel engine is a reciprocating-piston engine with internal (and thus heterogeneous) mixture formation and auto-ignition. During the compression stroke intake air is compressed to 30...55 bar in naturally aspirated engines or 80...110 bar in supercharged engines, so that its temperature increases to 700...900 °C. This temperature is sufficient to induce auto-ignition in the fuel injected into the cylinders shortly before the end of the compression stroke, as the piston approaches TDC. In heterogeneous processes the mixture formation is decisive in determining the quality of the combustion which then follows, and the efficiency with which the inducted combustion air is utilized, and thus in defining the available mean effective pressure levels.

Relationship between engine speed and mean pressure of friction loss
5-liter gasoline engine, P_e = 130 kW = constant at increasing engine load, p_e Mean pressure.



PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

Mixture formation

In heterogeneous mixtures, the air-fuel ratio λ extends from pure air ($\lambda = \infty$) in the spray periphery to pure fuel ($\lambda = 0$) in the spray core.

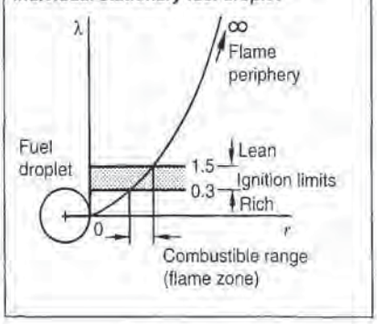
The figure provides a schematic illustration of the λ distribution and the associated flame zone for a single stationary droplet. Because this zone always occurs for every drop of injected mixture, load control with heterogeneous mixture formation can be performed by regulating the fuel supply. This is termed mixture-quality control.

As with homogenous mixtures, combustion takes place in the relatively narrow range between $0.3 < \lambda < 1.5$. The mass transport necessary for generating these combustible mixtures relies on diffusion and turbulence; these are produced by the mixture formation energy sources described below as well as by the combustion process itself.

Kinetic energy of the fuel spray

The spray's kinetic energy varies according to the pressure differential at the nozzle orifice. Along with the spray pattern (as determined by the nozzle geometry) and the fuel's exit velocity it determines the configuration of the space in which the air and fuel interact as well as the range of droplet sizes in the chamber. The spray energy is influenced by the delivery rate of the fuel-injection pump and the dimensions of the metering orifice in the injector nozzle.

Curve of the air-fuel (A/F) ratio λ in an individual stationary fuel droplet



Thermal energy

Thermal energy from the combustion-chamber walls and the compressed air vaporize the injected fuel (as a film layer on the walls and as droplets).

Combustion-chamber shape

The shape of the combustion chamber and the action of the piston can be utilized to create turbulence (squish), or to distribute liquid fuel or the fuel-air vapor jet.

Controlled air patterns (swirling action)

If the direction of fuel flow is roughly perpendicular to the direction of the vortex and droplet vaporization is taking place, a movement imparted to the combustion air inside the combustion chamber, usually in the form of solid-particle rotating flow, promotes the flow of air toward the fuel stream, and removes the combusted gases from the stream.

As the wall film evaporates, the air's swirling motion absorbs the vapor layer and provides thermal insulation between the combusted and fresh gases, while the microturbulence patterns superimposed upon the solid-particle vortex ensure rapid mixture of air and fuel. The air's controlled solid-particle swirl can be induced using special induction tract geometries or by shifting a portion of the cylinder charge into a rotationally symmetric auxiliary chamber (by transporting it through a side passage).

Partial combustion in a swirl chamber

When fuel is partially combusted in an auxiliary chamber its pressure rises above that in the main combustion chamber. This increase then propels the partially oxidized combustion gases and vaporized fuel through one or more passages into the main combustion chamber, where they are thoroughly mixed with the remaining combustion air.

The diesel combustion process makes use of at least one (but usually an appropriate combination) of these mixture formation methods.

PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

Direct injection
This term refers to all designs with a single unified combustion chamber.

Low-swirl or static-charge spray-injection combustion
This combustion process operates virtually without swirl of the air mass in the cylinder, or even with none at all, and relies on the energy in the injection jets to ensure mixture formation. It was formerly used in large medium- and slow-speed diesels operating with high excess-air factors on account of thermal design considerations.

The combustion chamber is a wide, usually ω-shaped recess centered in the piston crown. Fuel is injected through a central, vertical multi-hole nozzle with 5...8 orifices.

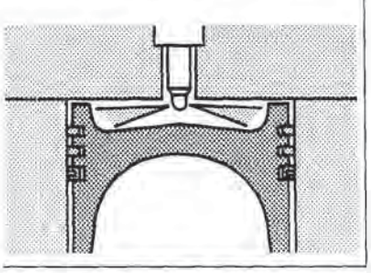
Higher excess-air factors and higher injection-jet energies (resulting from higher injection pressures) are two consequences of the increasingly lower emission limits for NO_X and particulates in the exhaust gas. At the same time, all modern commercial-vehicle engines with displacements down to approximately 1 l per cylinder are turbocharged and are fitted with governors to restrict their speed ranges for the sake of improved fuel consumption. These are the reasons why this method has become commonplace for commercial-vehicle engines, particularly since the retarded start of injection for the sake of further reductions in NO_X emissions goes hand in hand with higher thermal loads in the engines themselves.

Swirl-assisted multiple-orifice nozzle combustion process
The mixture-forming energy of the injection jets alone is not enough for sufficiently uniform and rapid mixture preparation in high-revving diesels with wide operating-speed ranges and small swept volumes (in other words, the engines most frequently found in passenger cars and vans). Supportive motion of the air inside the combustion chamber is required. This is achieved by installing pistons with considerably shallower recesses. These recesses are constricted at the top in order to create highly turbulent squish from the piston gap in the vicinity of the injection-jet contact points and to accelerate the swirl of the air charge induced by the design of the inlet elements (swirl inlet ducts). The total swirl velocity of the in-cylinder air mass achieved in this way is selected to ensure that the air-fuel mixture formed over the injection period from the injection jet issuing from the nozzle, and the air swirl rotating about an axis normal to this jet, completely fills and utilizes the downflow region of the combustion chamber until the next injection jet is received.

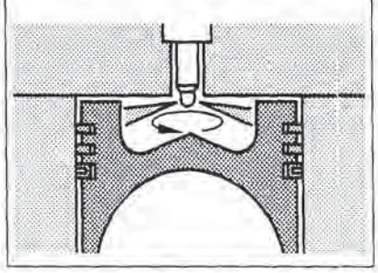
Consequently, this method employs nozzles with a significantly smaller number of orifices, namely 4...6. In this case, too, the nozzle is positioned as close as possible to an imaginary line extending though the center of the piston recess.

If the air-fuel mixture fails to completely fill the combustion-chamber segment, both air utilization and power output will suffer. On the other hand, if there is an

Combustion-chamber shape and nozzle location for the static-charge spray-injection process without air swirl



Combustion-chamber shape and nozzle location for multiple-orifice process with air swirl



PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

overlap and the mixture extends beyond this space between the individual injection events, the resulting excessive local fuel concentration leads to air deficiencies and increased formation of soot.

M System

In the MAN wall-distribution combustion system (M system) most of the fuel is sprayed against the wall of the combustion chamber. This process supplements the energy in the injection spray by exploiting the heat from the combustion-chamber wall and the swirling action of the air to form the mixture. The single-orifice nozzle, projecting into the narrow piston-crown chamber at an angle, sprays the fuel into the swirl and against the combustion-chamber wall. Here the fuel forms a film which vaporizes and forms a very homogenous mixture with the swirling combustion-chamber air as it passes. This process combines excellent air utilization with low exhaust-gas opacity (soot emissions).

The fuel film evaporates more slowly from the combustion-chamber wall than the droplets in the compressed air, so combustion processes of this nature are no longer of practical value in terms of the requirements applicable to fuel consumption and gas-phase emissions.

Divided-chamber (two-chamber) combustion systems

Divided-chamber combustion systems are well-suited to small, high-speed diesel engines, usually installed in passenger cars. Here, very stringent requirements are placed on mixture formation speed and air utilization (potential λ). At the same time, economic considerations dictate that the use of the expensive injection equipment required to generate high injection energy be avoided.

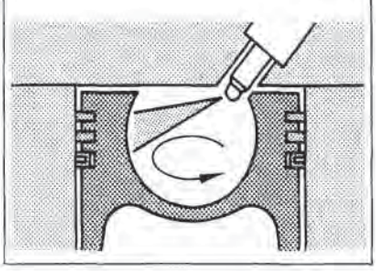
In addition, swirl-type intake passages present difficulties with regard to volumetric efficiency.

The divided-chamber method combines rich mixtures in the auxiliary chamber with relatively lean charges in the main chamber to achieve extremely low NO_x and HC emissions.

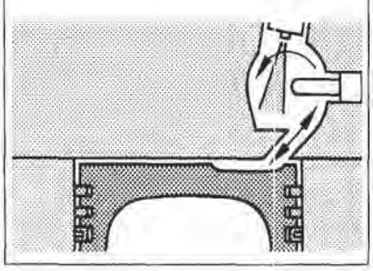
Swirl-chamber system

This process features an almost spherical auxiliary chamber, comprising approx. 50 % of the total compression volume, located at the edge of the main combustion chamber. The connection between the auxiliary and main chambers is furnished by a passage which enters the main chamber at an angle directed toward the center of the piston. The swirl chamber houses the injector and the glow plug (starting aid). The compression stroke generates an intense air vortex in the swirl chamber. As in the M process, the fuel is injected eccentrically to converge with the swirl pattern and land against the chamber's wall. Critical factors are the design of the swirl chamber (for instance, with supplementary mixture vaporization surfaces where the injection spray contacts the wall) and the locations of the injector and the glow plug; these factors define the quality of the combustion process. This process combines very high engine speeds (> 5000 min^{-1}), with good air utilization and low particulate emissions.

Combustion-chamber shape and nozzle location in the MAN M system



Combustion-chamber shape and nozzle location for the swirl-chamber system



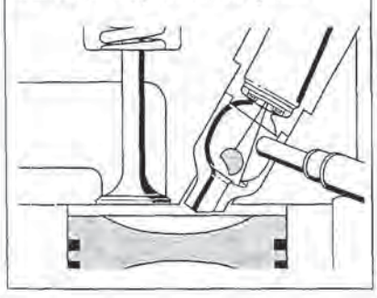
PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

Prechamber system

The prechamber system features an auxiliary chamber (prechamber) which is centrally located relative to the main combustion chamber, with 35...40 % of the compression volume. Here too, the injection nozzle and glow plug (starting aid) are located in the prechamber. It communicates with the main combustion chamber through several orifices to allow the combustion gases to mix as completely as possible with the main combustion air. One optimized prechamber concept utilizes the deflection surface below the injector nozzle to simultaneously induce rapid mixture formation and a controlled turbulence pattern (on some designs) in the prechamber. The turbulent flow meets the injection spray, which is also aimed into the swirl at an angle. The entire system, including the downstream glow plug, provides combustion with very low emissions and major reductions in particulates. The process is distinguished by a high air-utilization factor, and is also suitable for high engine speeds.

On account of the inherently high fuel consumption, prechamber combustion is becoming less and less viable for passenger-car applications. Common rail injection (P. 551) and refined turbocharging techniques for small engines have rendered direct injection increasingly commonplace in passenger-car diesel engines.

Combustion-chamber shape and nozzle location for the prechamber system



Combustion process

The start of injection (and thus the start of mixture formation) and the initiation of the exothermal reaction (start of ignition) are separated by a certain period of time, called ignition lag. The actual delay is defined by:

- the ignitability of the fuel (cetane number),
- the compression end pressure (compression ratio, boost factor),
- the compression end temperature (compression ratio, component temperatures, intercooling), and
- the fuel-management system.

The combustion process, which begins with the start of ignition, can be subdivided into three phases. In the "premixed flame" phase, the fuel injected prior to the start of ignition and mixed with the air combusts. The fuel injected once ignition has started combusts in a "diffusion flame". That portion of the combusted fuel which burns as a very rapid premixed flame is primarily responsible for the pressure rise, and thus is the primary cause of both combustion noise and oxides of nitrogen. The slower-burning diffusion flame is the main source of particulates and unburned hydrocarbons.

The third, post-injection, phase is when the soot formed primarily during the second phase is oxidized. This phase is becoming increasingly significant in modern combustion processes.

The heat release of a diesel engine depends on the combustion process itself, but also to a very large extent on the start of injection, the injection rate, and the maximum injection pressure. In direct-injection diesel engines, the number of orifices in the nozzle is another crucial factor. The injection system, moreover, requires a pre-injection capability (pilot injection) in order to reduce combustion noise and ensure that injection for the main injection phase commences as early as possible. This reduces fuel consumption for given levels of nitrogen-oxide emissions.

The diagram illustrates the thermal-release patterns which are characteristic of the various injection methods. The dual-stage combustion available with the di-

PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

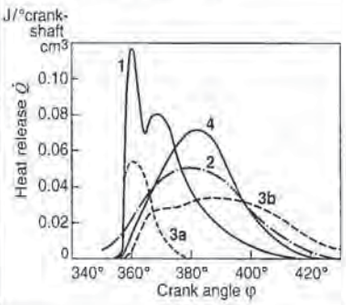
vided-chamber process provides yet another means of influencing the combustion process by allowing selection of different diameters for the passage between the auxiliary and main chambers.

Problems and limits of combustion

Because the fuel injected into diesel engines must ignite through auto-ignition, highly volatile fuel is required (cetane number CN ≈ 45...50). Since at low starting speeds compression does not begin until after intake-valve closure (that is, significantly after BDC), the effective compression ratio, and with it the compression temperature, are greatly reduced. This means that despite high compression ratios, ignition problems can occur during starting, particularly when the engine is cold.

In addition, cold engine components tend to absorb thermal energy from the compressed air (polytropic exponent: $1.1 < n < 1.2$). The equation $T_1 = T_0 \cdot \varepsilon^{n-1}$ indicates that a reduction in the effective compression or the polytropic exponent causes a reduction in the final compression temperature. At the same time, mixture formation is unsatisfactory at low engine speeds (low injection pressure, large fuel droplets) and air movement is inadequate. Extended vaporization times (injection begins sooner) and an increase in the injected fuel quantity – to significantly higher levels than the full-load quantity (providing more low-boiling fuel) can only partially solve the starting problem, because the higher-boiling fuel constituents leave the engine in the form of white or blue smoke. Thus starting aids designed to increase the temperature, such as glow plugs or flame starting systems, are essential, especially in small engines.

Thermal-release curves
1 Air-distributed direct fuel injection (naturally-aspirated engine tuned for maximum economy), 2 Wall-distribution direct injection (designed for minimal noise), 3 Divided-chamber process in auxiliary chamber (3a) and main chamber (3b), 4 Minimum emissions static-charge spray injection (intercooled turbocharged engine).



Because a significant portion of the mixture-formation process occurs during combustion in heterogeneous processes, it is important to avoid local concentrations of excessively rich mixture in the diffusion flame, as the result would be an increase in soot emissions, even with extremely lean overall mixtures. The air-fuel ratio limits at the officially mandated tolerance level for smoke provide an index of air-utilization efficiency. Divided-chamber engines reach the smoke limit with excess air of 5...15%, while the comparable figure is 10...80% for direct-injection diesels. It should be noted that large-volume diesel engines must also be run with significant levels of excess air owing to thermal component load.

Soot is an inevitable byproduct of heterogeneous combustion, so a sootless diesel engine must inevitably remain a conditional development and will require significant improvement in soot oxidation.

It has, however, proven possible to reduce the particulate emissions from modern diesel engines to below the visibility threshold using a variety of measures. These include raising injection pressures at the nozzle, and the transition to optimized spray injection processes featuring larger combustion recesses in the piston, multiple injector orifices, exhaust-gas turbocharging, and charge-air cooling. The planned limits for particulates, however, dictate the need for the development of particulate filters employing the requisite regenerative systems.

Due to the abrupt combustion in that portion of the fuel that vaporizes and mixes with air during the ignition-lag period, the auto-ignition process may be characterized by "hard", loud combustion during operation under those conditions where this fuel comprises a large propor-

PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

tion of the total. These conditions include idle, low part-throttle on turbocharged engines, and high-load operation on high-speed naturally-aspirated powerplants.

The situation can be improved by decreasing the ignition lag (preheating the intake air, turbocharging or increasing compression) and/or by reducing the injected fuel quantity during the ignition-lag period. On direct-injection engines, this reduction is usually obtained by applying pilot injection, whereas on divided-chamber engines a special injector configuration is employed (throttling pintle nozzle).

Not to be confused with the "hard" combustion inherent to the design is the "knock" to which turbulence-chamber arrangements with pintle nozzles are particularly susceptible, and which occurs primarily in the medium- and low-load areas of the diesel's operating curve. This phenomenon is traceable to inadequacies in the mixture-formation system (poor injector "chatter" or soot at the injectors), and is characterized by a pulsating metallic sound.

The diesel engine must be designed for operation with high peak pressures, and its materials and their dimensions must be selected accordingly. The reasons include:

- High compression ratios required for reliable starting and noise reductions,
- Combustion process with maximum ignition propagation for fuel economy, and
- Increasingly frequent use of turbochargers featuring higher charge pressures.

Owing to the fact that diesel engines must also operate with excess air (lean) at full throttle, they generally have lower specific outputs than their spark-ignition counterparts.

Hybrid processes

Hybrid engines share characteristics with both diesel and spark-ignition engines.

Charge stratification

In stratified-charge SI engines the mixture directly adjacent to the spark plug is enriched to ensure reliable ignition, while the rest of the combustion process proceeds in an extremely lean mixture. The objective is to combine the part-throttle economy comparable to that of a diesel with (especially important) low NO_x and CO emissions.

Research on open-chamber combustion systems with many characteristics similar to those of diesel systems (mixture quality control, high-pressure injection, etc.) is focusing on employing internal mixture formation (Texaco TCCS, Ford PROCO, Ricardo, MAN-FM, KHD-AD) to generate an ignitable mixture at the spark plug while using progressively leaner mixtures (down to pure air) in the remainder of the combustion chamber.

Processes which use internal mixture formation have an air-utilization factor comparable to that of diesel engines.

Divided-chamber combustion systems tend to resemble spark-ignition engines in their basic layout (throttle control, mixture induction, etc.). In these the spark plug is located within a small auxiliary chamber – the ignition chamber – corresponding to roughly 5...25 % of the total compression volume. One or several passages connect this primary ignition chamber with the main combustion chamber.

The auxiliary combustion chamber features an additional injector which injects a portion of the fuel directly (VW, Porsche-SKS), or a supplementary valve to supply fuel-air mixture to the ignition chamber (Honda-CVCC).

A disadvantage of these processes is their more complex design and the higher HC emissions stemming from lower exhaust-gas temperatures and the attendant reductions in secondary combustion activity in the exhaust tract.

PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

Multifuel engines

In multifuel engines, the ignitability and knock resistance of the fuel can be relatively insignificant; these engines must be able to burn fuels of varying qualities without sustaining damage. Because fuels used with multifuel engines can have a very low knock-resistance rating, external mixture formation would be accompanied by the danger of combustion knock or pre-ignition. For this reason, multifuel engines always use internal mixture formation and retarded injection timing (similar to the diesel engine). The injection pump for multifuel operation has an annular lubrication channel in the pumping element to lubricate the plungers with engine oil from the forced-feed lubrication system. This lubrication channel also prevents fuel from entering the pump's camshaft chamber. Because the low ignition quality of the fuels makes auto-ignition difficult or even impossible, multifuel engines operate with extremely high compression ratios (Mercedes-Benz, MTU: ε = 25:1). As an alternative they may be equipped with an auxiliary ignition source such as spark plugs or glow plugs (MAN-FM). At ε = 14...15, the compression ratio in these engines with externally-supplied ignition lies between that of spark-ignition and diesel engines.

A special type of ignition is the ignition spray process used in alcohol and gas engines (KHD, MWM), in which supplementary diesel fuel – corresponding to 5...10 % of the full-throttle supply of diesel fuel – is sprayed directly into the combustion chamber to ensure ignition. In this process, the main energy source can be supplied via external or internal mixture formation.

Whereas stratified charging with new mixture-formation systems is a focus of attention within the framework of ongoing development of GDI engines, multifuel engines are of virtually no significance at this time.

Gas exchange

In combustion engines employing open processes, the gas-exchange (exhaust and refill) system must serve two decisive functions:

1. Replacement is employed to return the gas medium to its initial (start of cycle) condition, and
2. The oxygen required to burn the fuel is provided in the form of fresh air.

The parameters defined in DIN 1940 can be used to evaluate the gas-exchange process. For overall air flow (air expenditure $\lambda_a = m_g/m_{th}$) the entire charge transferred during the work cycle m_g is defined with reference to the theoretical maximum for the specific displacement. In contrast, the volumetric efficiency $\lambda_{a1} = m_z/m_{th}$ is based exclusively on the fresh charge m_z actually present or remaining in the cylinder. The difference between m_z and the total charge transfer m_g consists of the proportion of the gas that flows directly into the exhaust tract in the overlap phase, making it unavailable for subsequent combustion. The retention rate

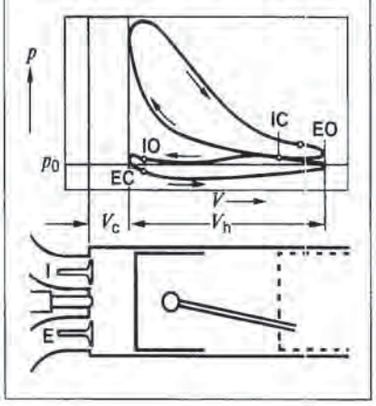
$\lambda_z = m_z/m_g$

is an index of the residual charge in the cylinder. The scavenge efficiency

$\lambda_S = m_z/(m_z+m_r)$

indicates the volume of the fresh charge m_z relative to the existing total charge, consisting of the fresh charge and the

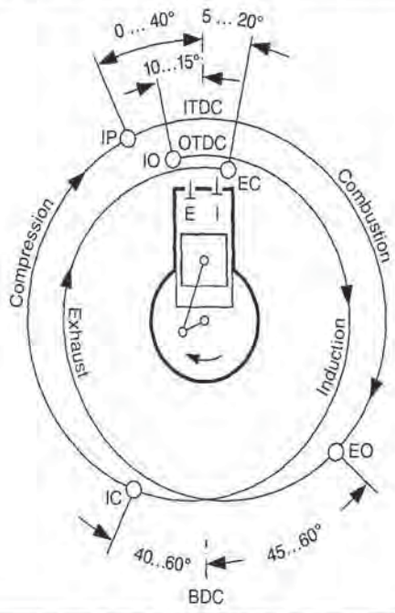
Representation of the four-stroke gas exchange process in the p-V diagram



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US PATENT NO. 6,581,565

4-stroke gas-exchange process

E Exhaust,
EO Exhaust opens,
EC Exhaust closes,
I Intake,
IO Intake opens,
IC Intake closes,
TDC Top dead center,
OTDC Overlap TDC,
ITDC Ignition TDC,
BDC Bottom dead center,
IP Ignition point.



residual gas m_r. Here, the parameter m_r indicates the amount of residual gas from earlier working cycles remaining in the cylinder after the exhaust process.

In a 2-stroke cycle, the gas is exchanged with every rotation of the crankshaft at the end of the expansion in the area around bottom dead center. In a 4-stroke cycle, separate intake and exhaust strokes provide a supplementary gas-exchange cycle.

4-stroke process

Valve timing – and thus gas exchange – are regulated by a control shaft (camshaft) rotating at half the frequency of the crankshaft by which it is driven. The camshaft opens the gas-exchange valves by depressing them against the valve springs to discharge the exhaust gas and to draw in the fresh gas (exhaust and intake valves respectively). Just before piston bottom dead center (BDC), the exhaust valve opens and approx. 50 % of the combustion gases leave the combustion chamber under a supercritical pressure ratio during this predischarge phase. As it moves upward during the exhaust stroke, the piston sweeps nearly all of the combustion gases from the combustion chamber.

Shortly ahead of piston top dead center (TDC) and before the exhaust valve has closed, the intake valve opens. This crankshaft top dead center position is called the gas-exchange TDC or overlap TDC (because the intake and exhaust processes overlap at this point) in order to distinguish it from the ignition TDC. Shortly after gas-exchange TDC, the ex-

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US PATENT NO. 6,581,565

haust valve closes and, with the intake valve still open, the piston draws in fresh air on its downward stroke. This second stroke of the gas-exchange process, the intake stroke, continues until shortly after BDC. The subsequent two strokes in the 4-stroke process are compression and combustion (expansion).

On throttle-controlled gasoline engines, during the valve overlap period exhaust gases flow directly from the combustion chamber into the intake passage, or from the exhaust passage back into the combustion chamber and from there into the intake passage. This tendency is especially pronounced at low throttle openings with high manifold vacuum. This "internal" exhaust-gas recirculation can have negative effects on idle quality, but it is impossible to avoid entirely, as a compromise has to be found between adequate high-speed valve lift and a satisfactory idle.

Early exhaust valve timing allows a high degree of blowdown, and thus guarantees low residual-gas compression as the piston sweeps through its upward stroke, although at the price of a reduction in the work index for the combustion gases.

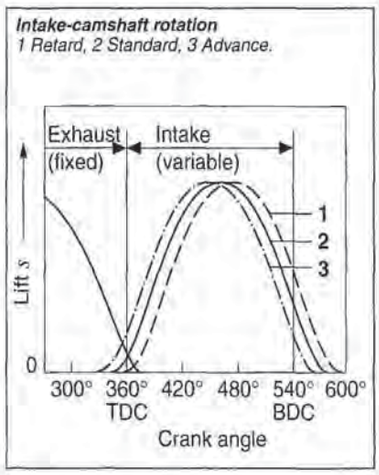
The "intake valve closes" (IC) timing exercises a decisive effect on the relationship between air expenditure and engine speed.

When the intake valve closes early (IC) the maximum charge efficiency occurs at low engine speeds, while late closing shifts the efficiency peak toward the upper end of the min^{-1} spectrum.

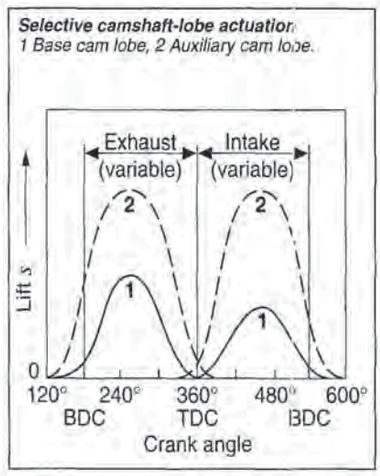
Obviously, fixed valve timing will always represent a compromise between two different design objectives: Maximum brake mean effective pressure – and thus torque – at the most desirable points on the curve, and the highest possible peak output. The higher the min^{-1} at which maximum power occurs, and the wider the range of engine operating speeds, the less satisfactory will be the ultimate compromise. Large variations in the valves' effective flow opening relative to stroke (i.e. in designs featuring more than two valves) will intensify this tendency.

At the same time, the demands for minimal exhaust emissions and maximum fuel economy mean that low idle speeds and high low-end torque (despite and along with high specific outputs for reasons of power-unit weight) are becoming increasingly important. These imperatives have led to the application of variable valve timing (especially for intake valves), whereby the most significant concepts (with attention focused on high-speed gasoline engines in series production) have been the following:

Variable camshaft timing by rotation: A hydraulic control mechanism rotates the intake camshaft to vary the valve timing for "intake opens" and "intake closes" as a



Intake-camshaft rotation
1 Retard, 2 Standard, 3 Advance.



Selective camshaft-lobe actuation
1 Base cam lobe, 2 Auxiliary cam lobe.

PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

function of engine speed (Alfa-Romeo, Mercedes-Benz).

The camshaft is set for delayed closing of the intake valve at idle and at high engine speeds. This results in a certain amount of valve overlap at overlap top dead center (OTDC) for stability at idle and enhanced output at high min^{-1}. The camshaft is rotated to close the intake valve early (IC) under full load in midrange operation, furnishing higher volumetric efficiency with correspondingly high levels of torque.

Selective camshaft-lobe actuation: The timing of the intake and exhaust valves is varied by alternating between separate lobes featuring two different lift profiles (Honda).

The first lobe features a profile tailored for optimum intake and exhaust timing and valve lift in the lower and middle engine-speed ranges. At high min^{-1}, a rocker arm that pivots freely at low speeds is coupled to the standard rocker arm; this rocker arm rides on a second cam lobe to provide greater lift and extended opening times.

Infinitely-variable valve lift and timing: Continuous modification of valve timing as a function of engine speed (Fiat).

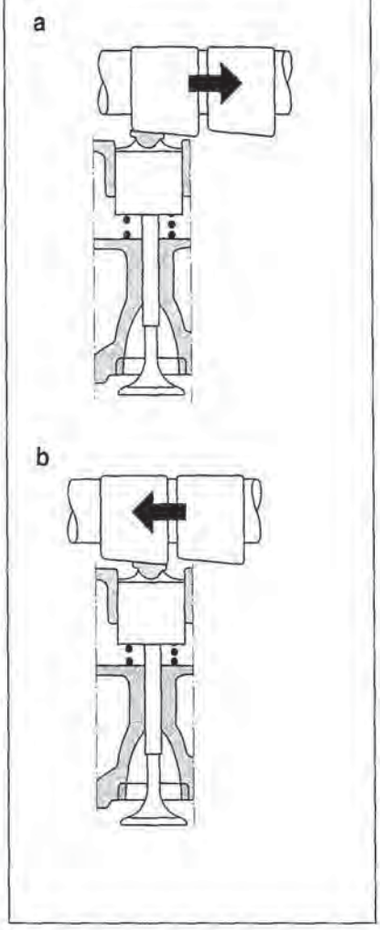
This is the optimum concept, but it is also the most difficult to implement. It employs cam lobes with a curved three-dimensional profile and lateral camshaft shift to achieve substantial increases in torque over the engine's entire min^{-1} range.

High-speed solenoid injectors (injection valves) have opened the way for development of electro-hydraulically actuated valve timing systems. Work is also under way on valve timing arrangements with solenoid actuation. At this time, however, solutions of this nature have not progressed as far as series production.

Evaluating gas-exchange components

The intake and exhaust tracts can be evaluated using stationary flow testing with flow numbers or passage efficiency levels. It is useful to evaluate the exhaust valves in the lower lift range with reference to supercritical pressures of the kind occurring in the blowdown phase.

Infinitely-variable valve lift and timing
a) Minimum lift, b) Maximum lift.



Alongside assessment of the flow number, analysis of in-cylinder flow is becoming increasingly significant. These studies can also be based on stationary flow testing and the derived parameters for swirl and tumble. Increasing use is being made of 3D computer models which, unlike the available measuring techniques, can furnish local information about flow conditions. Highly developed engine models are widely used today in the theoretical assessment of the overall gas cycle.

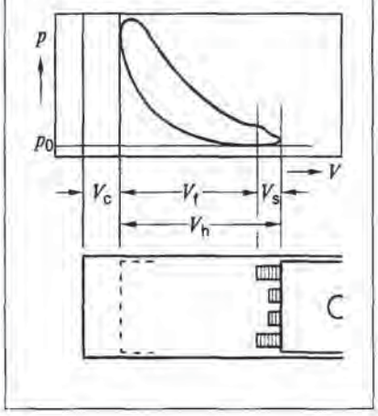
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US PATENT NO. 6,581,565

Advantages of the 4-stroke process:
Very good volumetric efficiency over the entire engine-speed range, low sensitivity to pressure losses in the exhaust system, along with relatively good control of the charging-efficiency curve through selection of appropriate valve timing and intake system designs.

Disadvantages of the 4-stroke process:
Valve control is highly complex. The power density is reduced because only every second shaft rotation is used to generate work.

2-stroke process
To maintain gas exchange without an additional crankshaft rotation, the gases are exchanged in the two-stroke process at the end of expansion and at the beginning of the compression stroke. The intake and exhaust timing are usually controlled by the piston as it sweeps past intake and exhaust ports in the cylinder housing near BDC. This configuration, however, requires symmetrical control times and involves the problem of short-circuit scavenging. In addition, 15...25 % of the piston stroke cannot produce work because only charge volume V_f and not displacement volume V_h can be exploited for power generation. Because the two-stroke process lacks separate intake and exhaust strokes, the cylinder must be filled and scavenged using positive pressure, necessitating the use of scavenging pumps. In an especially simple and very frequently-used design, the bottom surface of the piston works in conjunction with a crankcase featuring a minimal dead volume to form a scavenging pump. The illustrations show a 2-stroke engine with crankcase scavenging or crankcase precompression along with the associated control processes. The processes which take place on the scavenging pump side are shown in the inner circle, while those occurring on the cylinder side are shown in the outer circle. Satisfactory cylinder scavenging is available using crossflow scavenging, loop scavenging and uniflow scavenging.

Graphical representation of the 2-stroke gas-exchange process in the p-V diagram



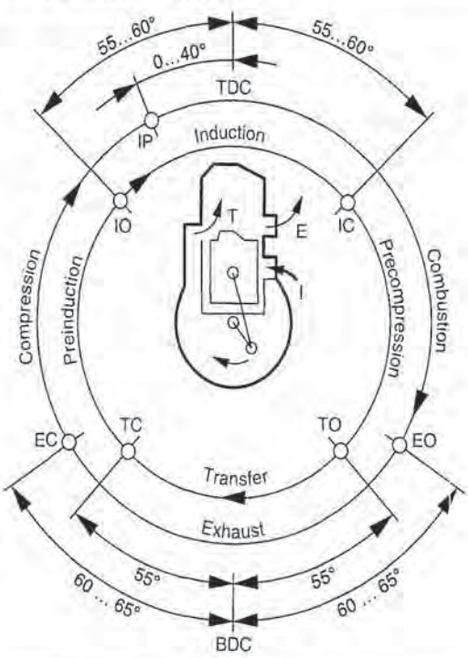
Advantages of the 2-stroke process:
Simple engine design, low weight, low manufacturing costs, better torsional force pattern.

Disadvantages of the 2-stroke process:
Higher fuel consumption and higher HC emissions (cylinder scavenging is problematic), lower mean effective pressures (poorer volumetric efficiency), higher thermal loads (no gas-exchange stroke), poor idle (higher percentage of residual gas).

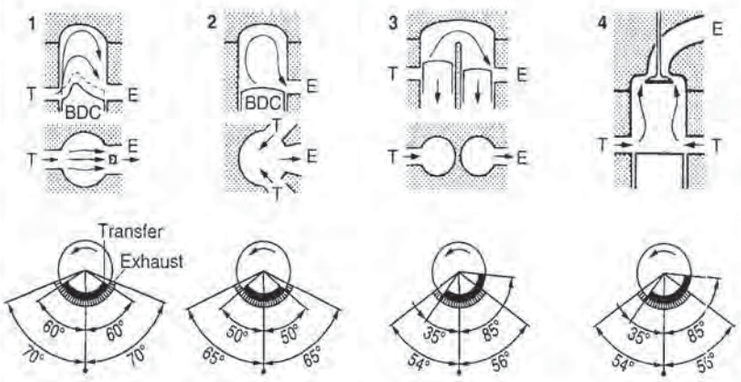
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US PATENT NO. 6,581,565

2-stroke gas-exchange process with crankcase compression

E Exhaust,
EO Exhaust opens,
EC Exhaust closes,
I Intake,
IO Intake opens,
IC Intake closes,
T Transfer passage, closes,
TO Transfer passage
TC Transfer passage opens,
TDC Top dead center,
BDC Bottom dead center,
IP Ignition point.



2-stroke scavenging.
1 Cross scavenging, 2 Loop scavenging, 3 and 4 Uniflow scavenging.



PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

Supercharging processes

The power of an engine is proportional to the processed air mass m_z. Because this air throughput, in turn, is proportional to air density, the power of an engine, given a specific displacement and engine speed, can be increased by precompressing the air before it enters the cylinders, i. e. by supercharging.

The supercharging ratio indicates the density rise as compared to a naturally-aspirated engine. One determining factor is the system used (potential pressure ratio). The maximum ratio for a given pressure increase is obtained when the temperature of the compressed air (boost air) is not increased or is returned to its initial level by intercooling.

In the spark-ignition engine, the supercharging ratio is restricted by the pre-ignition threshold. In the diesel engine maximum permissible peak pressures are the limiting factor. In order to avoid these problems, supercharged engines usually have lower compression ratios than their naturally aspirated counterparts.

Dynamic supercharging

The simplest type of supercharging exploits the dynamic properties of the intake air.

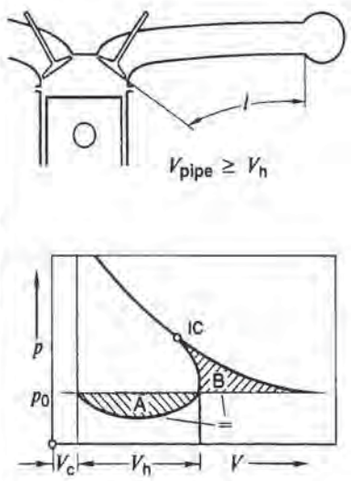
Ram-pipe supercharging

Each cylinder has a special intake manifold of a specific length, usually connected to a common plenum chamber. The energy balance is characterized by the fact that the intake work of the piston is converted into kinetic energy in the column of gas upstream from the intake valve, and this kinetic energy, in turn, is converted into charge compression work.

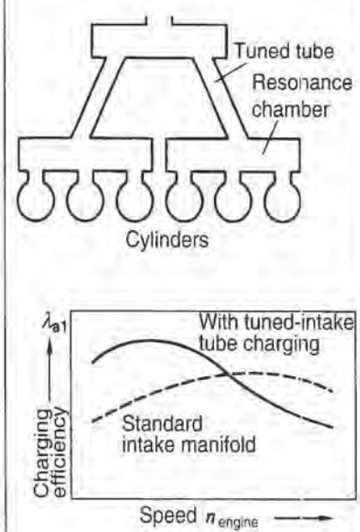
Tuned-intake-tube charging

In tuned-intake charging, short ducts connect groups of cylinders with the same ignition intervals to resonance chambers. These resonance chambers are connected to the atmosphere or a common chamber by tuned tubes, and act as Helmholtz resonators.

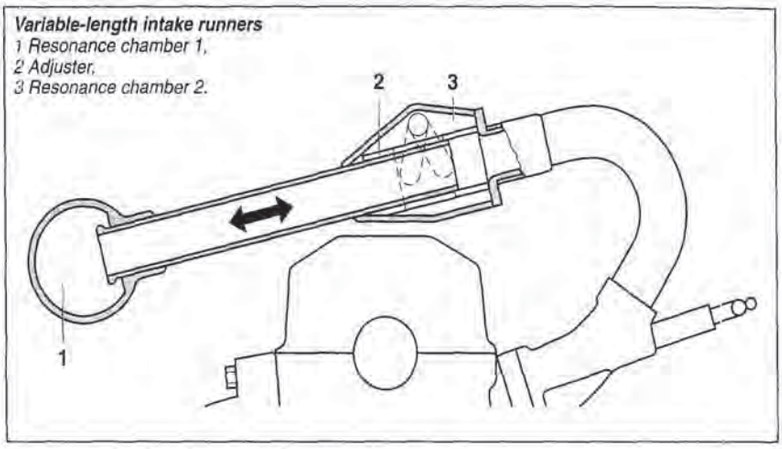
Ram-pipe supercharging
Configuration and energy balance. Induction work A of the piston corresponds to compression work B.



Tuned-intake-tube charging
Configuration and charging-efficiency curve.



PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565



Variable-length intake runners
1 Resonance chamber 1,
2 Adjuster,
3 Resonance chamber 2.

Variable-geometry intake manifold

Several manufacturers (BMW, Citroën, Opel, Ford) have already introduced systems employing the principles of dynamic boost (including combination designs). These systems enhance charge efficiency, above all at low min^{-1}.

Switch-over in the variable-geometry system uses flaps or similar devices to connect or isolate the different groups of cylinders as a function of engine speed.

At low min^{-1} the variable-length intake runners operate in conjunction with an initial resonance chamber. The length of the intake runners is adjusted continually as engine speed increases, the entire process culminating with the opening of a second resonance chamber.

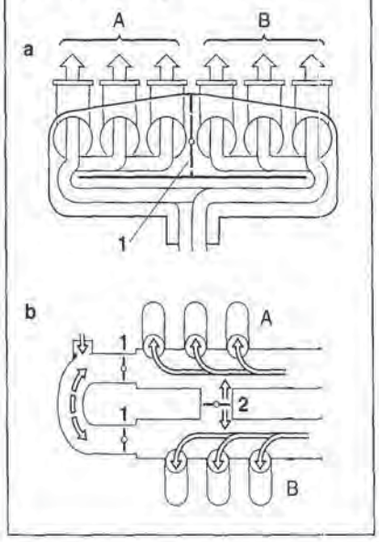
Mechanical supercharging

In mechanical supercharging, the supercharger is powered directly by the engine, which usually drives the supercharger at a fixed transmission ratio. Mechanical or electromagnetic clutches are often used to control supercharger activation.

A clear illustration of the relationship between the engine and the supercharger is provided by the diagram for pressure vs. volumetric flow rate, in which the pressure ratio π_c of the supercharger is plotted against volumetric flow rate $\dot{V}$.

The curves for unthrottled 4-stroke engines (diesel) are particularly descriptive because they contain sloped straight lines (engine mass flow characteristics) which represent increasing engine air-throughput values as the pressure ratio $\pi_c \triangleq p_2/p_1$ increases at constant engine speed.

The diagram shows pressure ratios which result at corresponding constant supercharger speeds for a positive-displacement supercharger and a hydrokinetic compressor.



Variable-geometry intake manifold
a) Two-stage operation.
b) Three-stage operation.
A, B Cylinder groups;
1, 2 Flaps, min^{-1}-regulated

PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

Only superchargers whose delivery rates vary linearly with their rotational speeds are suitable for vehicle engines. These are positive-displacement superchargers of piston or rotating-vane design or Roots blowers (see P. 440). Hydrokinetic flow compressors are not suitable.

Advantages of mechanical supercharging:
Relatively simple superchargers on "cold" side of engine. Engine exhaust gas is not involved. Supercharger responds immediately to load changes.

Disadvantages of mechanical supercharging:
The supercharger must be engine-powered, causing increased fuel consumption.

Exhaust-gas turbocharging
In exhaust-gas turbocharging, the energy for the turbocharger is extracted from the engine's exhaust gases. Although the process exploits energy that remains unused (owing to crankshaft assembly expansion limits) by naturally-aspirated engines, exhaust back-pressure increases as the gases furnish the power required to turn the compressor.

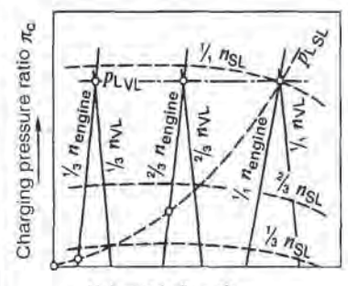
Current turbocharged engines employ an exhaust-driven turbine to convert the energy in the exhaust gas into mechanical energy, making it possible for the turbocharger to compress the induction gas.

The exhaust-gas turbocharger is a combined exhaust-driven turbine and flow compressor (see P. 442).

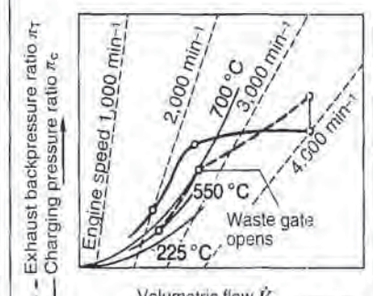
Exhaust-gas turbochargers are usually designed to generate a high boost pressure even at low engine speeds. Conversely, however, boost pressure at the high end of the engine min^{-1} range can increase to levels that could place excessive load on the engine. Engines with wide speed ranges in particular therefore require a waste gate bypassing the turbine, although this means a loss of exhaust-gas energy. Much more satisfactory results can be achieved with a compromise between high charge pressure at low min^{-1} and avoidance of engine overload at the high end of the min^{-1} range by employing Variable Turbine Geometry (VTG). The blading of a VTG turbine adjusts to suit the flow cross-section and thus the gas pressure at the turbine by variation of the flow cross-section.

Advantages of exhaust-gas turbocharging:
Considerable increase in power output per liter from a given configuration; improved torque curve within the effective engine-speed range; significant improvement in fuel consumption figures relative to naturally-aspirated engines with the same output power; improvement in exhaust-gas emissions.

Pressure vs. volumetric flow-rate map of mechanically driven positive-displacement supercharger and turbocharger
VL *Positive-displacement supercharger,*
SL *Hydrokinetic flow compressor.*



Pressure vs. volumetric flow-rate map of an exhaust-gas turbocharger showing boost pressure and exhaust backpressure curves



PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

Disadvantages of exhaust-gas turbocharging:
Installation of the turbocharger in the hot exhaust-gas tract requiring materials resistant to high temperatures; complexity and space requirements for turbocharger and intercooler; low base torque at low engine speed; throttle response is extremely sensitive to the efficient matching of the turbocharger to the engine.

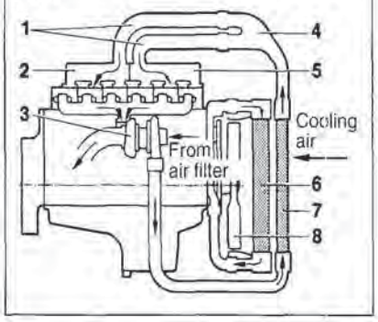
Pressure-wave supercharging
The pressure-wave supercharger uses direct energy exchange between exhaust gas and the intake air to increase the latter's density. This is accomplished by utilizing the differing speeds of the gas particles and pressure waves on the one side, and the reflection properties of these pressure waves on the other (see P. 446).

The pressure-wave supercharger consists of a cell rotor with an air casing on one side and an exhaust casing on the other. These incorporate specific timing edges and gas-pocket configurations.

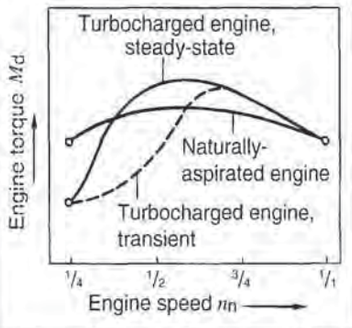
Advantages of pressure-wave supercharging:
Rapid throttle response because energy exchange between exhaust gas and boost air takes place at the speed of sound; high compression at low engine speeds.

Disadvantages of pressure-wave supercharging:
Restrictions on installation flexibility owing to the belt drive and gas lines; increased quantities of exhaust gas and scavenge air; loud operation; extremely sensitive to increased resistance on the low-pressure side.

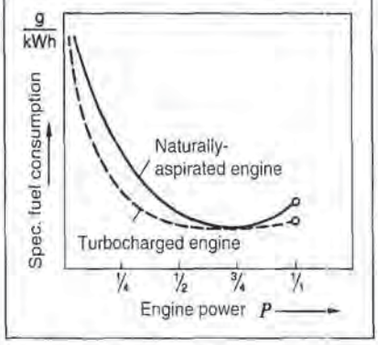
Truck diesel engine with exhaust-gas turbocharging, tuned-intake-tube supercharging and intercooler
1 Tuned tubes, 2 Resonance chamber for cylinders 4 – 5 – 6, 3 Turbocharger, 4 Balance chamber, 5 Resonance chamber for cylinders 1 – 2 – 3, 6 Radiator, 7 Intercooler, 8 Fan



Static and transient torque curves for turbocharged and naturally-aspirated engines



Specific part-load fuel consumption curves for atmospheric-induction and turbocharged engines of the same power



PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

Power transfer in reciprocating-piston engines

Engine types

Single-piston-power unit
The working chamber is formed by cylinder head, cylinder liner, and piston.

In-line engine (1)
Cylinders arranged consecutively in a single plane.

V-engine (2)
The cylinders are arranged in two planes in a V configuration.

Radial engine (3)
The cylinders are arranged radially in one or more planes.

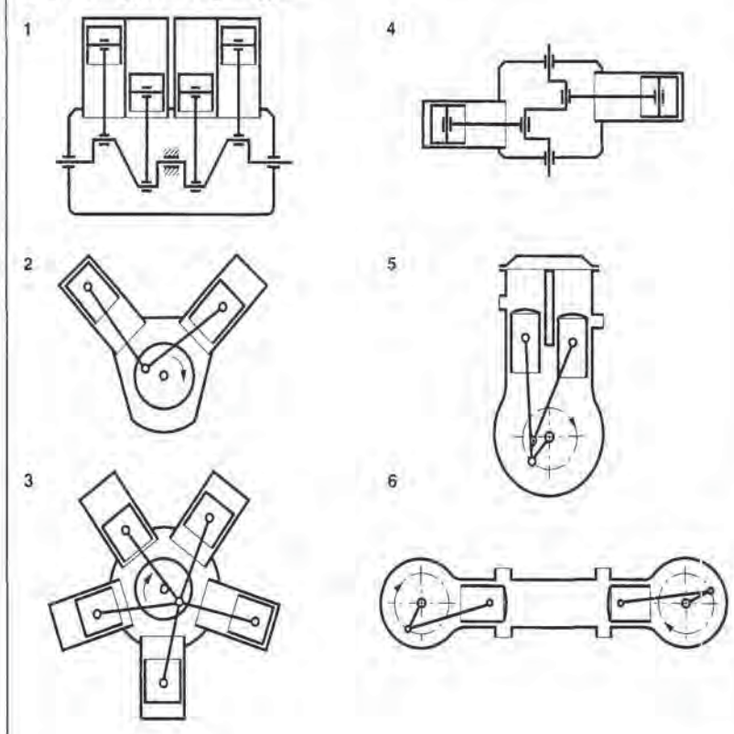
Opposed-cylinder (boxer) engine (4)
The cylinders are horizontally opposed.

Multi-piston-power unit
More than one (usually two) working pistons share a common combustion chamber.

U-engine (5)
The pistons move in the same direction.

Opposed-piston engine (6)
The pistons move in opposite directions.

Reciprocating-piston engine types



PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

Direction of rotation
(DIN 73021)[1])

Clockwise rotation: as viewed looking at the end of the engine opposite the power-output end. Abbreviation: cw.

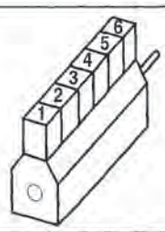
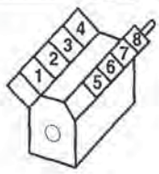
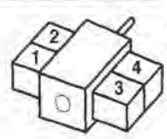
Counterclockwise rotation: as viewed looking at the end of the engine opposite the power-output end. Abbreviation: ccw.

Numbering the cylinders
(DIN 73021)[1])

The cylinders are numbered consecutively 1, 2, 3, etc. in the order in which they would be intersected by an imaginary reference plane. As viewed looking at the end of the engine opposite the power-output end. This plane is located horizontally to the left when numbering begins; the numerical assignments then proceed clockwise around the longitudinal axis of the engine (see figures below). If there is more than one cylinder in a reference plane, the cylinder nearest the observer is assigned the number 1, with consecutive numbers being assigned to the following cylinders. Cylinder 1 is to be identified by the number 1.

Firing order

The firing order is the sequence in which combustion is initiated in the cylinders. Engine design configuration, uniformity in ignition intervals, ease of crankshaft manufacture, optimal crankshaft load patterns, etc., all play a role in defining the firing order.

Design	Number of cylinders	Normal firing order (examples)
 Power output	4 5 6 8	1 3 4 2 or 1 2 4 3 1 2 4 5 3 1 5 3 6 2 4 or 1 2 4 6 5 3 or 1 4 2 6 3 5 or 1 4 5 6 3 2 1 6 2 5 8 3 7 4 or 1 3 6 8 4 2 7 5 or 1 4 7 3 8 5 2 6 or 1 3 2 5 8 6 7 4
 Power output	4 6 8	1 3 2 4 1 2 5 6 4 3 or 1 4 5 6 2 3 1 6 3 5 4 7 2 8 or 1 5 4 8 6 3 7 2 or 1 8 3 6 4 5 2 7
 Power output	4	1 4 3 2

[1]) Applies to motor-vehicle engines only. For internal-combustion engines for general and marine use, the reverse direction (as viewed looking at the power-output end) is standardized (ISO 1204 and 1205, DIN 6265).

PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

Crankshaft-assembly operation and dynamic properties

The purpose of the piston, connecting rod and crankshaft assembly in the reciprocating-piston engine is to transform the gas forces generated during combustion within the working cylinder into a piston stroke, which the crankshaft then converts into useful torque available at the power-output end of the engine. The cyclic principle of operation leads to unequal gas forces, and the acceleration and deceleration of the reciprocating power-transfer components generate inertial forces. It is usual to distinguish between internal and external effects of the gas-pressure and inertial forces.

The external effects, consisting of free forces or moments, impart movement to the engine. This is then transmitted to the engine supports in the form of vibration.

In this context, the smooth running of an engine is understood to mean freedom from low-frequency vibration, while quiet running means freedom from high-frequency, audible vibration.

The internal forces induce periodically variable loads in block, piston, connecting rod, crankshaft assembly and force-transfer components. These factors must be included in calculations for defining their dimensions and fatigue resistance.

Crankshaft assembly and gas force

The crankshaft assembly of a single-cylinder powerplant comprises the piston, connecting rod (conrod) and crankshaft. These components react to gas forces by generating mass inertial forces of their own.

The gas force F_G which acts on the piston can be subdivided into the side forces F_N applied by the piston to the cylinder wall and supported by it, and the connecting-rod force F_S. The connecting-rod force, in turn, causes the tangential force F_T to be applied at the crankshaft journal. This force together with the crank radius generates the shaft torque and the radial force F_R.

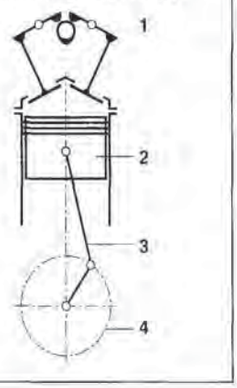
These forces can be calculated as a function of the gas force using the crank angle α, the pivoting angle of the connecting rod β and the stroke/connecting rod ratio λ:

Connecting-rod force: $F_S = F_G/\cos\beta$
Piston side force: $F_N = F_G \cdot \tan\beta$
Radial force: $F_R = F_G \cdot \cos(\alpha+\beta)/\cos\beta$
Tangential force: $F_T = F_G \cdot \sin(\alpha+\beta)/\cos\beta$
where $\lambda = r/l$; $\sin\beta = \lambda \cdot \sin\alpha$;

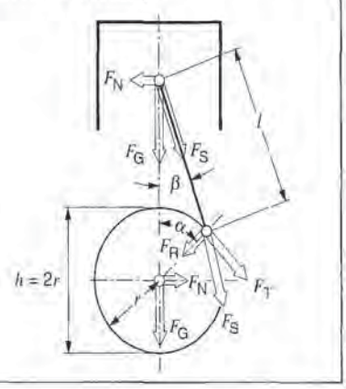
$$\cos\beta = \sqrt{1-\lambda^2 \cdot \sin^2\alpha}$$

All of these relationships can be represented in the form of a Fourier series, which can be useful in vibration calculations.

Piston and crankshaft assembly of the reciprocating-piston engine *(concept)*
1 Valve gear,
2 Piston,
3 Connecting rod,
4 Crankshaft



Gas-force components shown on a basic crankshaft assembly



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US PATENT NO. 6,581,565

Inertial forces and moments of inertia

The mass inertial properties of the piston, connecting rod and crankshaft assembly are a composite of the forces of the rotating masses of the crankshaft around their axis (x-axis) and the reciprocating masses in the cylinder direction (z-axis for in-line engines). With multiple-cylinder machines, free moments of inertia occur owing to different points of application for gas and inertial forces. The inertial properties of a single-cylinder engine can be determined using the piston mass m_K (exclusively oscillating mass), the crankshaft mass m_W (exclusively rotating mass) and the corresponding connecting-rod mass components (usually assumed to consist of oscillating and rotating rod masses amounting to one third and two thirds of the total mass respectively):

Oscillating mass

$$m_o = m_{Pl}/3 + m_K$$

Rotating mass

$$m_r = 2\,m_{Pl}/3 + m_W$$

The rotating inertial force acting on the crankshaft is as follows:

$$F_r = m_r \cdot r \cdot \omega^2$$

Oscillating inertial force:

$$F_o = m_o \cdot r \cdot \omega^2 \cdot (\cos\alpha + \lambda \cdot \cos 2\alpha + \ldots)$$

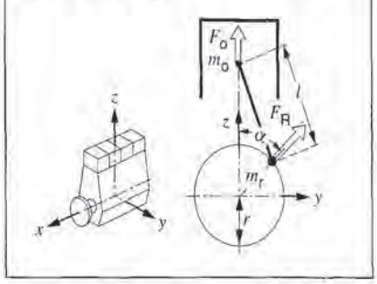
(1st Order; 2nd Order)

The following approximations also apply:

$$F_y = r \cdot \omega^2 \cdot m_r \cdot \sin\alpha$$

$$F_z = r \cdot \omega^2 \cdot [m_r \cdot \cos\alpha + m_o \cdot (\cos\alpha + \lambda \cdot \cos 2\alpha)]$$

Reference coordinates and inertial forces *(single-cylinder engine)*



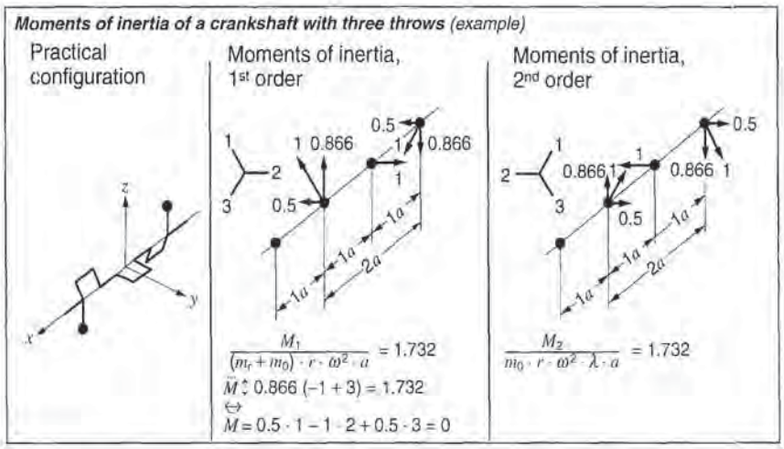
where $\lambda = r/l$

The inertial-force components are designated as inertial forces of the 1st, 2nd or 4th order, depending upon their rotational frequencies relative to engine speed.

In general, only the 1st and 2nd order components are significant; higher orders can be disregarded.

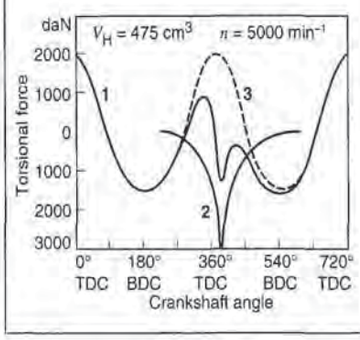
In the case of multiple-cylinder engines, free moments of inertia are present when all of the complete crankshaft assembly's inertial forces combine to produce a force couple at the crankshaft. The crankshaft assembly must therefore be regarded as a three-dimensional configuration when determining the free moments of inertia, while the inertial forces can be determined using a two-dimensional system.

Moments of inertia of a crankshaft with three throws *(example)*



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US PATENT NO. 6,581,565

Torsional-force diagram for a single-cylinder, four-stroke engine
1 Gas and inertial forces, 2 Gas forces, 3 Inertial forces.



Torsional-force diagram for the reciprocating-piston engine

If the periodic gas force acting on the piston and the periodic mass inertial forces acting on the piston, connecting rod and crankshaft assembly are grouped together, they generate a sum of tangential force components at the crankshaft journal. When multiplied by the crank radius, this yields a periodically variable torque value. If this torque value is referred to the piston surface and the crank radius, the result is a value valid for any engine size: tangential pressure. The torsional-force diagram shows the curve for this pressure as a function of crankshaft position. It is one of the most important characteristic curves in assessing dynamic engine behavior.

With multiple-cylinder engines, the tangential-pressure curves for the individual cylinders are superimposed with a phase shift in accordance with the number of cylinders, their configuration, crankshaft design and the firing order. The resulting composite curve is characteristic for the engine design, and covers a full working cycle (i.e., 2 crankshaft rotations for 4-stroke engines). It is also called a tangential-pressure diagram.

Harmonic analysis can be employed to replace the torsional-force diagram with a series of sinusoidal oscillations featuring whole-number multiples of the basic frequencies, and to obtain the "torsional harmonics". When defined according to engine speed these multiples are also called orders. When applied to a four-stroke engine this procedure generates half orders, e.g. the 0.5th order.

The cyclical fluctuations in torsional force encountered in all reciprocating-piston engines lead to variations in the crankshaft's rotation speed, the so-called coefficient of cyclic variation.

$$\delta_s = (\omega_{max} - \omega_{min})/\omega_{min}$$

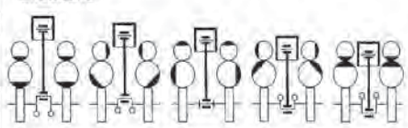
An energy storage device (the flywheel) provides adequate compensation for these variations in rotation rate in normal applications.

Complete 1st and 2nd order balancing of masses in a single-stroke system

Counterweights

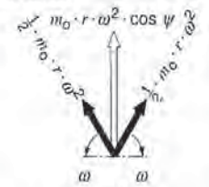


Unbalanced systems rotating in opposite directions



Vector diagrams of the 1st and 2nd order oscillating inertial forces

1st Order



2nd Order



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US PATENT NO. 6,581,565

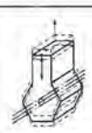
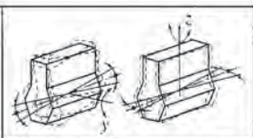
Balancing of masses in the reciprocating-piston engine

Mass balancing encompasses a wide array of measures employed to obtain partial or complete compensation for the inertial forces and moments of inertia emanating from the crankshaft assembly. All masses are externally balanced when no free inertial forces or moments of inertia are transmitted through the block to the outside. However, the remaining internal forces and moments subject the engine mounts and block to various loads as well as deformative and vibratory stresses. The basic loads imposed by gas-based and inertial forces are shown in Table 3.

Balancing of inertial forces in the single-stroke powerplant

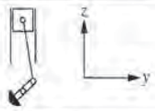
The simplest way to balance rotating masses is to use counterweights to generate an equal force to oppose the centrifugal force. Oscillating masses generate periodic forces. The 1st order forces are propagated at crankshaft speed, while the periodicity of the 2nd order forces is twice the crankshaft's rotation rate. Compensation for these forces is available in the form of a counterweight balance system designed for opposed rotation at a rate equal to or twice that of the crankshaft. The balance forces' magnitudes must equal those of the rotating inertial-force vectors while acting in the opposite direction.

Table 3. Forces and moments applied to the piston, connecting rod and crankshaft assembly

Forces and moments at the engine				
Designation	Oscillating torque, transverse tilting moment, reaction torque	Free inertial forces	Free inertial moment, longitudinal tilting moment about the y-axis (transverse axis) ("pitching" moment) about the z-axis (vertical axis) ("rolling" moment)	Internal flex forces
Cause	Tangential gas forces as well as tangential inertial forces for ordinals 1, 2, 3 and 4	Unbalanced oscillating inertial forces 1st order in 1 and 2 cylinders; 2nd order in 1, 2, and 4 cylinders	Unbalanced oscillating inertial forces as a composite of 1st and 2nd order forces	Rotating and oscillating inertial forces
Design factors	Number of cylinders, ignition intervals, displacement, p_i, ε, p_z, m_0, r, ω, λ	Number of cylinders, crank configuration m_0, r, ω, λ	Number of cylinders, crank configuration, cylinder spacing, counterweight size influences inertial torque components about the y- and z-axes m_0, r, ω, λ, a	Number of throws, crank configuration, engine length, engine-block rigidity
Remedy	Can only be compensated for in exceptional cases	Free mass effects can be eliminated with rotating balancing systems, however this process is complex and therefore rare; crank sequences with limited or no mass effects are preferable		Counterweights, rigid engine block
	Shielding of the environment through flexible engine mounts (in particular for orders ≥ 2)			

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US PATENT NO. 6,581,565

Table 4. Residual 1st order inertial forces with differing balancing rates

		Balancing rate		
		0%	50%	100%
Size of counter-weight	$m_G \triangleq$	m_r	$m_r + 0.5m_0$	$m_r + m_0$
Residual inertial force (z) 1st order	$F_{1z} =$	$m_0 \cdot r \cdot \omega^2$	$0.5 \cdot m_0 \cdot r \cdot \omega^2$	0
Residual inertial force (y) 1st order	$F_{1y} =$	0	$0.5 \cdot m_0 \cdot r \cdot \omega^2$	$m_0 \cdot r \cdot \omega^2$

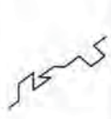
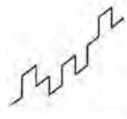
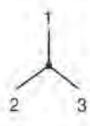
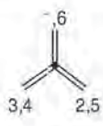
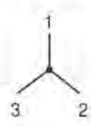
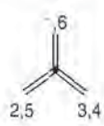
Balancing rate

The forces exerted by the counterweights used to balance the rotating masses can be increased by a certain percentage of the oscillating mass in order to reduce the oscillating forces acting in the direction of the cylinders (z). The percentage of this inertial force which is counteracted then appears in the y-axis. The ratio of the compensated inertial-force component in the z-axis relative to the initial value for the 1st order inertial force is termed the balancing rate (Table 4).

Balancing of inertial forces in the multi-cylinder engine

In multi-cylinder engines the mutual counteractions of the various components in the crankshaft assembly are one of the essential factors determining the selection of the crankshaft's configuration, and with it the design of the engine itself. The

Table 5. Star diagram of the 1st and 2nd order for three- to six-cylinder, in-line engines

	3-cylinder	4-cylinder	5-cylinder	6-cylinder
Crank sequence				
Star diagram 1st Order				
Star diagram 2nd Order				

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US PATENT NO. 6,581,565

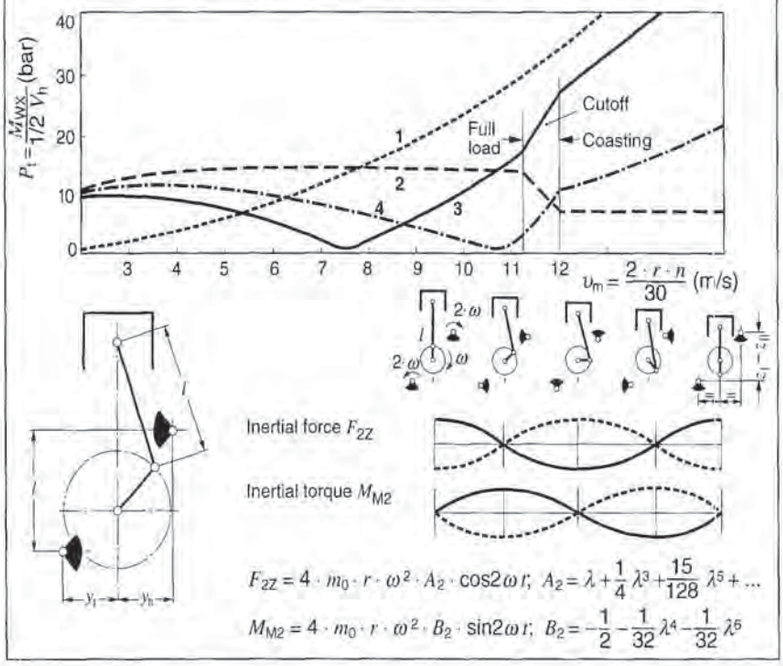
inertial forces are balanced if the common center of gravity for all moving crankshaft-assembly components lies at the crankshaft's midpoint, i.e. if the crankshaft is symmetrical (as viewed from the front). The crankshaft's symmetry level can be defined using geometrical representations of 1st- and 2nd-order forces (star diagrams). The 2nd order star diagram for the four-cylinder in-line engine is asymmetrical, meaning that this order is characterized by substantial free inertial forces. These forces can be balanced using two countershafts rotating in opposite directions at double the rate of the crankshaft (Lanchester system).

Balancing of inertial and gas forces

The tangential gas forces produce yet another periodic torque; this can be detected as reaction torque in the engine block. The composite forces generated in a four-cylinder in-line engine include free mass forces of the 2nd order as well as variable torque forces from the 2nd order mass and gas forces. Compensation for 2nd order mass forces, along with a reduction in the intensity of the 2nd order force transitions, is available from two offset balance shafts.

Balancing 2nd order inertial and transitional forces in a four-cylinder, in-line engine with two offset countershafts

1 Inertial torque only; 2 Gas torque only or complete balancing of inertial torque, $z_I - z_{II} = -2\,B_2/A_2 \cdot r$; 3 Gas and inertial torque without force compensation; 4 Gas and inertial torque with half of the inertial torque balanced, $z_I - z_{II} = 0.5 \cdot I$.



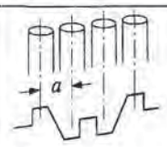
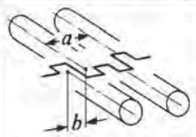
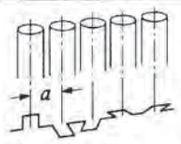
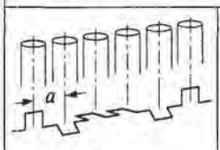
$$F_{2Z} = 4 \cdot m_0 \cdot r \cdot \omega^2 \cdot A_2 \cdot \cos 2\omega t;\quad A_2 = \lambda + \frac{1}{4}\lambda^3 + \frac{15}{128}\lambda^5 + \ldots$$

$$M_{M2} = 4 \cdot m_0 \cdot r \cdot \omega^2 \cdot B_2 \cdot \sin 2\omega t;\quad B_2 = -\frac{1}{2} - \frac{1}{32}\lambda^4 - \frac{1}{32}\lambda^6$$

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US PATENT NO. 6,581,565

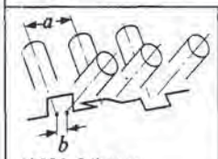
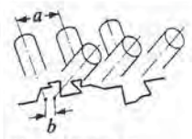
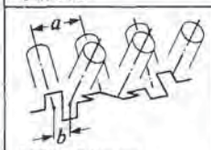
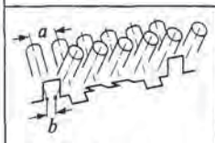
Table 6. Free forces and moments of the 1st and 2nd order, and ignition intervals of the most common engine designs

$F_r = m_r \cdot r \cdot \omega^2 \quad F_1 = m_0 \cdot r \cdot \omega^2 \cdot \cos\alpha \quad F_2 = m_0 \cdot r \cdot \omega^2 \cdot \lambda \cdot \cos 2\alpha$

Cylinder arrangement	Free forces of 1st order[1])	Free forces of 2nd order	Free moments of 1st order[1])	Free moments of 2nd order	Ignition intervals
3-cylinder					
 In-line, 3 throws	0	0	$\sqrt{3} \cdot F_1 \cdot a$	$\sqrt{3} \cdot F_2 \cdot a$	240°/240°
4-cylinder					
 In-line, 4 throws	0	$4 \cdot F_2$	0	0	180°/180°
 Opposed-cylinder (boxer), 4 throws	0	0	0	$2 \cdot F_2 \cdot b$	180°/180°
5-cylinder					
 In-line, 5 throws	0	0	$0.449 \cdot F_1 \cdot a$	$4.98 \cdot F_2 \cdot a$	144°/144°
6-cylinder					
 In-line, 6 throws	0	0	0	0	120°/120°

[1]) Without counterweights.

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US PATENT NO. 6,581,565

Cylinder arrangement	Free forces of 1st order[1])	Free forces of 2nd order	Free moments of 1st order[1])	Free moments of 2nd order	Ignition intervals
6-cylinder (continued)					
 V 90°, 3 throws	0	0	$\sqrt{3}\cdot F_1\cdot a$ [2])	$\sqrt{6}\cdot F_2\cdot a$	150°/90° 150°/90°
 Normal balance V 90°, 3 throws, 30° crank offset	0	0	$0.4483\cdot F_1\cdot a$ 	$(0.966\pm0.256)\cdot\sqrt{3}\cdot F_2\cdot a$ 	120°/120°
 Opposed-cylinder, 6 throws	0	0	0	0	120°/120°
 V 60°, 6 throws	0	0	$3\cdot F_1\cdot a/2$	$3\cdot F_2\cdot a/2$	120°/120°
8-cylinder					
 V 90°, 4 throws in two planes	0	0	$\sqrt{10}\cdot F_1\cdot a$ [2])	0	90°/90°
12-cylinder					
 V 60°, 6 throws	0	0	0	0	60°/60°

[1]) Without counterweights.
[2]) Can be completely balanced by using counterweights.

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US PATENT NO. 6,581,565

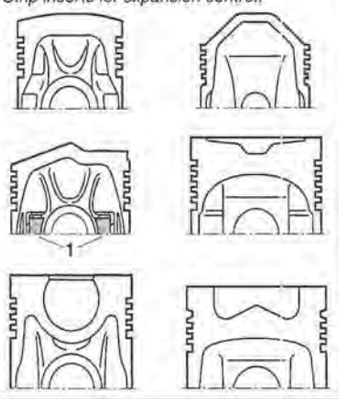
Main components of reciprocating-piston engine

Piston

Pistons in today's motor-vehicle engines must perform a wide range of functions:
- They transmit the force generated by the combustion gas to the connecting rods.
- They serve as crosstails to define the connecting rods' travel paths within the cylinders.
- They support the normal force applied against the cylinder walls while the cylinder pressure is conveyed to the connecting rod.
- Together with their sealing elements, they seal the combustion chamber from the crankcase.
- They absorb heat for subsequent transfer to the cooling system.

Both the piston's design and the wristpin configuration employed to transfer the combustion gas forces to the connecting rod are largely determined by the combustion chamber's shape, including the geometry of the piston crown, while other variables include the selected combustion process and the associated pressure maxima. The priority is to produce the lightest possible piston in a unit capable of withstanding intense forces during operation in an environment with temperatures that can approach the physical limits of the materials used in its manufacture.

Piston shapes in various engine designs
1 Strip inserts for expansion control.

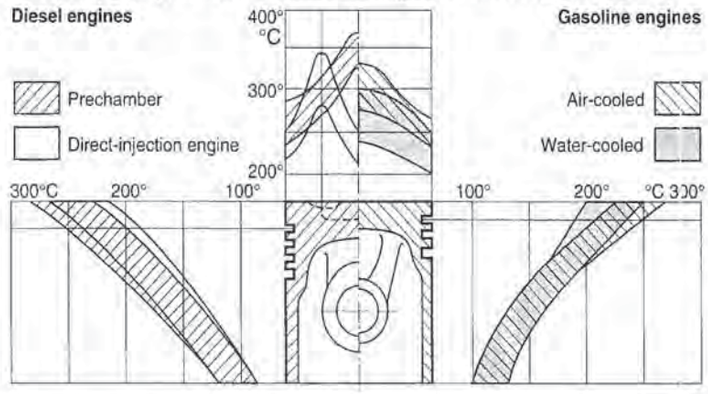


Precise definition of the dimensions for the piston, wristpin and wristpin bushings are essential for achieving this goal.

The most frequently used materials for cylinder liners and pistons are gray cast iron and aluminum. Variations in piston clearance within the cylinder must be minimized to reduce noise (piston slap) and improve sealing, notwithstanding the fact that piston and cylinder liner have different coefficients of expansion. To this end, steel strips or similar elements are sometimes cast into the piston to limit its expansion.

Piston rings form the sealing element between the combustion chamber and the crankcase. The upper two – the compression rings – serve as gas seals. At

Piston operating temperatures in motor-vehicle engines at full load (schematic)

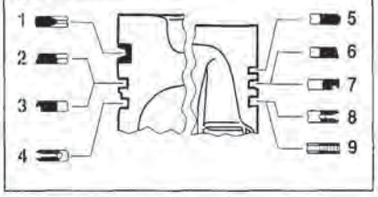


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US PATENT NO. 6,581,565

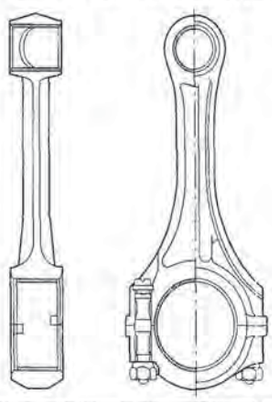
Piston-ring shapes and configurations

Diesel engine: *1 Keystone ring, crowned, 2 Taper-face compression ring with inner bevel, 3 Stepped compression ring, 4 Double-beveled ventilated oil control ring with spiral-type expander.*

Spark-ignition engine: *5 Plain compression ring, crowned, 6 Taper-face compression ring, 7 Stepped ring, 8 Double-beveled ring, 9 Multipart steel oil ring.*



Passenger-car engine connecting rod

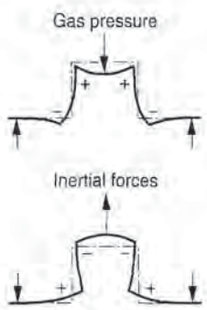


least one additional ring (generally of a different design), the oil control ring, is also present. This is a "scraper" ring and ensures correct lubrication of the piston and compression rings. Owing to the rings' extreme tension and the force that they exert against the cylinder walls, they are a major source of friction within the reciprocating-piston engine.

Connecting rod

The connecting rod (conrod) is the joining element between piston and crankshaft. It is subject to extreme tensile, compression and flex stresses, while it also houses the wristpin bushings and crankshaft bearings. Connecting-rod length is determined by the piston stroke and the counterweight radius; whereby the engine height can also be an important factor (usually the case in vehicle engines).

Crankshaft throw
Primary stresses and deformations due to gas pressure and inertial forces.



Crankshaft

The crankshaft with its rod extensions, or throws, converts the reciprocating motion of the pistons – conveyed to it by the connecting rods – into rotary motion, making effective torque available at the crankshaft's end. The forces acting upon the crankshaft are characterized by highly variable periodicities and vary greatly according to location. These torques and flex forces, and the secondary vibrations which they generate, all represent intense and highly complex stress factors for the crankshaft itself. As a result, its structural properties and vibrational response patterns rely upon precise calculations and carefully defined dimensions. Calculations and dimensioning though are further complicated by the fact that too many multiple journal bearings are practically always installed as a precautionary measure.

The number of crankshaft bearings is primarily determined by overall load factor and maximum engine speed. To accommodate their intense operating pressures, all diesel-engine crankshafts incorporate a main bearing journal between each crankshaft throw and at each end of the crankshaft. This arrangement is also found in high-speed spark-ignition (SI) en-

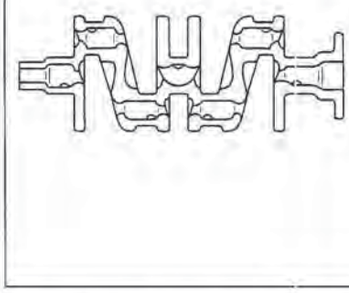
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US PATENT NO. 6,581,565

gines designed for high specific outputs.

Crankshafts in some smaller SI engines designed for operation at lower load factors sometimes extend the interval between main bearings to 2 cylinders to reduce expense. The number of counterweights also depends upon the criteria cited above.

Stresses and load factors are also primary considerations in the selection of both materials and manufacturing processes. Highly-stressed crankshafts are usually drop-forged. In smaller and less highly stressed engines, cast crankshafts, incorporating the dual advantages of lower weight and less expense, are becoming increasingly popular.

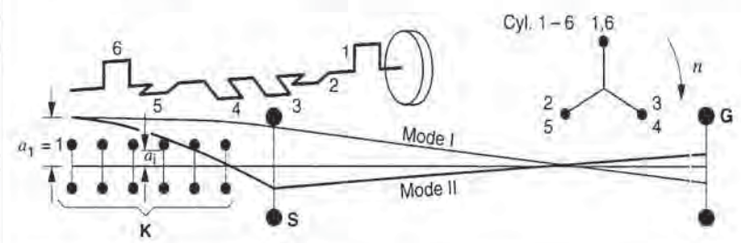
Cast crankshaft



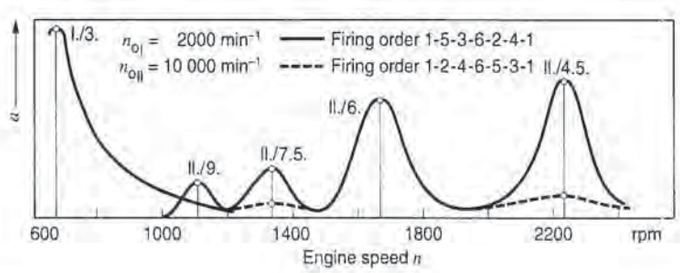
Crankshaft vibrations

Flexural vibration is significant only on engines with a small number of cylinders, because the crankshaft and the necessary large flywheel form an oscillatory system with a low natural frequency. Flexural vibration is not a critical factor on engines of 3 cylinders or more. By logical extension, this also applies to the longitudi-

Vibrational schematic of a 6-cylinder crankshaft **(K)** ***with flywheel*** **(S)** ***and transmission*** **(G)**
a Relative amplitudes, n Engine speed.



Order analysis of the crankshaft vibrations of a 6-cylinder engine,
with differing firing orders, a Relative amplitudes



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US PATENT NO. 6,581,565

nal crankshaft vibrations induced by flexural vibrations.

At the same time, the torsional vibrations of the resonant system formed by crankshaft, connecting rods and pistons become increasingly critical with higher numbers of cylinders. This system, in which the mass moments of inertia for connecting rods and pistons vary according to crankshaft angle, can be calculated by reducing it to a smooth, flexible shaft free of inertia with equivalent masses mounted on it. The oscillation reduction model makes it possible to determine both the system's inherent frequency and the intensity of the vibration forces. The oscillations emanate from the tangential forces generated by a combination of gas forces and oscillating mass forces at the crank pin. Vibration dampers are required to reduce the crankshaft's torsional vibrations to acceptable levels (e.g. bonded rubber vibration dampers or viscous vibration dampers).

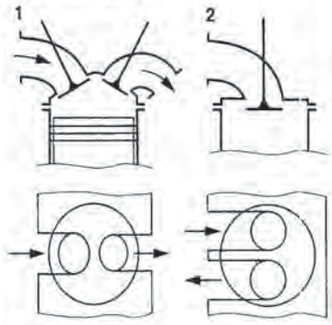
Engine block and crankcase

The block and crankcase unit supports the force-transfer mechanism between cylinder head and crankshaft assembly; it bears the crankshaft assembly's support bearings, and incorporates (or holds) the cylinder sleeves. Also included in the block are a separate water jacket and sealed oil chambers and galleries. The block also serves as a mounting and support surface for most of the engine's ancillary units.

A cast block and crankcase unit is the standard configuration for automotive applications. The cylinder-head bolts oppose the gas forces to facilitate a force transfer of maximum linearity and minimal flexural tendency through transverse support walls and to the main bearings. For greater strength the crankcase is frequently extended to below the crankshaft's center axis. The pistons in spark-ignition engines almost always run in integral cylinders machined from the block casting. In diesel engines, separate dry or wet liners made of special wear-resistant materials are usually used.

Whereas, virtually all blocks for truck engines continue to be manufactured in cast iron, aluminum passenger-car blocks are becoming increasingly popular owing to their weight-savings potential.

Cylinder-head designs according to intake and exhaust tract location
1 Crossflow design,
2 Counterflow design.



Cylinder head

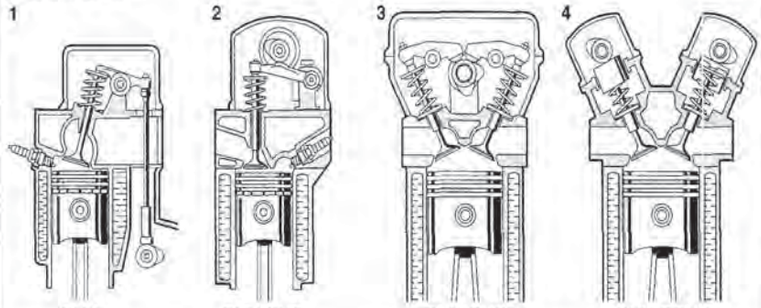
The cylinder head seals off the upper end of the block and cylinder(s). It houses the gas-exchange valves as well as the spark plugs and/or injectors. Together with the piston, it also provides the desired combustion-chamber shape. In the vast majority of passenger-car engines, the entire valve gear is also mounted in the cylinder head. Based on the gas-exchange concepts, one differentiates between two basic design configurations:

- Counterflow cylinder head: Intake and exhaust passages open onto the same side of the cylinder head. This limits the space available for the intake and exhaust gas passages, but due to the short flow tracts this represents a substantial advantage in supercharged applications. This design, with the gas supply and discharge tracts on a single side, also provides practical advantages in transverse-mounted engines.
- Crossflow cylinder head: Intake and exhaust passages are located on opposite sides of the engine, furnishing a diagonal flow pattern for the intake and exhaust gases. This layout's advantages include more latitude in intake and exhaust-tract design as well as less complicated sealing arrangements.

In truck and large industrial engines, individual cylinder heads are often used on each cylinder for better sealing-force distribution and easier maintenance and re-

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US PATENT NO. 6,581,565

Valve-gear designs (source: Hütten "Motoren")
1 Push-rod assembly, 2 Single rocker-arm assembly, 3 Twin rocker-arm assembly, 4 Overhead bucket-tappet assembly, OHV Overhead valves, OHC Overhead camshaft, DOHC Double overhead camshaft.



pair. Separate cylinder heads are also specified for improved cooling efficiency on air-cooled engines.

In passenger-car and low-power engines, one cylinder head is usually employed for all cylinders together. The cylinder heads on water-cooled diesel truck engines are usually made of cast iron.

Superior heat dissipation and lower weight have combined to make aluminum the material of choice in the construction of cylinder heads for air-cooled engines as well as on virtually all spark-ignition and diesel engines for passenger cars.

Valve gear

It is the function of the valve-gear assembly in a 4-stroke engine to permit and to control the exchange of gases in the IC engine (see P. 384). The valve gear includes the intake and exhaust valves, the springs which close them, the camshaft drive assembly and the various force-transfer devices.

Valve-actuation concepts

In the following widely-used designs the camshaft is located in the cylinder head:

- Overhead bucket-tappet assembly, in which a "bucket" moving back and forth in the cylinder head absorbs the cam lobe's lateral force while transferring its linear actuating pressure to the valve stem.
- Cam follower or single rocker-arm assembly actuated by an overhead cam, in which the cam lobe's lateral and linear forces are absorbed and relayed by a cylinder-head-mounted lever rocking back and forth between cam lobe and valve. In addition to transferring forces, the rocker arm can also be designed to magnify the effective lobe profile for greater valve travel.
- Twin rocker-arm assembly actuated by overhead cam, in which the rocker arm's tilt axis is located between the camshaft and the valve. Here too, the rocker arm is usually designed as a cam lift multiplier to produce the desired valve travel.

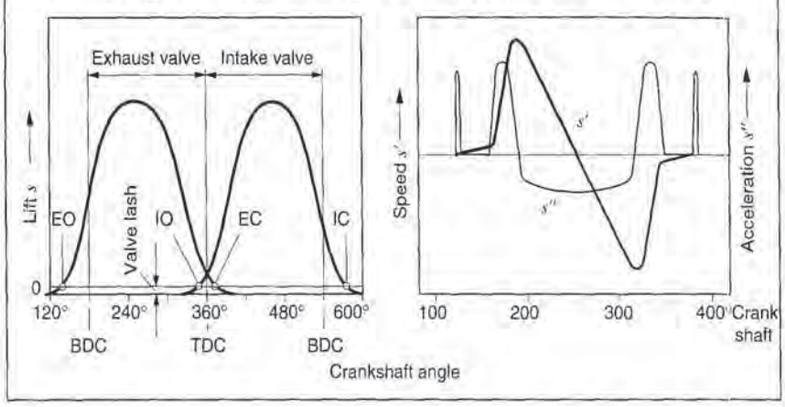
When the camshaft is installed within the block, the camshaft lobe acts against an intermediate lifter and pushrod assembly instead of directly against the valve.

Valve arrangements

The valve control arrangement and the design of the combustion chamber are closely interrelated. Today, nearly all valve assemblies are overhead units mounted in the cylinder head. In diesels and simpler spark-ignition engines, the valves are parallel to the cylinder axis, and are usually actuated by twin rocker arms, bucket tappets or single rocker arms. With increasing frequency, those current spark-ignition engines designed for higher specific outputs tend to feature intake and exhaust valves which are inclined towards each other. This configuration allows larger valve diameters for a given cylinder bore while also providing greater latitude

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US PATENT NO. 6,581,565

Valve timing diagram showing valve lift (s), valve velocity (s') and valve acceleration (s")



for optimizing intake and exhaust passage design. Twin rocker-arm assemblies actuated by overhead cams are used most often here. High-performance and racing engines are increasingly using four valves per cylinder and overhead-bucket-tappet valve assemblies.

An engine's valve-timing diagram shows the opening and closing times of the valves, the valve-lift curve with maximum lift, and the valve's velocities and acceleration rates.

Typical valve acceleration rates for passenger-car OHC (overhead camshaft) valve assemblies:

$s'' = 60 \ldots 65$ mm $(b/\omega^2) \triangleq 6400$ m/s² at 6000 min⁻¹ for single and twin rocker-arm assemblies,

$s'' = 70 \ldots 80$ mm $(b/\omega^2) \triangleq 7900$ m/s² at 6000 min⁻¹ for overhead-bucket tappet assemblies. For heavy commercial-vehicle engines with block-mounted camshafts:

$s'' = 100 \ldots 120$ mm $(b/\omega^2) \triangleq 2000$ m/s² at 2400 min⁻¹.

Valve, valve guide and valve seat

The materials employed in manufacturing valves are heat and scale-resistant. The valve seat's contact surface is frequently hardened. A proven method for improving the thermal transfer characteristics of exhaust valves is to fill their stems with sodium. To extend service life and improve sealing, valve-rotating systems (rotocaps) are now in common use.

The valve guides in high-performance engines must feature high thermal conductivity and good antifriction properties. They are usually pressed into the cylinder head and are often supplemented by valve stem seals at their cold ends for reducing oil consumption.

Valve-seat wear is generally reduced by making the valve seats of cast or sintered materials and shrink-fitting them into the cylinder head.

Lobe design and timing dynamics

The cam lobe must be able to open (and close) the valve as far as possible, as fast as possible and as smoothly as possible. The closing force for the valves is applied by the valve springs, which are also responsible for maintaining contact between the cam lobe and the valve. Dynamic forces impose limits on cam and valve lift.

The entire valve-gear assembly can be viewed as a spring/mass system in which the conversion from stored to free energy causes forced vibration. Valve-gear assemblies with overhead camshafts can be represented with sufficient accuracy by a 1-mass system (consisting of the propelled mass, valve-gear assembly stiffness, and the corresponding damping effects).

Dual-mass systems are becoming increasingly popular for use with block-mounted camshafts and pushrods.

The maximum permissible surface pressure, usually regarded as the decisive parameter limiting cam-lobe radius and the rate of opening on the flank, currently lies

PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

between 600 and 750 N/mm², depending upon the employed material pairings.

Cooling

In order to avoid thermal overload, combustion of the lubricating oil on the piston's sliding surface, and uncontrolled combustion due to excessive component temperatures, the components surrounding the hot combustion chamber (cylinder liner, cylinder head, valves and in some cases the pistons themselves) must be intensively cooled.

Direct cooling

Direct air cooling removes heat directly from the components. The underlying principle is based on intensive air flow, usually through a finned surface. Although primarily used in motorcycle and aircraft engines, this form of cooling is also employed for some passenger-car and commercial-vehicle diesel and spark-ignition engines. Its main advantage is its high reliability and freedom from maintenance. On the negative side, the design measures required to ensure efficient heat dissipation to the cooling air increase the cost of the components.

Indirect cooling

Because water has a high specific heat capacity and provides efficient thermal transition between the materials, most contemporary vehicle engines are water-cooled. The air/water recirculation cooling system is the most prevalent system. It comprises a closed circuit allowing the use of anti-corrosion and anti-freeze additives. The coolant is pumped through the engine and through an air/water radiator. The cooling air flows through the radiator in response to vehicle movement and/or is forced through it by a fan. The coolant temperature is regulated by a thermostatic valve which bypasses the radiator as required.

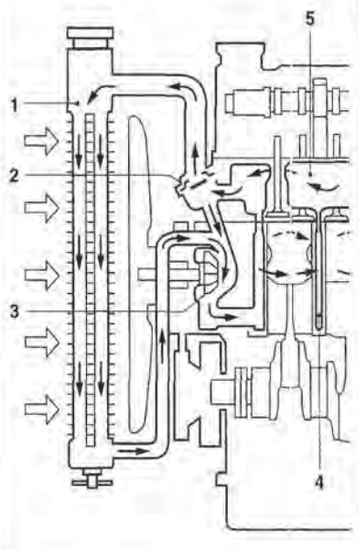
Lubrication

The internal-combustion engine employs oil to lubricate and cool all of the power-transmission components. This oil is also used to remove dirt and neutralize chemically-active combustion products, as well as for transmitting forces and damping vibration. The oil can only fulfill all these requirements if it is transported in adequate quantities to the engine's critical points, and if its properties are adapted to the specific requirements by appropriate measures taken during manufacture (e.g., inclusion of additives).

In total-loss lubrication (fresh-oil lubrication), a metering system supplies oil to the lubrication points, where it is subsequently consumed. A special case of this type of lubrication is mixture lubrication in which oil is either added to the fuel in a ratio ranging from 1:20 to 1:100, or metered to the engine (this process is used primarily in small two-stroke engines).

In most motor-vehicle engines, force-feed lubrication systems are used in combination with splash and oil mist lubrication. The basic force-feed system pumps pressurized oil (usually by gear pump) to all bearing points, while sliding parts are lubricated by splash lubrication systems

Water cooling system with coolant circuit
1 Radiator, 2 Thermostat, 3 Water pump, 4 Water passages in cylinder block, 5 Coolant passages in cylinder head.



PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

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

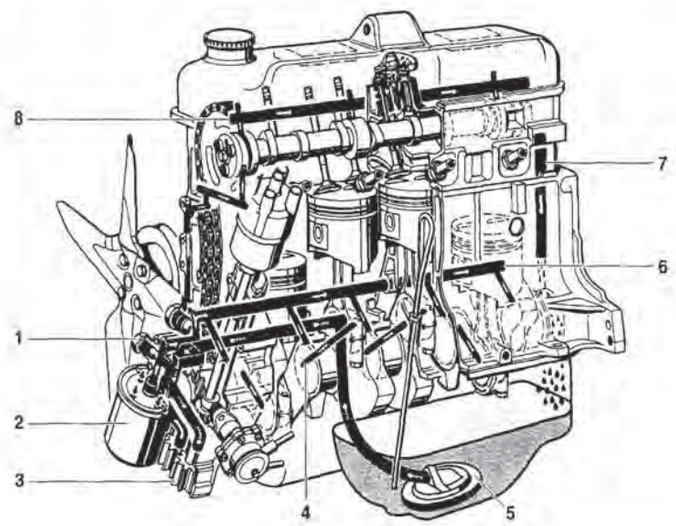
Oil filters remove solid particles from the engine oil (combustion residue, metal particles and dust) and maintain the lubricating capability of the oil in the intervals between changes. Oil-filter dimensions are governed by the degree to which impurities are generated in the engine and by the maintenance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

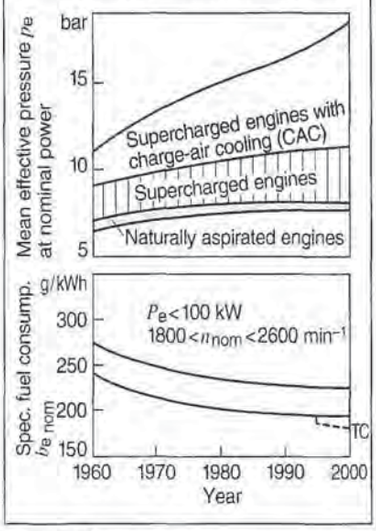


PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

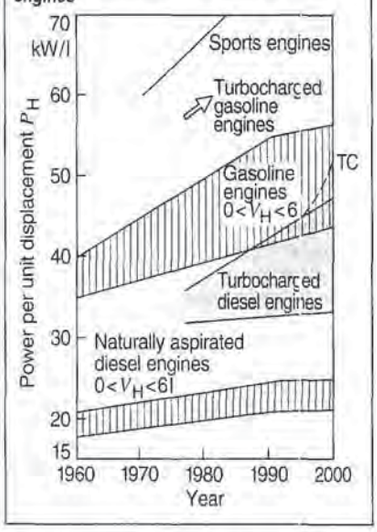
Empirical values and data for calculation

Comparisons

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

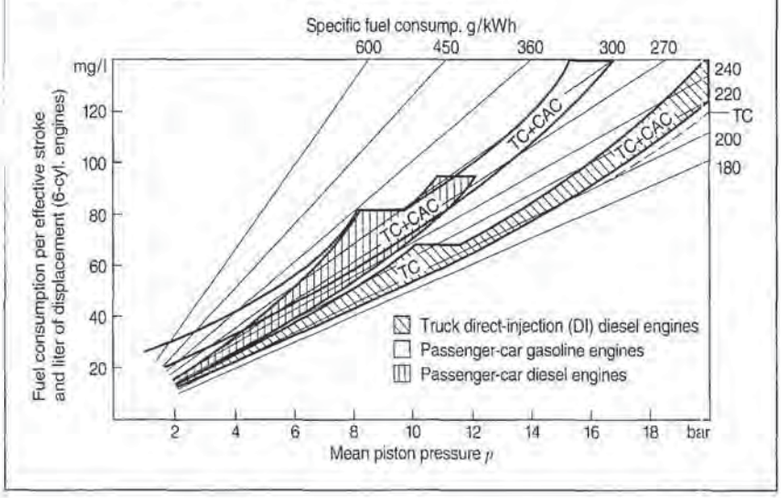


Trend in power per unit displacement for diesel and spark-ignition passenger-car engines



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

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



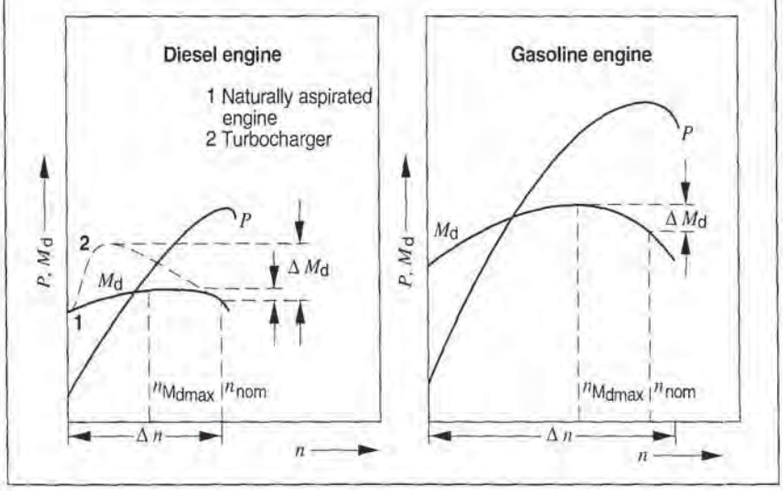
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Comparative data

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.

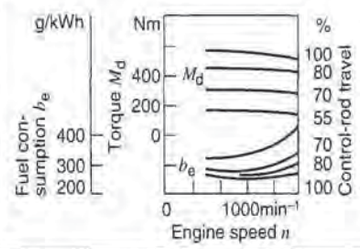
Power and torque curves



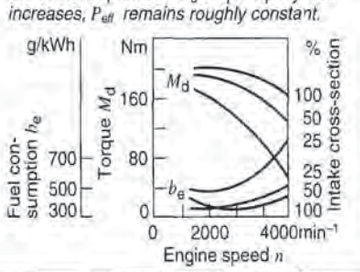
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US PATENT NO. 6,581,565

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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US PATENT NO. 6,581,565

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_{PE} 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	$\alpha_a = A^{1.2} \cdot B^{0.6}$ $A = 99/p_{PT}$ $B = T_p/298$				$\alpha_a = A \cdot B^{0.5}$ $A = 101.3/p_{PE}$ $B = T/293$
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

PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

Calculation

Quantity		Unit
a_K	Piston acceleration	m/s²
B	Fuel consumption	kg/h; dm³/h
b_e	Spec. fuel consumption	g/kWh
D	Cylinder diameter 2 · 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/dm³
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	mm²
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	m³

[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).

Symbol		Unit
V_c	Cylinder compression volume	dm³
V_E	Injected quantity per pump stroke	mm³
V_t	Charge volume of a cylinder (2-stroke)	dm³
V_F	Charge volume of a 2-stroke engine	dm³
V_h	Displacement of a cylinder	dm³
V_H	Displacement of the engine	dm³
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/m³
φ, α	Crank angle (φ_0 = 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

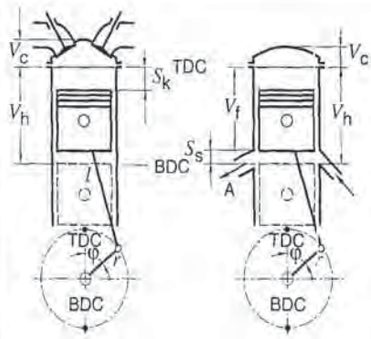
Conversion of units

(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

PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

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 · v_m | 1.62 · v_m | 1.61 · 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".

PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

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 $m_p = \frac{m}{P_{eff}}$	

PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

Calculation equations (continued)

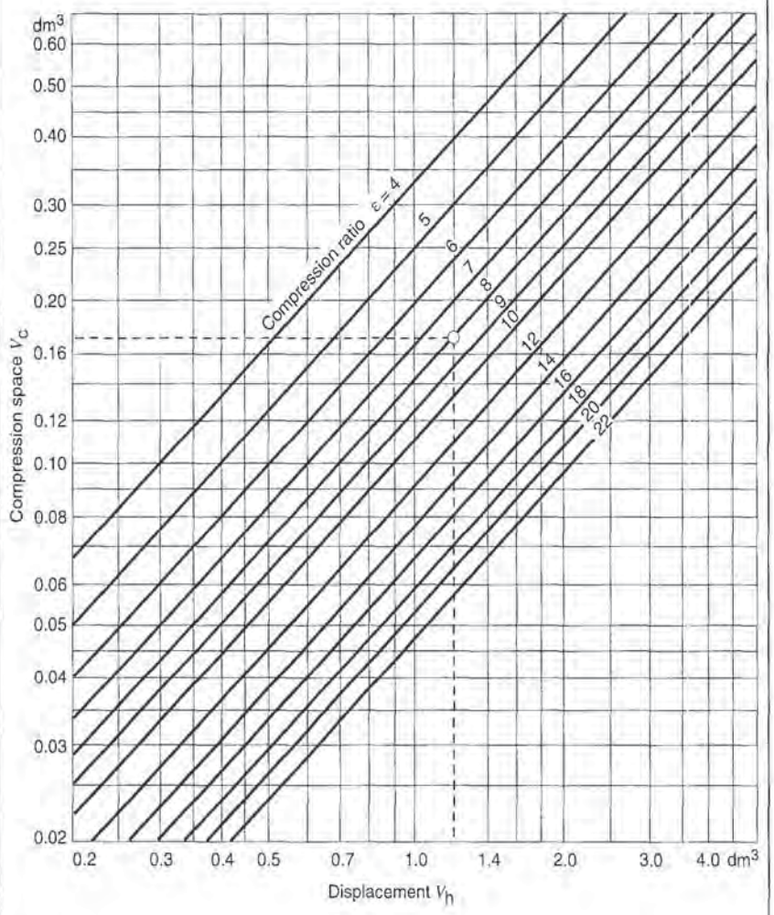
Mathematical relationship between quantities		Numerical relationship between quantities	
Mean piston pressure (mean pressure, mean working pressure)			
4-stroke engine $p = \frac{2 \cdot P}{V_H \cdot n}$	2-stroke engine $p = \frac{P}{V_H \cdot n}$	4-stroke engine $p = 1200 \frac{P}{V_H \cdot n}$	2-stroke 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.

PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

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.

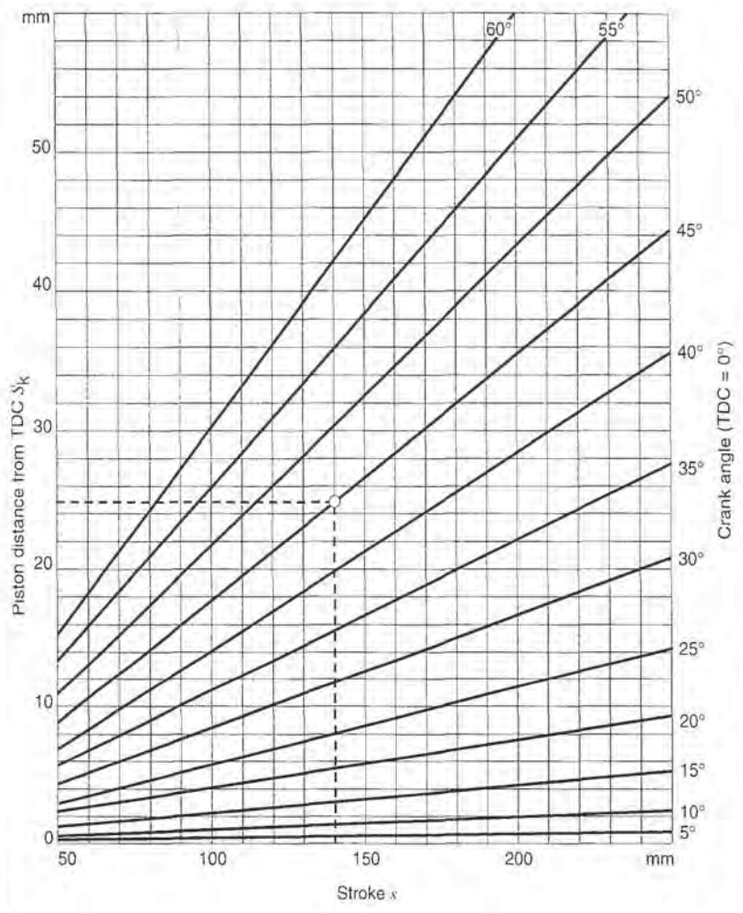


Example:
An engine with a displacement of 1.2 dm³ and a compression ratio $\varepsilon = 8$ has a compression area of 0.17 dm³.

PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

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

Conversion of degrees crank angle to mm piston travel.

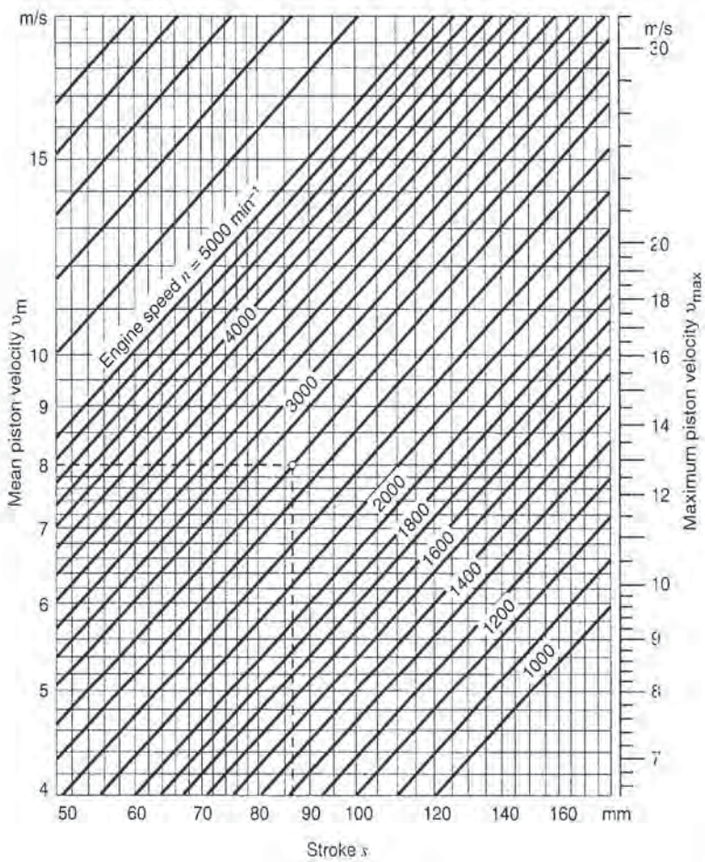


Example:
The piston clearance from top dead center is 25 mm for a stroke of 140 mm at 45° crankshaft.

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.

PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

Piston velocity *See P. 417 for equation.*



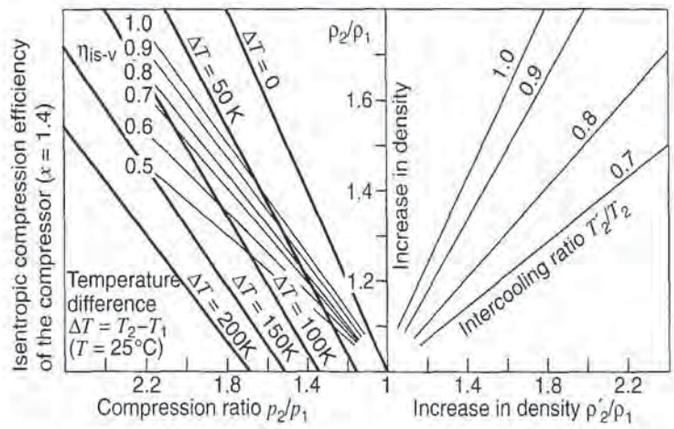
Example:
Mean piston velocity v_m = 8 m/s and maximum piston velocity v_{max} = 13 m/s for stroke s = 86 mm and at an engine speed of n = 2800 min^{-1}.

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

PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

Charge density increase in the cylinder with turbo/supercharging

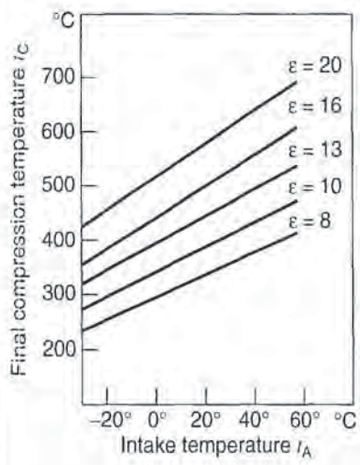
Increase in density on supercharging as a function of the pressure ratio in the compressor, the compressor efficiency and the intercooling rate for charge-air cooling (CAC).



$p_2/p_1 = \pi_c$ = 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

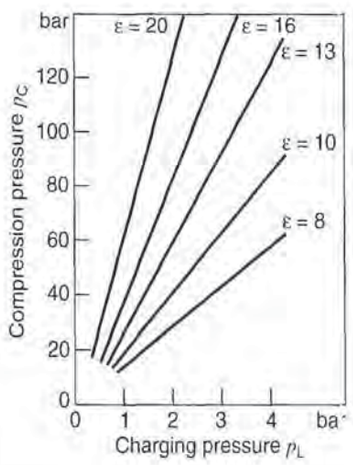
Final compression pressure and temperature

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.



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

PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

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.

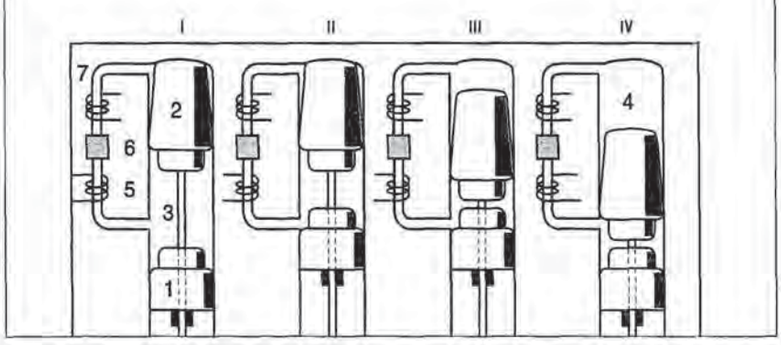
After being heated the gas enters the "hot" area above the displacer. During the transition from phase III to phase IV the hot gas expands; the power piston and displacer are pushed into their lowermost positions and power is produced. The cycle is completed with the transition from phase IV back to 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.

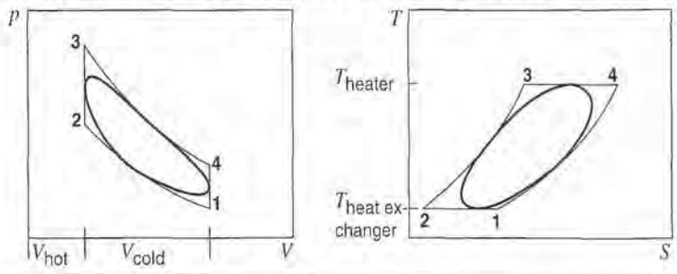
If both pistons are connected to a shaft, i.e., via a rhombic drive, they carry out phase-shifted sinusoidal movements leading to a rounded work diagram with the same cycle efficiency – similar to the efficiency of the Carnot cycle – but with sacrifices in power and net efficiency.

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.



PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

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



Design and operating characteristics

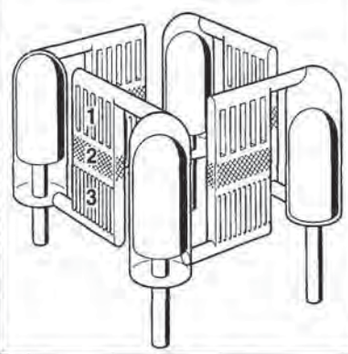
Modern Stirling engines are double-acting engines with (for instance) 4 cylinders operating with a defined phase shift. Each cylinder has only one piston whose top surface acts as a power piston, and whose bottom surface acts as a displacer for the following cylinder. The heat exchanger, regenerator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

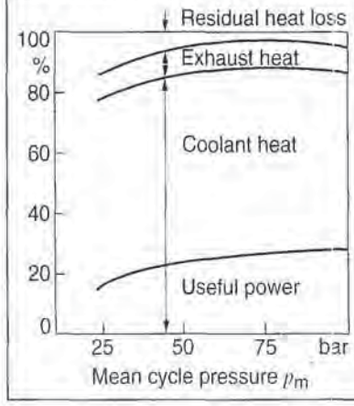
Advantages of the Stirling engine: very low concentrations of all the pollutants which are subject to legislation (HC, CO and NO_x); quiet operation without combustion noise; burns a wide variety of different fuels (multifuel capability); fuel consumption (in program map) roughly equivalent to that of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



PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

Wankel rotary engine

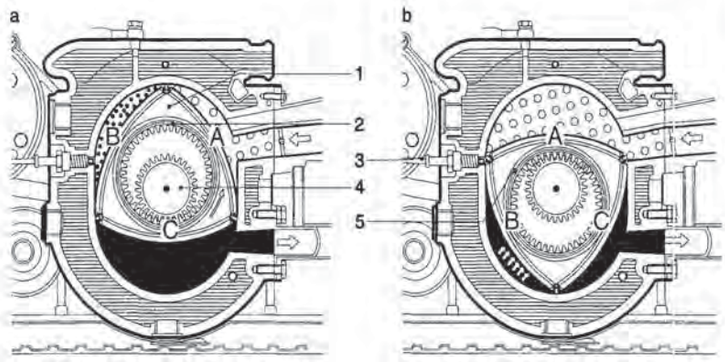
The rotary engine is an unconventional piston powerplant in which the crankshaft mechanism is replaced by an eccentric drive unit operated by a rotary piston. The piston forms the combustion chambers as it proceeds through its trochoidal rotation pattern.

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.*



PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

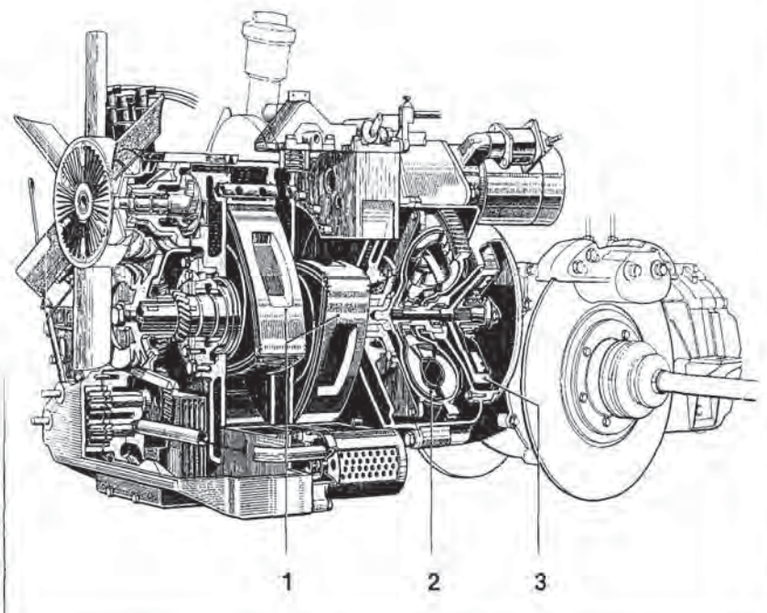
Gas exchange is regulated by the piston itself as it moves past slots in the housing. An alternative to this arrangement – with peripheral intake ports located along the apices' trochoidal path – is represented by intake ports in the side of the block (side ports).

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.



PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

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.

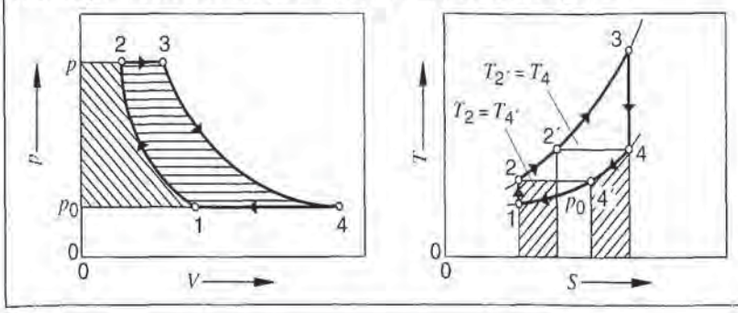
The comparative thermodynamic cycle for the gas turbine is the constant-pressure or Joule cycle. It consists of isentropic compression (process 1 → 2), isobaric heat addition (process 2 → 3), isentropic expansion (process 3 → 4) and isobaric heat dissipation (process 4 →1). High levels of thermal efficiency 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

PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

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

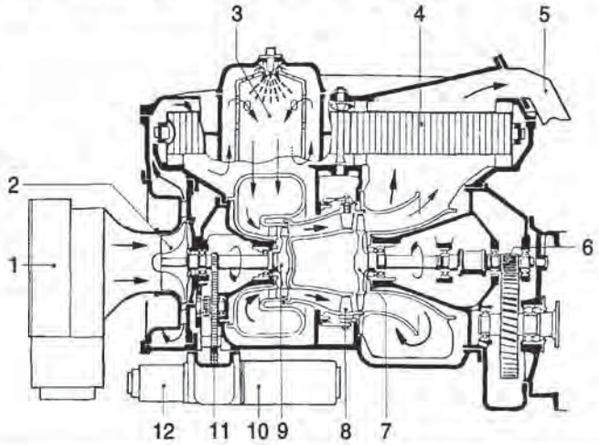
$$\eta_{th} = 1 - (T_2/T_3).$$

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.



PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

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

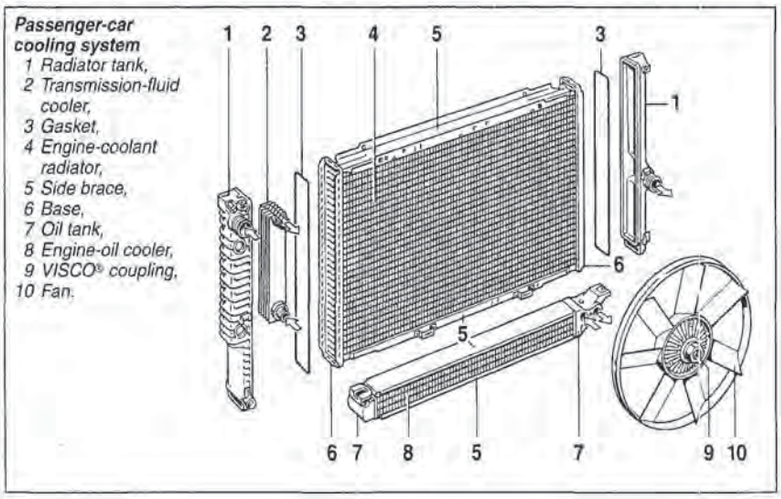
The cores of the coolant radiators in modern passenger cars are almost always made of aluminum, which is also being used in an increasing number of heavy-vehicle radiators throughout the world. There are two basic assembly variations: brazed and mechanically joined radiators.

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.

PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

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.

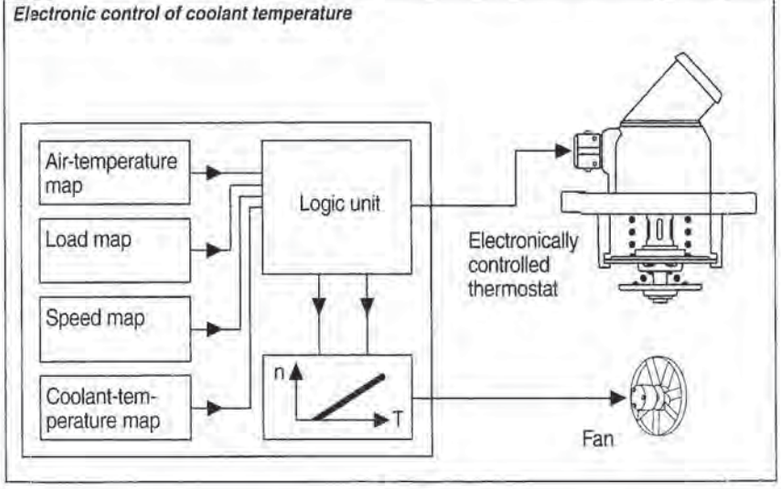
Aside from reliable cooling, other priorities in radiator design include minimizing the power required to operate the fan and maintaining low aerodynamic drag. The mass of the cooling air stream is a decisive factor, as there is an inverse relationship between fan and radiator capacities: A more powerful 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



PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

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

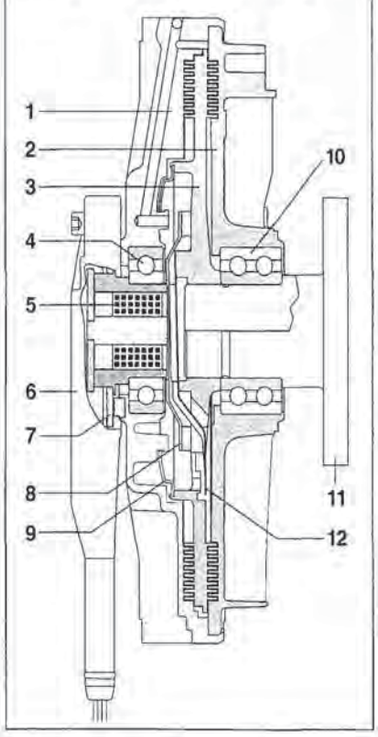
Coolant expansion tank

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

PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

(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

PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

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.

PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

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

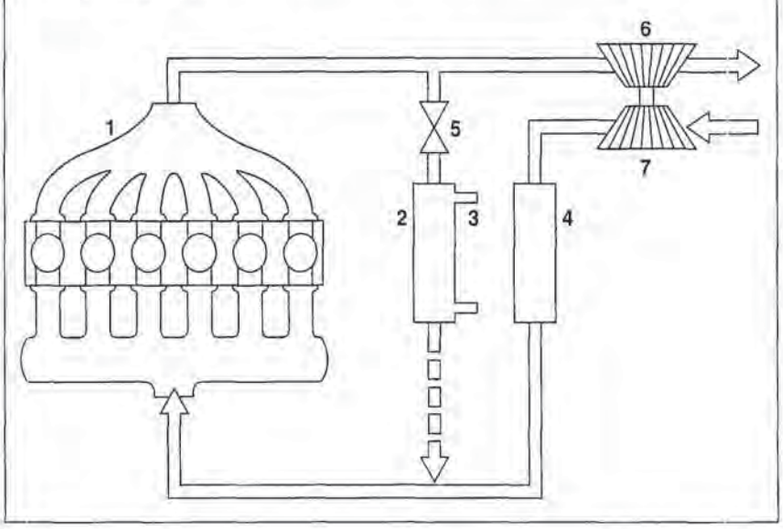
Because of the introduction of new, stricter exhaust-emission regulations for diesel engines, new technologies for reducing emissions have become the focus of engineers' attention.

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.



PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

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

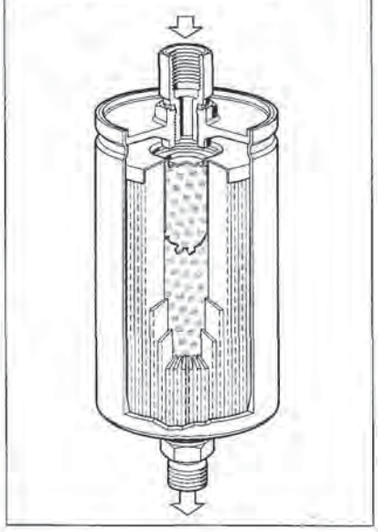
Fuel filters for spark-ignition engines are located between the fuel tank and the fuel pump and/or on the pressure side downstream of the fuel pump. The preferred design is the in-line filter. In addition, easy-change filters screwed to a base, and housing filters with non-metal filter elements are used.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



PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

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

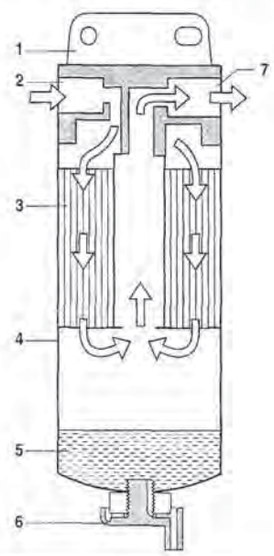
As well as high levels of superfine particulate separation, the fuel filter is expected to feature a particulate-retention capability which will permit compliance with the maintenance intervals as stipulated by the vehicle manufacturer. This can only be achieved by using special filter mediums, e.g. 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.

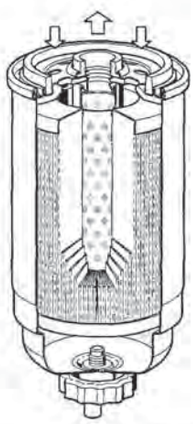
Diesel fuel is more heavily contaminated than gasoline (particularly due to products which result from the aging process). Diesel filters are therefore designed as easy-change filters. Screw-on easy-change filters with radial or spiral vee-shaped filter elements are widely used. Fuel filters with aluminum/plastic 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



PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

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.

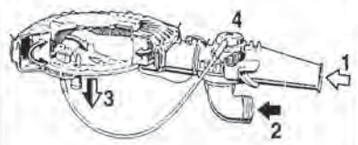
Passenger-car air filters

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.

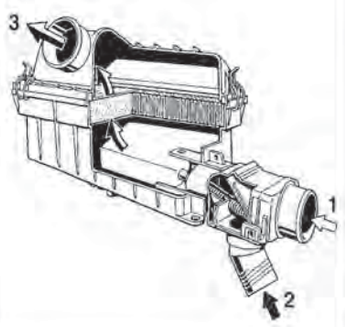
The required amount of hot air is drawn in in the vicinity of the exhaust and added to the cold intake air at the filter inlet by means of a flap-valve mechanism. The regulating mechanism is usually an automatic arrangement employing either a pneumatic vacuum unit connected to the intake manifold or 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

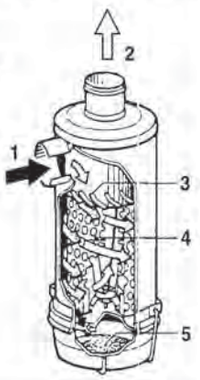
Central passenger-car air filter
1 Fresh-air intake, 2 Warm-air intake
3 Outlet for warm/fresh air mixture,
4 Vacuum unit.



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.



PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

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

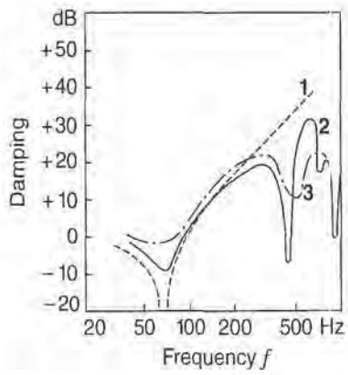
$$f_0 = \frac{c}{2\pi} \cdot \sqrt{\frac{A_m}{l \cdot V}}$$

Where

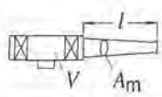
c Speed of sound in air
l Length of the intake manifold
A_m Mean cross section of the intake manifold
V Filter volume

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.



PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

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).

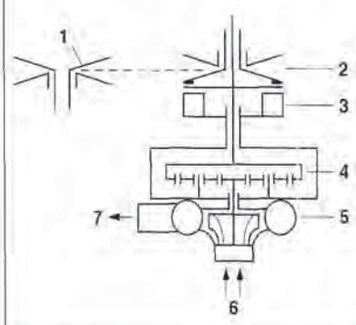
Although the pressure-wave supercharger also derives its compression force from the exhaust gases, it requires a supplementary mechanical drive (combination of mechanical and fluid coupling).

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.

PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

- 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.

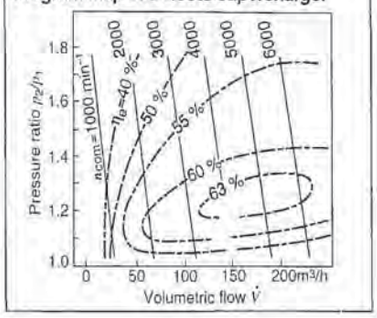
The two twin-bladed rotary pistons of the Roots supercharger operate without directly contacting each other or the housing. The size of the sealing gap thus created is determined by the design, the choice of materials and the manufacturing tolerances. An external gear set synchronizes the motion of the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

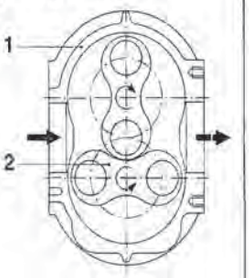
The spiral-type supercharger employs an eccentrically-mounted displacement element which is designed to respond to rotation of the input shaft by turning in a double-eccentric oscillating pattern. In sequence, the working chambers open for charging, close for transport and open once again for discharge 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.

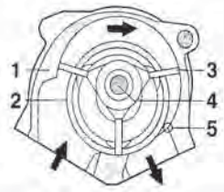
Program map of a Roots supercharger



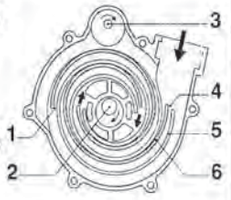
Cross-section through a Roots supercharger
1 Housing, 2 Rotary piston.



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.



PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

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.

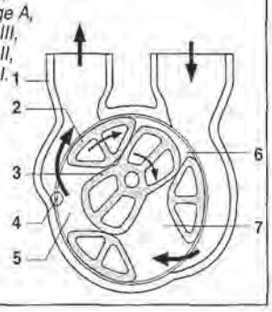
Exhaust-gas turbochargers

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.

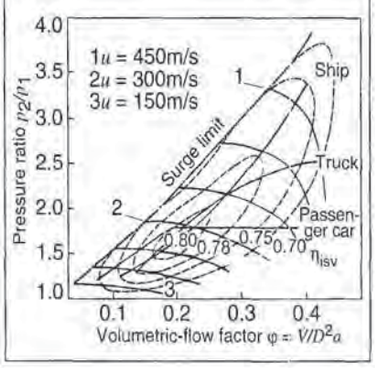
Exhaust-gas turbochargers are used on engines in passenger cars, trucks and heavy-duty engines (marine and locomotive power plants, stationary power generators).

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



PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

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.

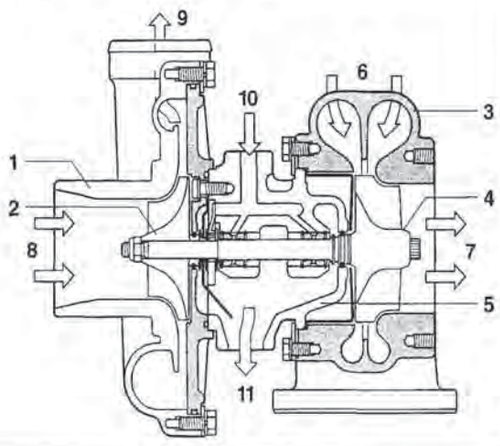
Piston rings are installed on both the exhaust and intake sides to seal off the bearing housing's oil chamber. In some special applications sealing is enhanced by trapped air or a compressor-side carbon axial face seal. Friction bearings are generally used, installed radially as either floating double 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.

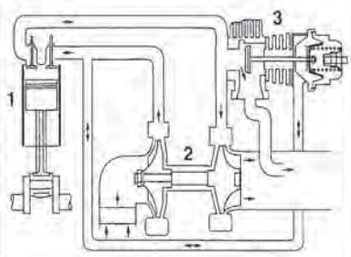
In contrast, in the case of constant-pressure turbocharging, only the pressure energy of the exhaust-gas is utilized, and single-flow turbine housings can be employed. This configuration has become especially popular for use in conjunction with water-cooled turbine housings on marine engines. The exhaust-gas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

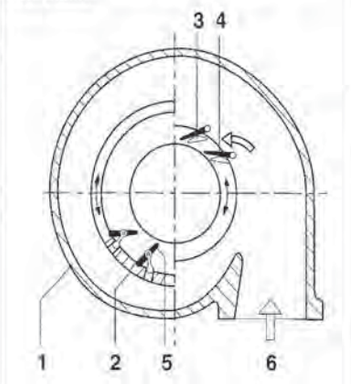


PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

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).

PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

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

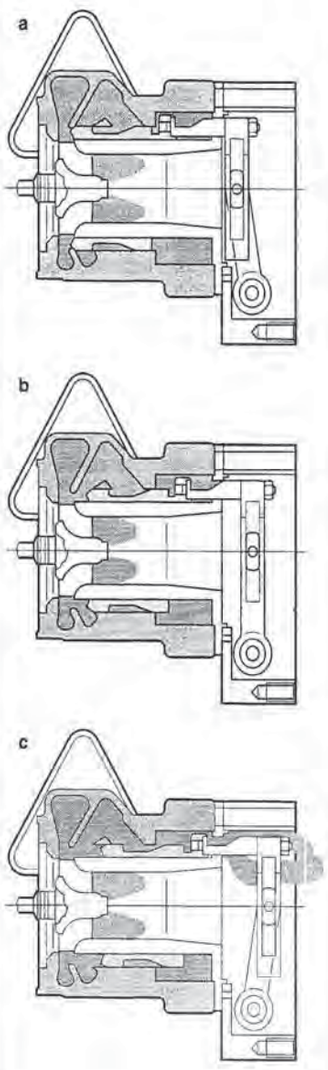
Because of the expensive charger switching equipment, sequential supercharging is predominantly used in ship propulsion systems or generator drives. In this case, as engine load and speed increases, one or more turbochargers are cut in to the basic supercharging process. Thus, in comparison with a large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



PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

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.

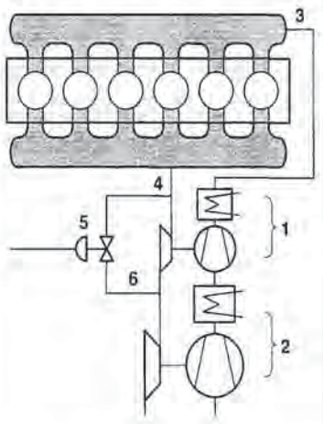
Two-stage controlled supercharging thus enables infinitely variable adaptation on the turbine and compressor sides to the requirements of engine operation.

Pressure-wave superchargers

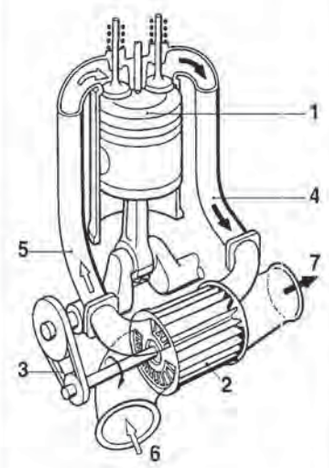
The pressure-wave supercharger exploits the dynamic properties of gases, using pressure waves to convey energy from the exhaust-gas to the intake air. The energy exchange takes place within the cells of the rotor (known as the cell rotor or cell wheel), which also depends upon an engine-driven belt for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



PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

conditions and provide a good overall boost curve.

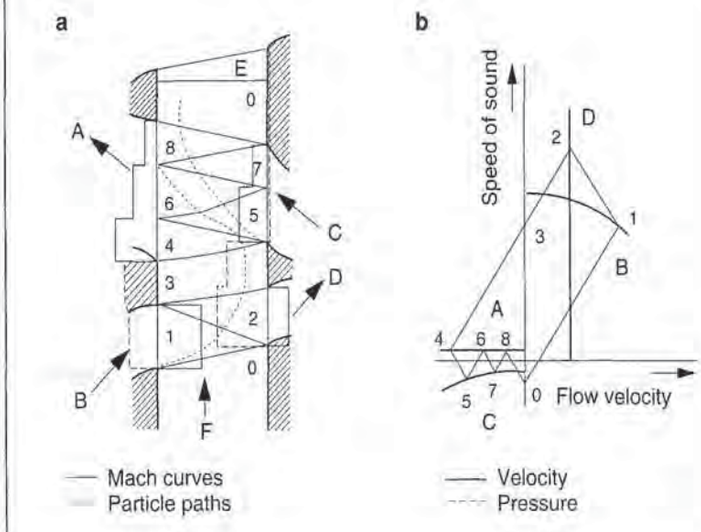
The exchange of energy occurring within the rotor at the speed of sound ensures that the pressure-wave supercharger responds rapidly to changes in engine demand, with the actual reaction times being determined by the charging processes in the air and exhaust tracts.

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.



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.

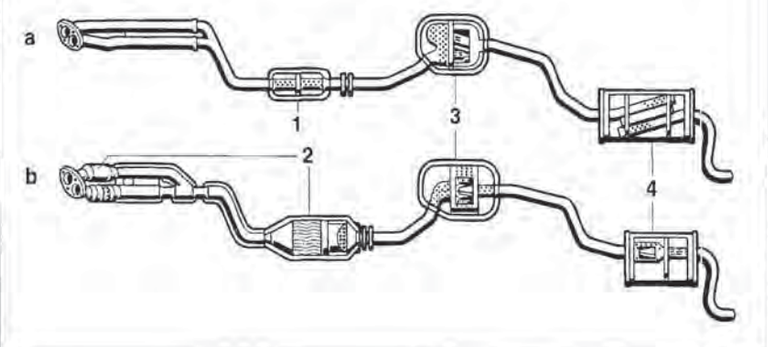
Depending on the size of the vehicle and the engine, one or several mufflers are used. In V-engines the left and right cylinder banks are frequently run separately, each being fitted with its own catalytic converter or muffler, and only brought together at the end of the vehicle in one large muffler.

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.



PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

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.

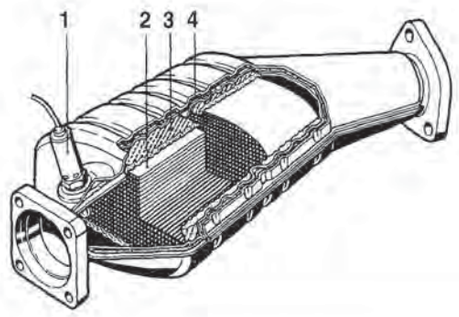
The swelling-mat mounting has come to the forefront in application. This mounting is made of ceramic fiber felt, composed of aluminum silicate fibers and expanding mica particles. The two substances are combined using acrylic latex. Under the influence of temperature the matting expands and presses the 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.



PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

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.

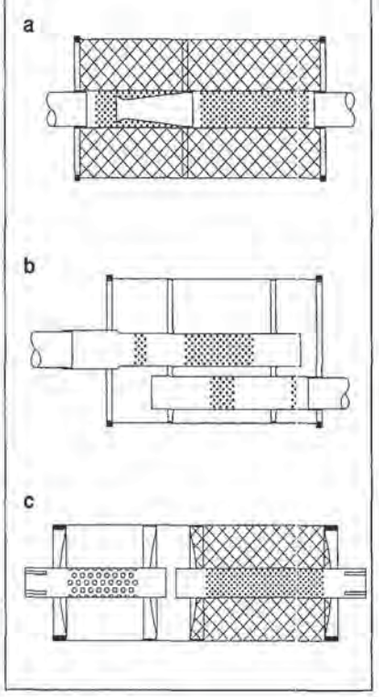
Reflection mufflers consist of chambers of varying lengths which are connected together by pipes. The differences in the cross-sections of the pipes and the chambers, the diversion of the exhaust gases, and the resonators formed by the connecting pipes with the chambers, produce muffling which is particularly 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).



PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

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.

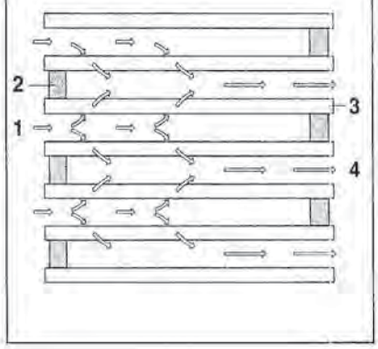
The ceramic-monolith filter currently represents the best compromise with regard to the requirements made of the filter. In contrast to the flow-through catalytic-converter monoliths, the channels for the soot filter are alternatingly opened and closed, so that the particle-laden exhaust gas is forced 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.



PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

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.

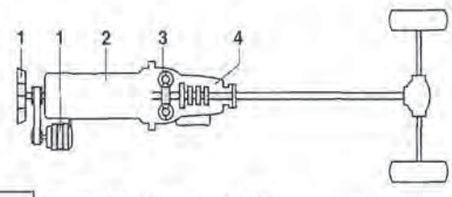
A/F mixture formation

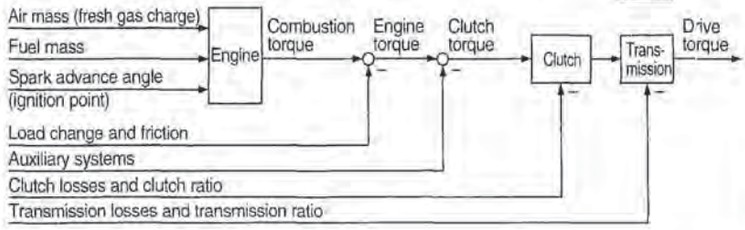
In the "mixture formation" subsystem, the associated fuel mass is calculated and from this the necessary injection time and the optimum injection point are determined.

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.





PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

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.

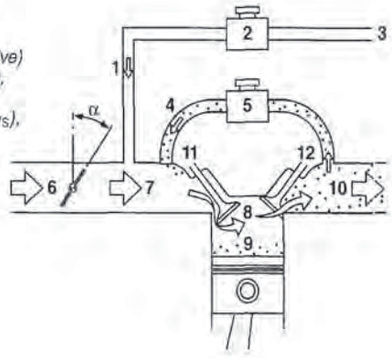
The residual exhaust-gas share is determined by the charge cycle.

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.



PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

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

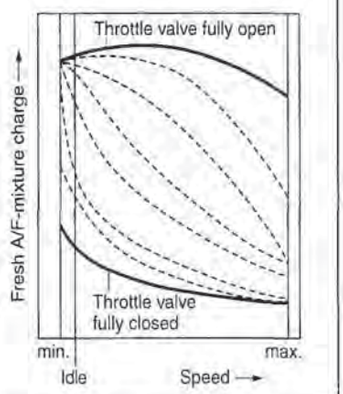
The charge cycle of fresh A/F mixture and residual exhaust gas is controlled by the appropriate opening and closing of the intake and exhaust valves. The cams on the camshaft determine the points at which the valves open and close (valve timing) and the course of the valve lift. This influences the charge-cycle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

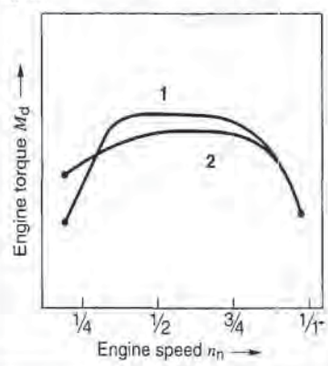
Supercharging

The obtainable torque is proportional to the charge of fresh A/F mixture. It is 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.



PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

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

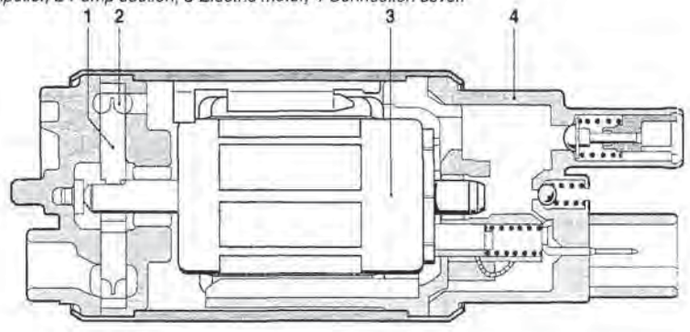
The electric fuel pump consists of:

- 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.



PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

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.

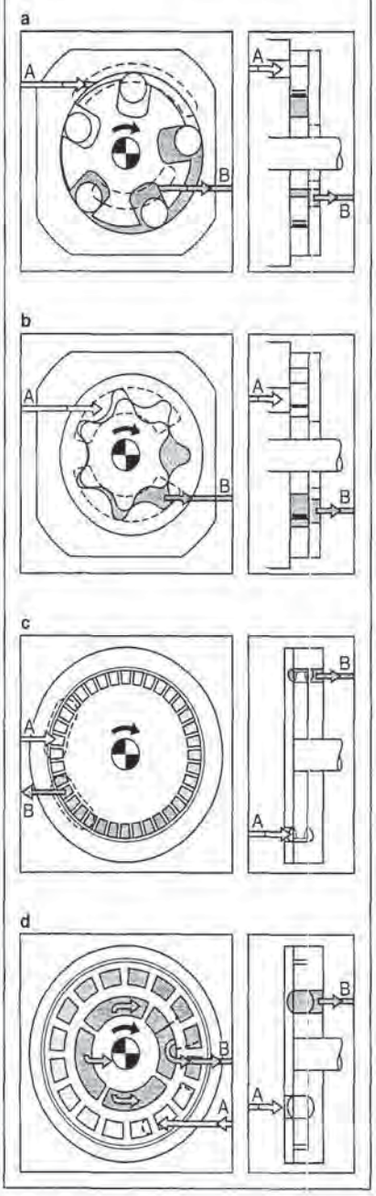
Flow-type pumps

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.



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.

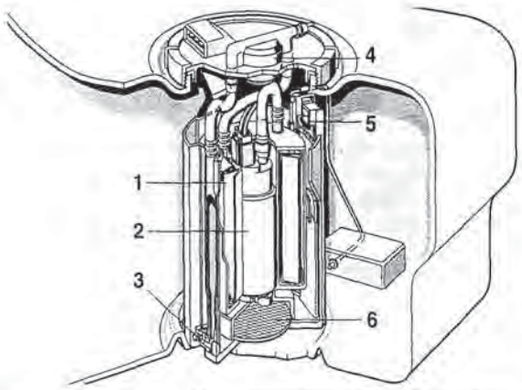
The efficiency ranges between 10 and approx. 20 %. The fuel systems of newly designed vehicles with spark-ignition engines rely almost exclusively on flow-type pumps for fuel delivery.

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.



PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

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):

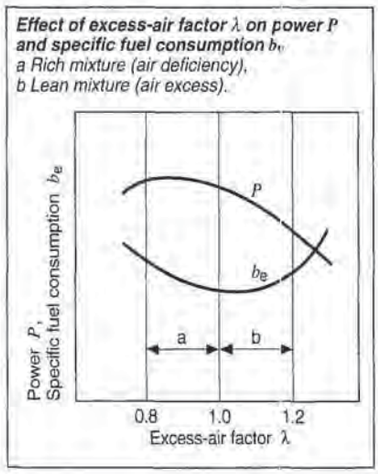
λ = Ratio of supplied air mass to air requirement with stoichiometric combustion.

$\lambda = 1$: 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).

PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

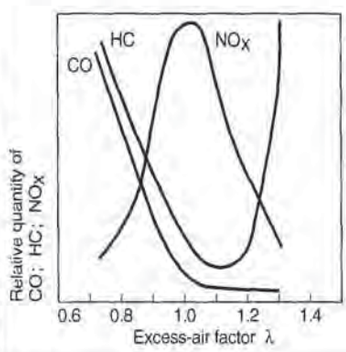
The graphs show the dependence of power output, specific fuel consumption and pollutant buildup on the excess-air factor for a typical engine with manifold injection. It can be deduced from these graphs that there is no ideal excess-air factor at which all the factors assume the most favorable value. 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



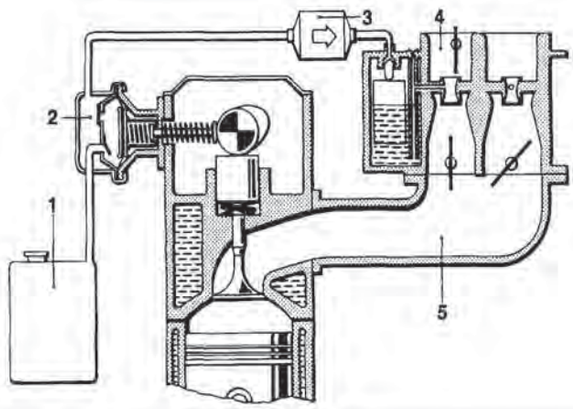
Mixture-formation systems

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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US PATENT NO. 6,581,565

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.

PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

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

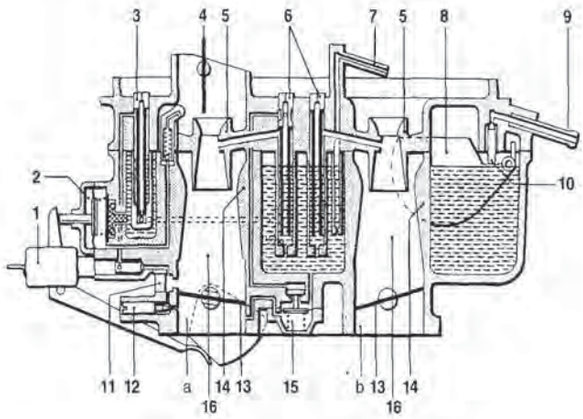
Main system
The fuel is metered by the main jet. Correction air is added as a delivery aid to the fuel through side orifices in the venturi tube.

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.



PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

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 ir coming 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, includir g regulation of the servo elements that operate the choke valve and main throttle valve.

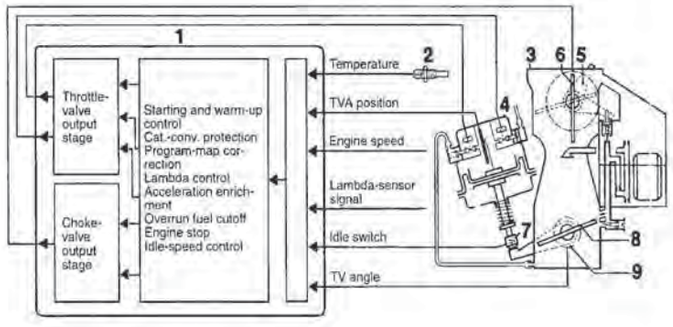
Basic functions

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.



PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

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).

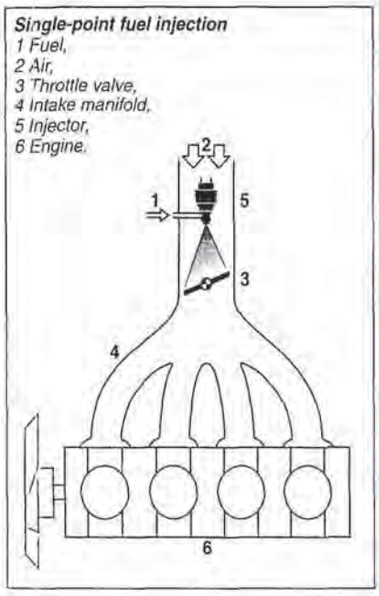
Single-point injection (SPI)
Single-point injection is an electronically controlled injection system in which an electromagnetic fuel injector injects the fuel intermittently into the intake manifold at a central point ahead of the throttle valve. The Bosch single-point injection systems are called Mono-Jetronic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

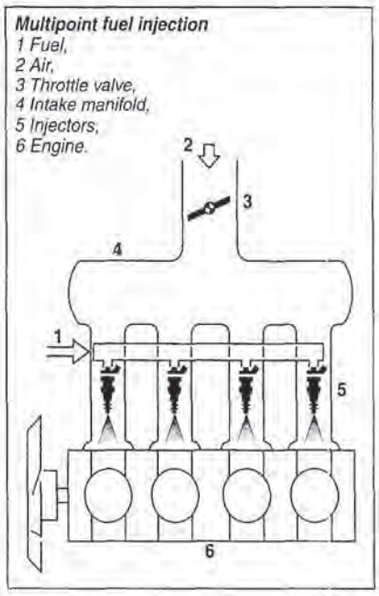
Mechanical injection system:
The K-Jetronic system operates without a drive and injects the fuel continuously. The injected fuel mass is not determined by the fuel injector but rather prespecified by the fuel distributor.

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).



Single-point fuel injection
1 Fuel,
2 Air,
3 Throttle valve,
4 Intake manifold,
5 Injector,
6 Engine.



Multipoint fuel injection
1 Fuel,
2 Air,
3 Throttle valve,
4 Intake manifold,
5 Injectors,
6 Engine.

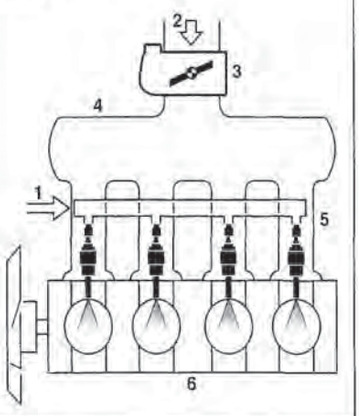
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US PATENT NO. 6,581,565

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.

PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

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).

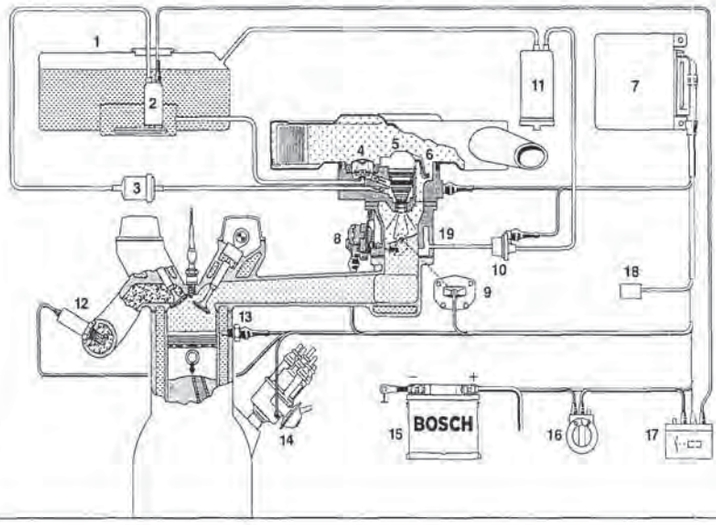
Mono-Jetronic

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.



PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

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.

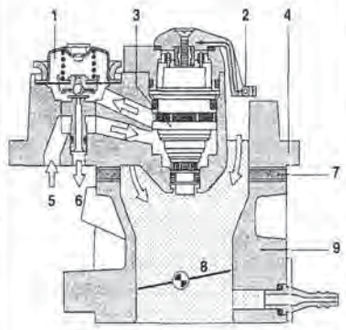
System control

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.

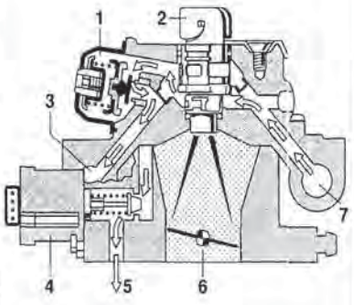
Adaptation functions

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.



PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

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.

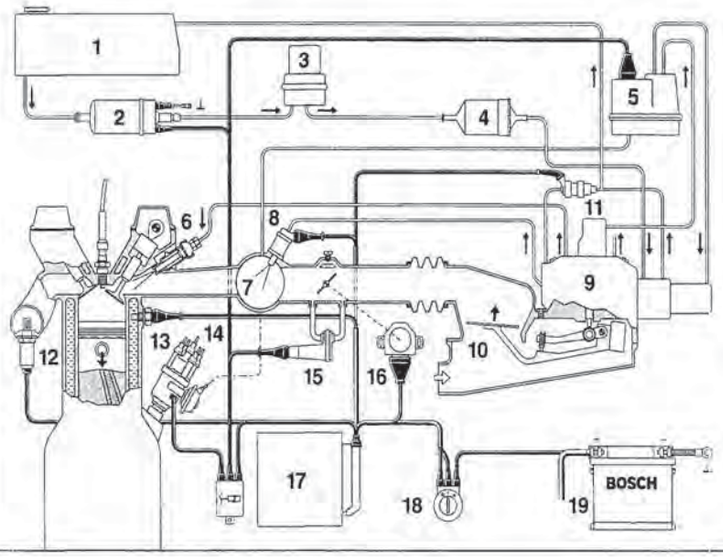
Operation
The intake air flows through the air filter, the air-flow sensor, and the throttle valve, before entering the intake manifold and continuing to the individual cylinders.

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.



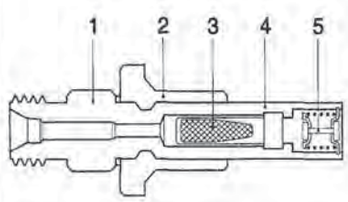
PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

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.

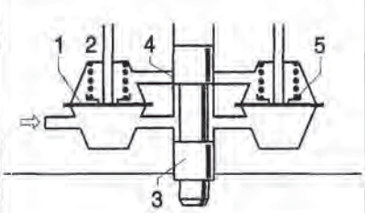
Fuel distributor

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

PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

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

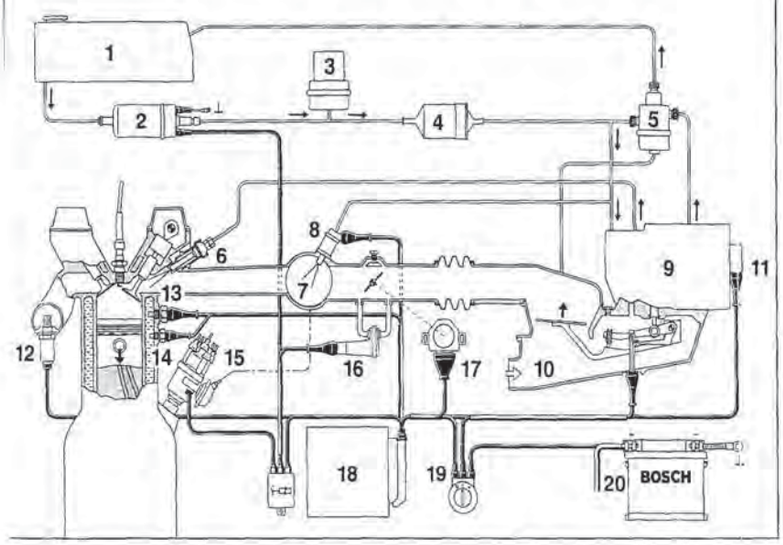
KE-Jetronic is an advanced version of the K-Jetronic system. KE-Jetronic includes an ECU for increased flexibility and supplementary functions. Additional components include:

- 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.

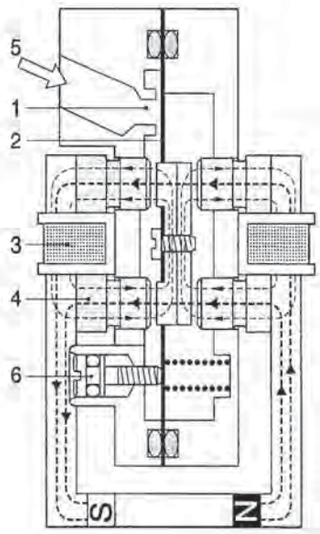


PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

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

PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

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

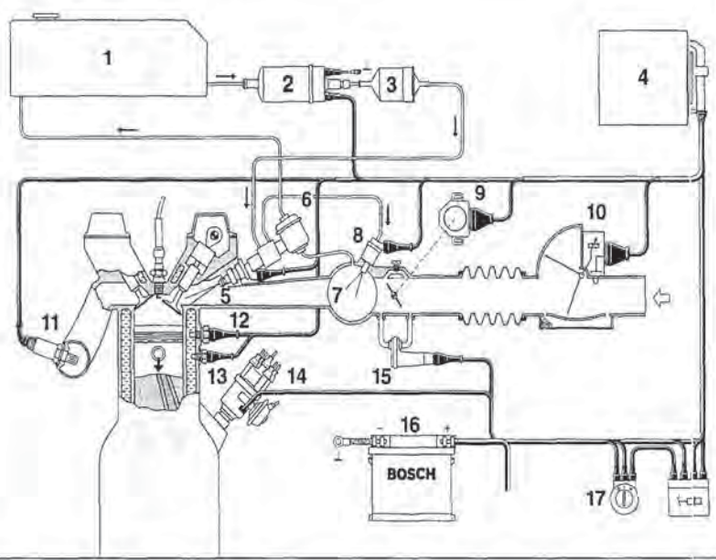
The fuel is injected through the engine's solenoid-operated injectors. A solenoid valve assigned to each cylinder is triggered once per crankshaft revolution. All of the injectors are wired in parallel to reduce the complexity of the electrical circuit. The pressure differential between fuel and intake-manifold 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.



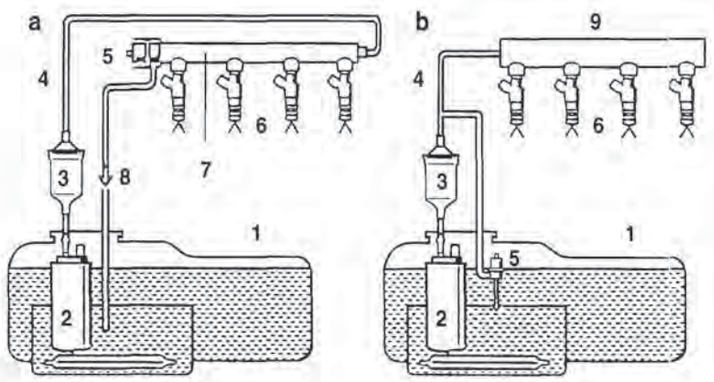
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US PATENT NO. 6,581,565

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.

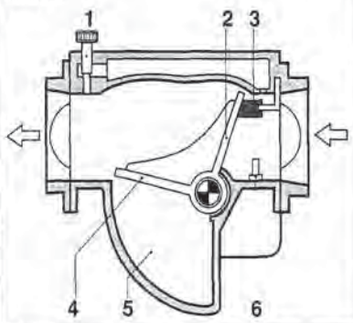
Returnless system:
A returnless fuel-supply system reduces the tendency of the fuel in the fuel tank to heat up, thus making it easier to comply with the legal requirements governing vehicular evaporative emissions. The pressure regulator is located either in the fuel tank or in its immediate vicinity, 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).

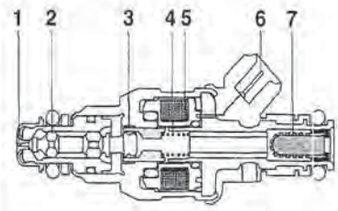


PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

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.

PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

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.

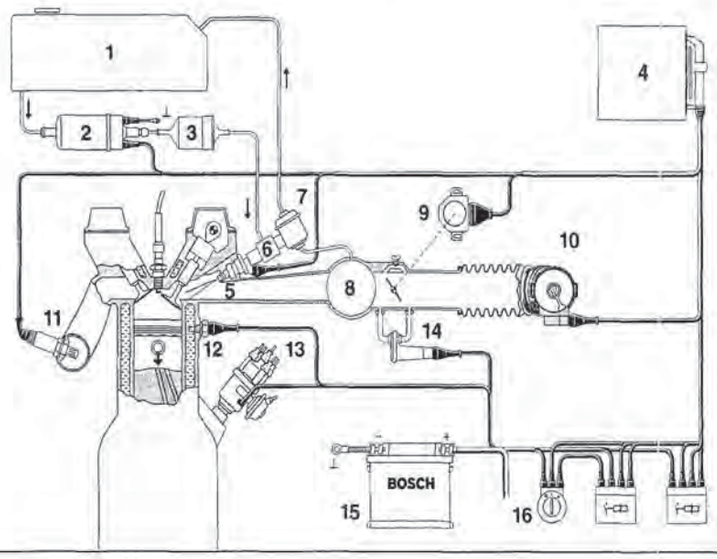
Operating-data processing in the ECU

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.



PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

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.

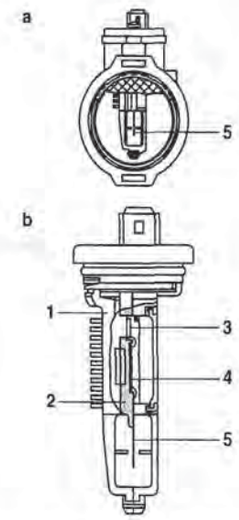
Hot-film air-mass flow meter

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.

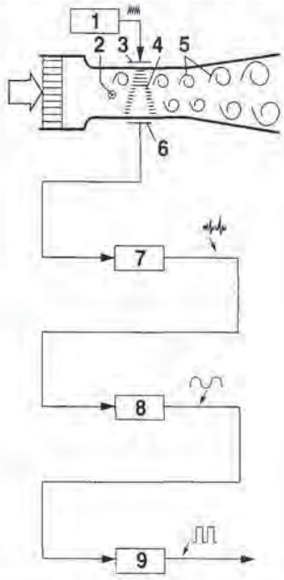
Kármán vortex volumetric flow meter

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.



PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

Electromagnetic fuel injectors

Design and operation

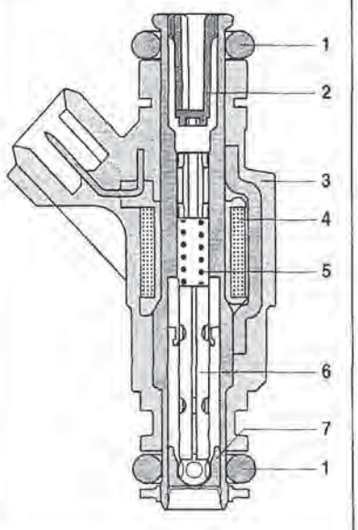
Fuel injectors essentially consist of a valve housing with current coil and electrical connection, a valve seat with spray-orifice disk and a moving valve needle with solenoid armature.

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.

PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

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).

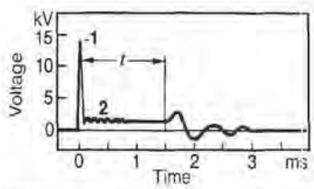
The spark plug determines the location and length of the spark; spark duration depends upon the type and design of the ignition system, as well as on the momentary ignition conditions.

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.

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.

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



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-

PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

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:

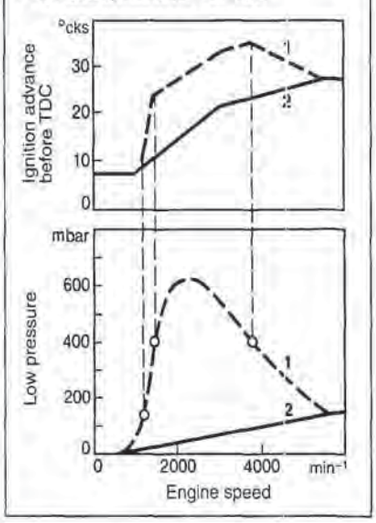
- maximum engine performance
- low fuel consumption
- no engine knock
- clean exhaust gas.

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

Example of cumulative ignition timing consisting of centrifugal and vacuum advance

1 Part-load operation, 2 Full load.



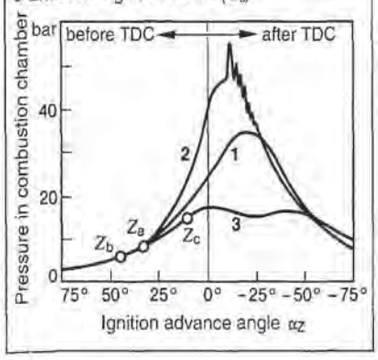
adjustments can be used e.g. for rapid torque adaptation or for swift heating of the catalytic converter.

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

PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

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.

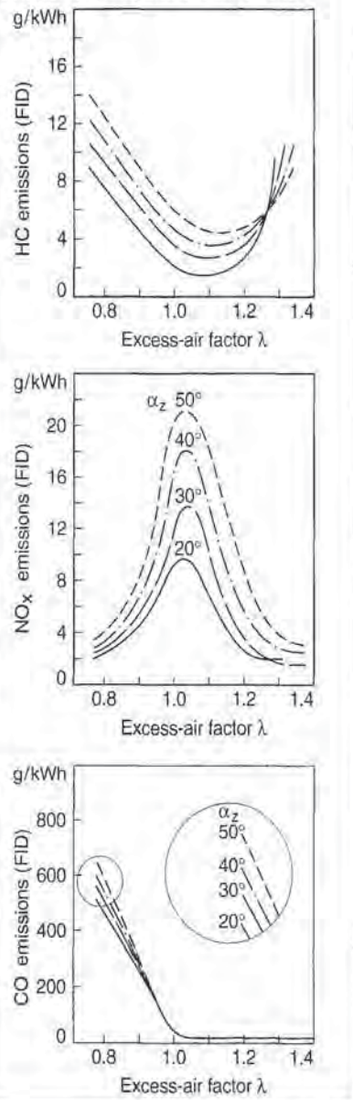
Overview of various ignition systems

Function	Ignition system			
	CI	TI	EI[1])	DLI
Designation	Coil ignition system	Transistorized ignition system	Engine management	
			Electronic ignition system	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

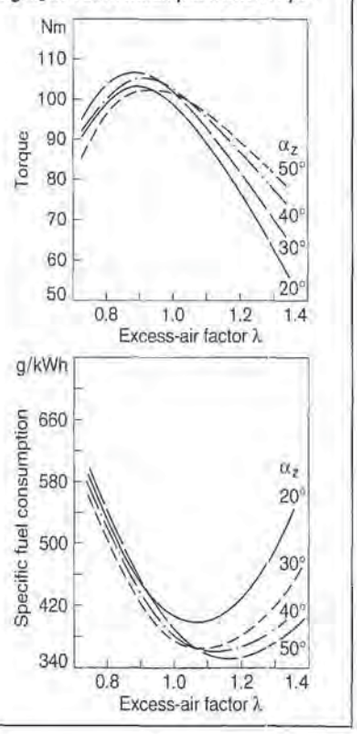
[1]) Any desired ignition timing is not possible with EI. DLI therefore dominates with integrated engine-management systems.

PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

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.

PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

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).

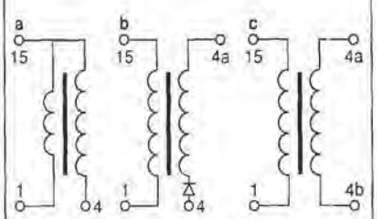
Design and operation

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.

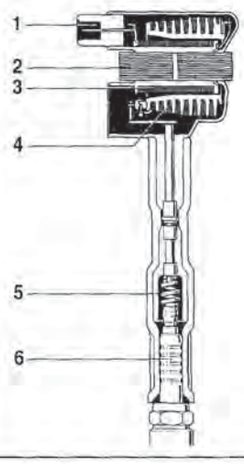
The coil's synthetic materials provide good adhesion between all of the high-voltage components and the molded epoxy resin, which penetrates into all the capillary spaces. Supplementary iron cores are sometimes embedded on the inside of the synthetic molding.

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.



PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

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.

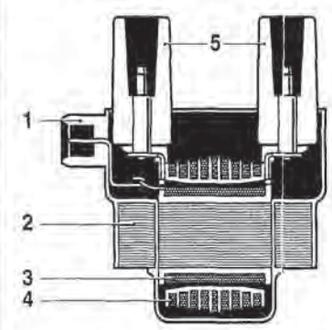
Single and dual-spark ignition coils are an alternative for use in ignition systems with distributorless ignition (DLI).

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

PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

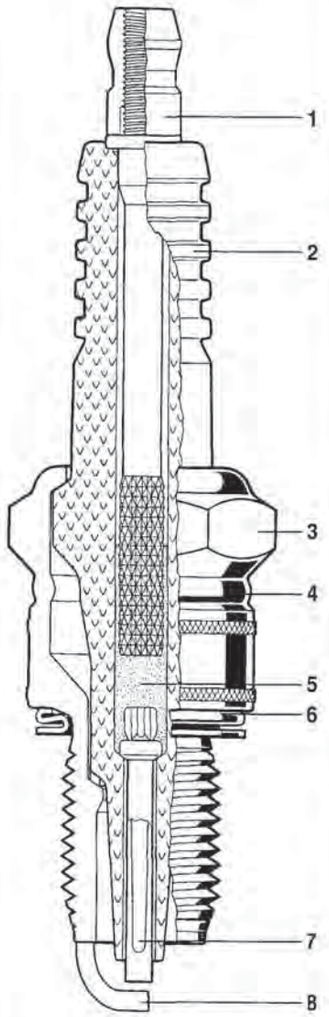
80 bar) within the combustion chamber, its materials must exhibit extreme resistance to thermal loads and continuous vibratory stress. At the same time, that section of the spark plug that protrudes into the combustion chamber is exposed to high-temperature chemical processes, making resistance to aggressive 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.



VW EX1012 (Part 3 of 4)
U.S. Patent No. 6,581,565

PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

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.

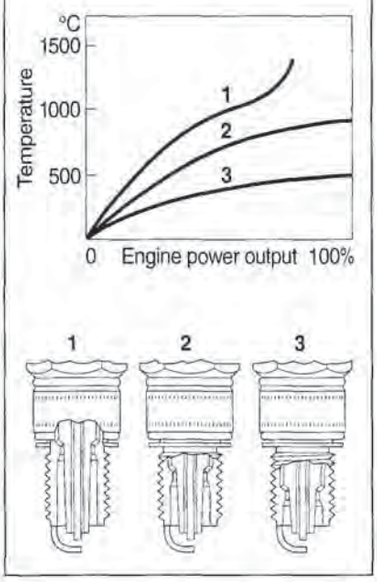
An upper temperature limit of approx. 900 °C should be observed because in this range wear of the spark-plug electrodes increases markedly (due to hot-gas corrosion) and if this limit is significantly exceeded this increases the risk of auto-ignition (ignition of the air-fuel mixture on hot surfaces). 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.

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.

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



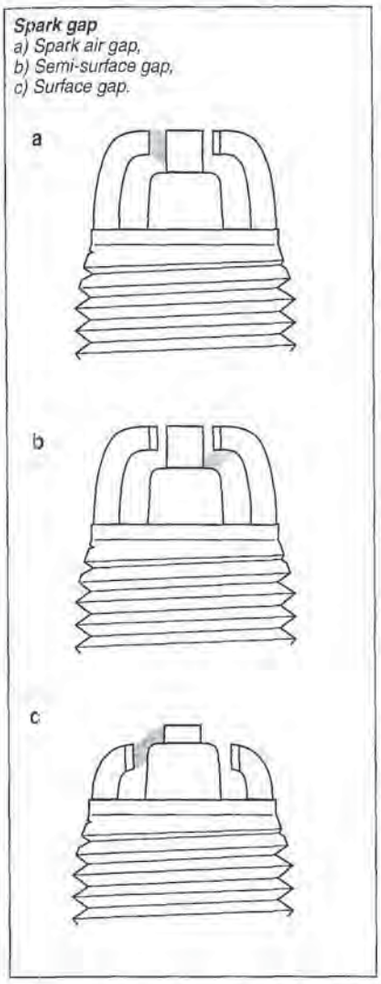
Electrode gap and ignition voltage

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

PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

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.

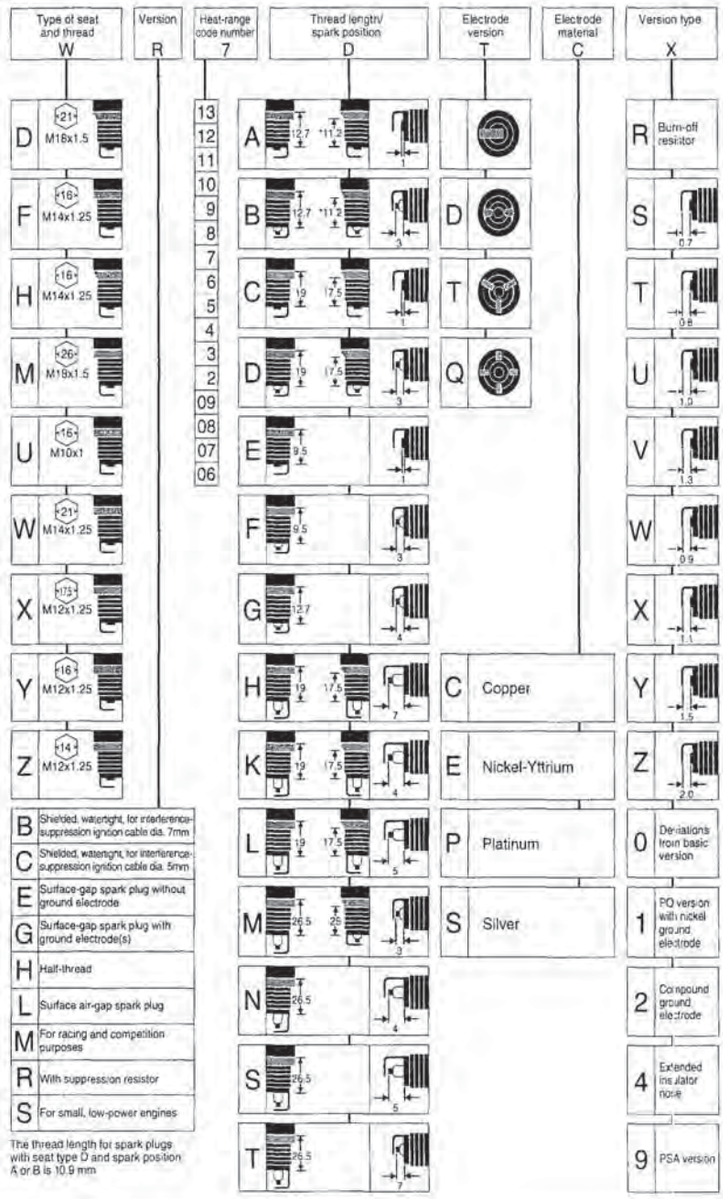
The increased electrode gap improves the ignition properties.

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.

PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

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.



PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

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:

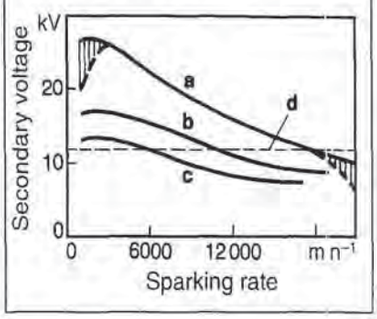
$$f = z \cdot n/2$$

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

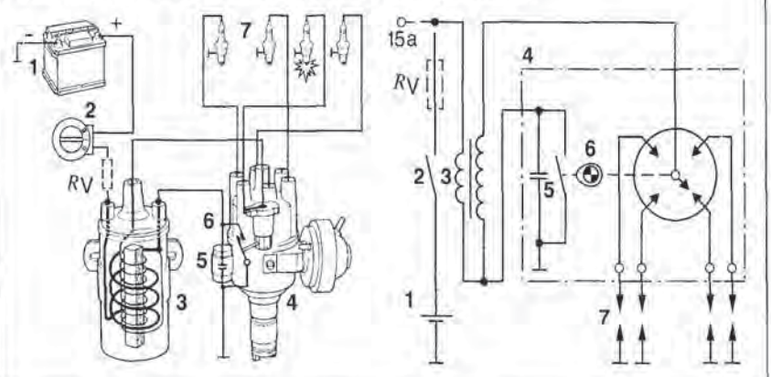
At low engine speeds, the contact-breaker points remain closed long enough to exploit the coil's full energy-storage potential. At higher engine speeds, this contact period – the dwell angle – is shorter, and the primary current is interrupted before maximum energy can be transferred to the coil. The 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).



PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

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.

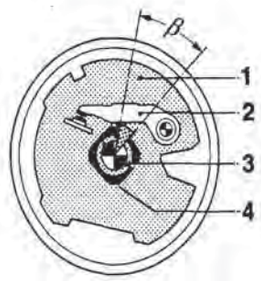
The contact-breaker points and the spark-advance mechanism perform separate functions from those of the distributor proper. They are combined with it in a single unit because they require a synchronized drive.

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.

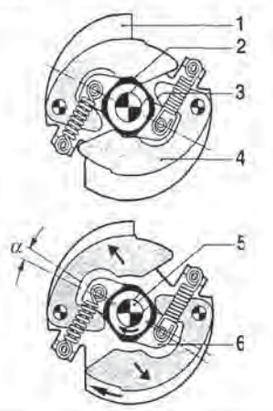
Contact breaker
A cam opens the contact-breaker points to interrupt the flow of primary current to the coil for ignition. The number of cam lobes corresponds to the number of engine cylinders. The portion of the distributor shaft's rotation during which the points remain closed is the dwell angle.

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

PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

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.

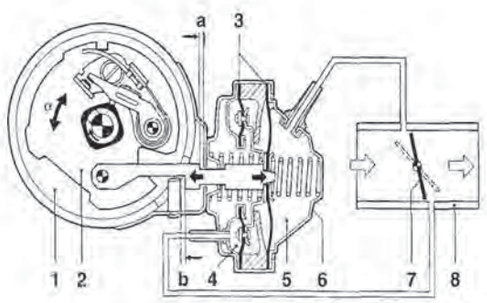
Vacuum adjustment mechanism
The vacuum mechanism adapts the ignition timing to changes in engine output and load factor. Intake manifold vacuum is monitored or tapped off in the vicinity of the throttle valve. The vacuum acts upon two aneroid capsules.

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.



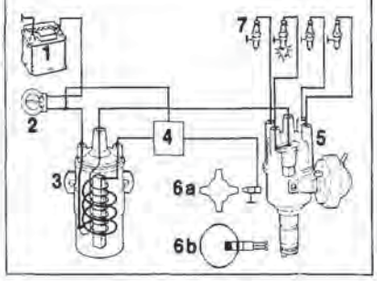
PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

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 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.



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.

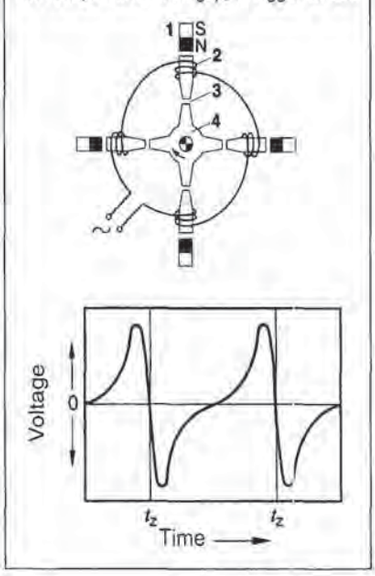
These triggering devices operate according to various principles.

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.

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



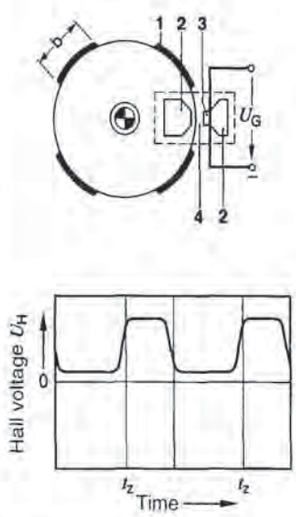
PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

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.

PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

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.

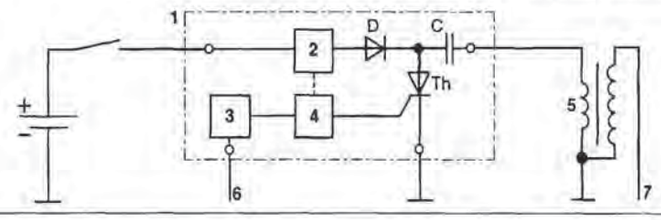
The salient characteristic of the CDI system is that it stores ignition energy in the electrical field of a capacitor. Capacitance and charge voltage of the capacitor determine the amount of energy which is stored. The ignition transformer converts the primary voltage discharged from the capacitor to 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.

PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

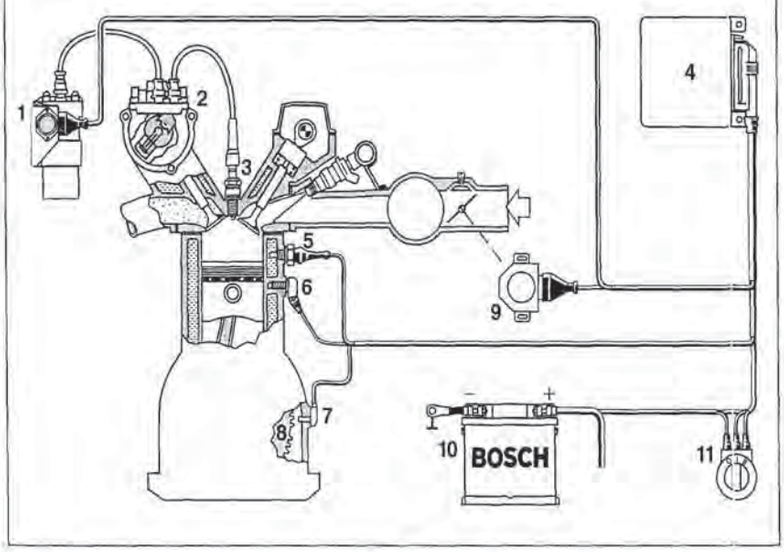
Electronic ignition (EI and DLI)

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.

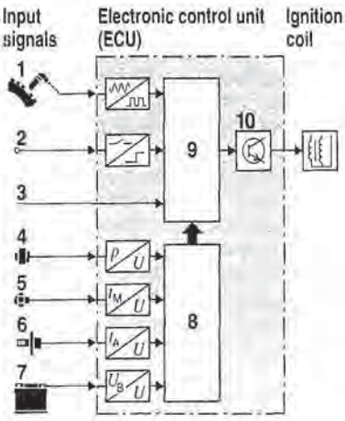
Electronic ignition can be combined with electronic fuel-injection (Motronic), knock control, ASR, etc., making it possible to employ sensors and/or signals from other units in more than one system. A serial bus (CAN, P. 884) further reduces the number of inputs and processing circuits on the ECU's input-side.

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.

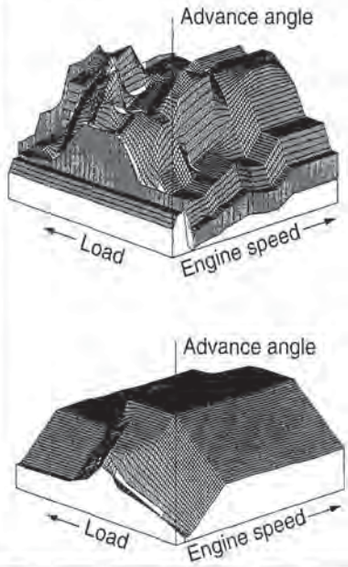


PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

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

PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

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 require any adjustments during the engine's service life.

Rotating voltage distribution

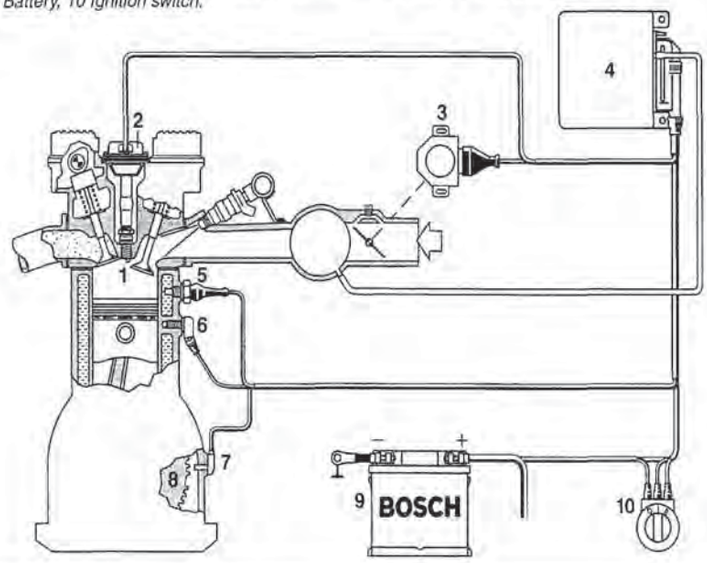
A high-voltage distributor distributes the ignition pulses to the spark plugs of the individual cylinders (as described earlier). If the distributor's adjustment range is insufficient for handling a larger number of cylinders, then two ignition circuits are employed, i.e., two 4-cylinder distributors 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.



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

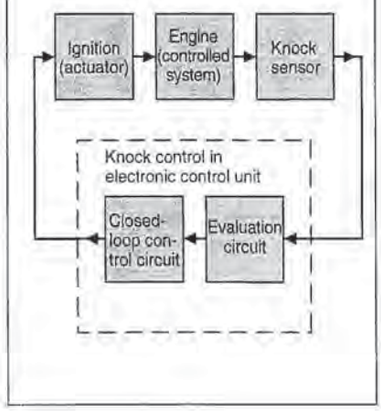
Application

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.

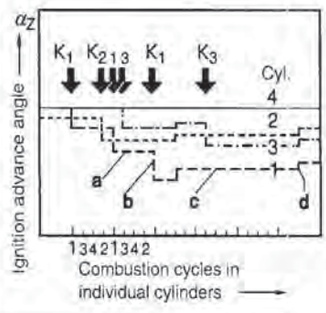
Nevertheless, without knock control a clear safety distance to the knock limit is required. This distance is necessary to ensure that even in the most knock-sensitive case with regard to engine tolerances, engine aging, environmental conditions and fuel quality, no cylinder reaches or exceeds the knock 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.



PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

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.

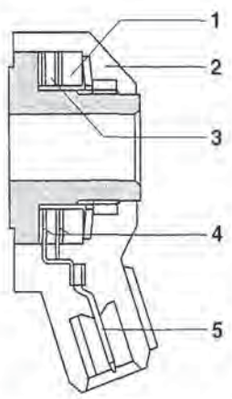
It is essential to work out carefully the number and installation positions of the required knock sensors. Reliable knock detection must be guaranteed for all cylinders and for all engine operating points, particularly at high engine speeds and loads. As a rule, 4-cylinder in-line engines are equipped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

PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

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.

PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

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.

PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

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.

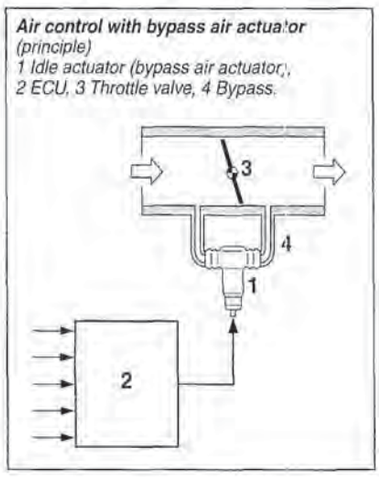
Vehicle management
The ME-Motronic can communicate via the CAN bus system (P. 884) with the ECUs of other vehicle systems. Among other things, in conjunction with the automatic-transmission ECU, it reduces torque at the gear-shift point so that the transmission is subjected to less loading and wear.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

PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

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.

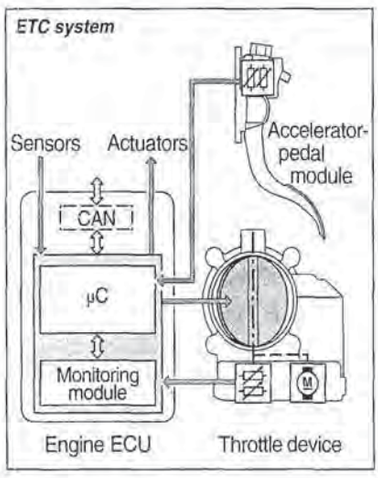
Charge-cycle control

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).



Operating-data acquisition

Vehicle-operator command

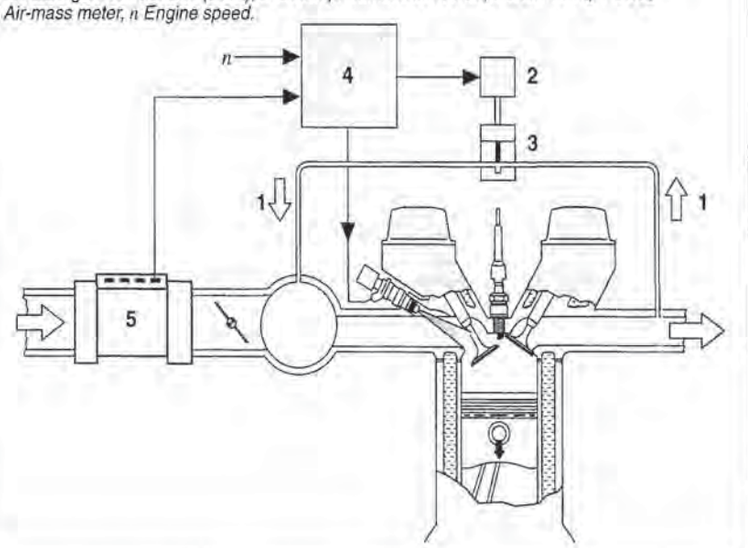
In the case of engine management with an electronic throttle valve (ETC), there is no longer a mechanical connection between the accelerator pedal and the throttle-valve actuator. Instead, the position of the accelerator pedal is sensed by a pedal-travel sensor or accelerator-pedal module and converted 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

PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

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-

PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

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

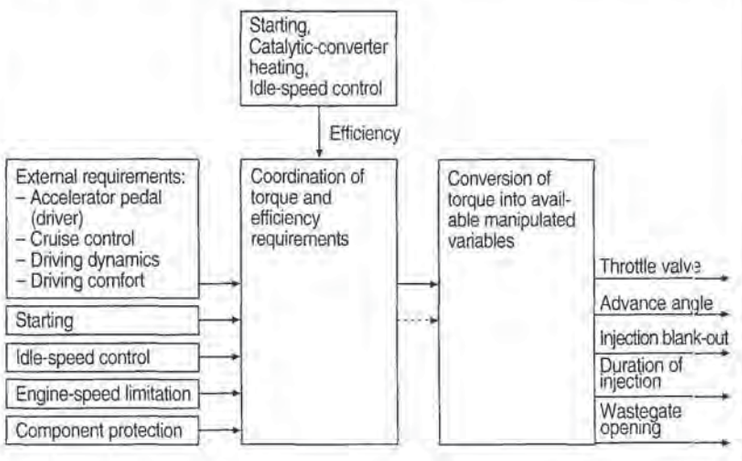
For setting the internal torque, the ME-Motronic's torque converter has a slow path (cylinder-charge path, steady-state operation) for the throttle valve (ETC), and a high-speed path (ignition-angle path, dynamic operation) for variation of the ignition angle and/or injection blank-out of individual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



PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

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-

PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

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

PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

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.

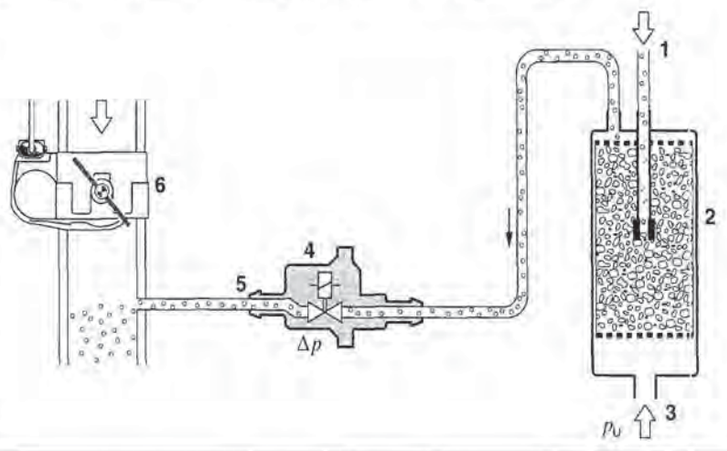
The comfort and convenience functions include:
- Load-change damping,
- Surge-damping function,
- Cruise control.

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.



PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

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

PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

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:

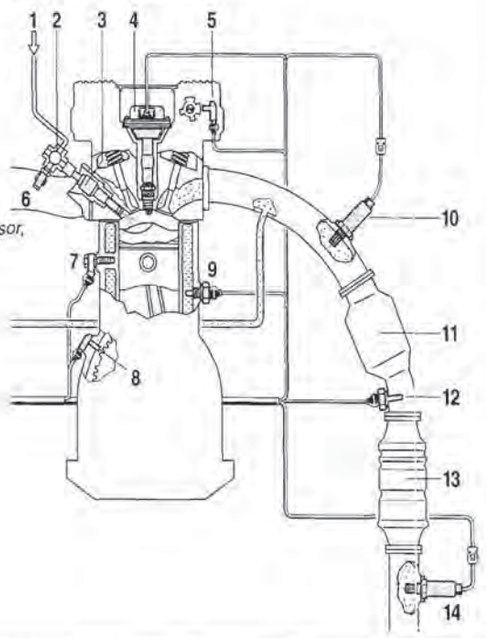
A primary pressure of 0.35 MPa (3.5 bar) is initially generated in a low-pressure circuit on the tank side which comprises an electric fuel pump with a mechanical pressure regulator connected in parallel. This feeds the high-pressure pump driven by the IC engine.

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).



PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

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.

PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

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

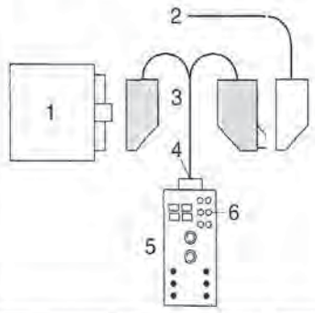
Testing of engine-related electronic systems is carried out using test equipment specifically designed to utilize on-board engine monitoring technology.

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.

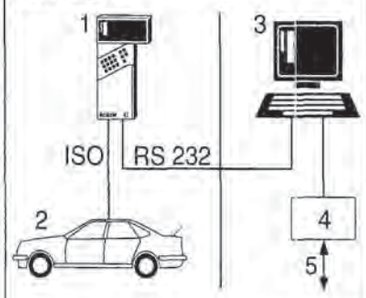
Once connected, the test unit provides a display of both measured values and of signals such as ignition and fuel-injection pulses.

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.



PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

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.

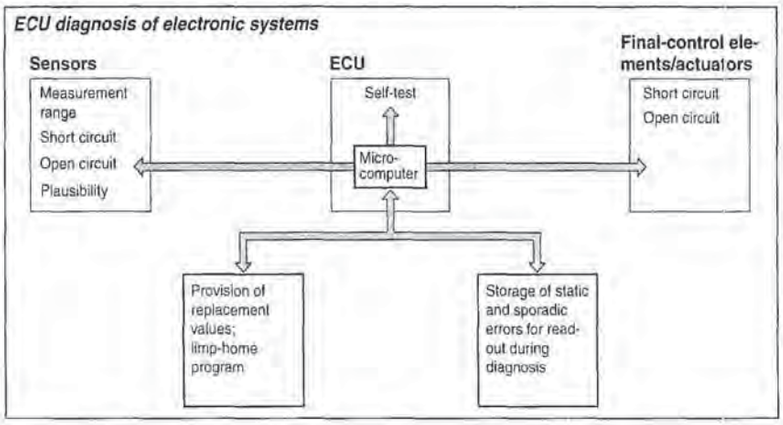
The test unit monitors the period between pulse flanks to determine the sender's baud rate, which it then adopts automatically. The key bytes that follow (assigned by the DIN Motor-Vehicle Committee) specify the protocol for subsequent data communications.

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



PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

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

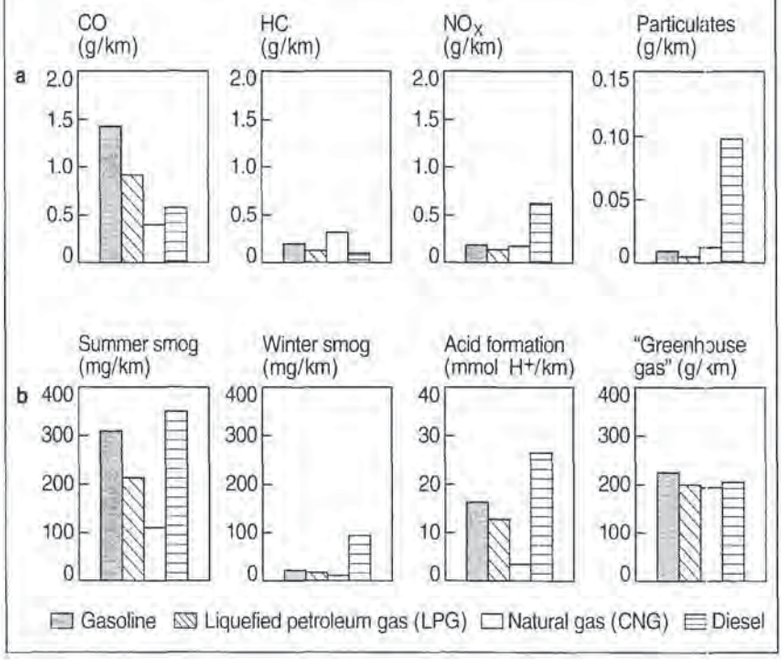
Reserves of natural gas are extensive. Together with it being used nowhere near as much as crude oil, this fact makes natural gas a very interesting alternative fuel for automotive applications. Both the LPG equipment configurations and the emissions produced would be similar to those for the combination 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).



PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

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

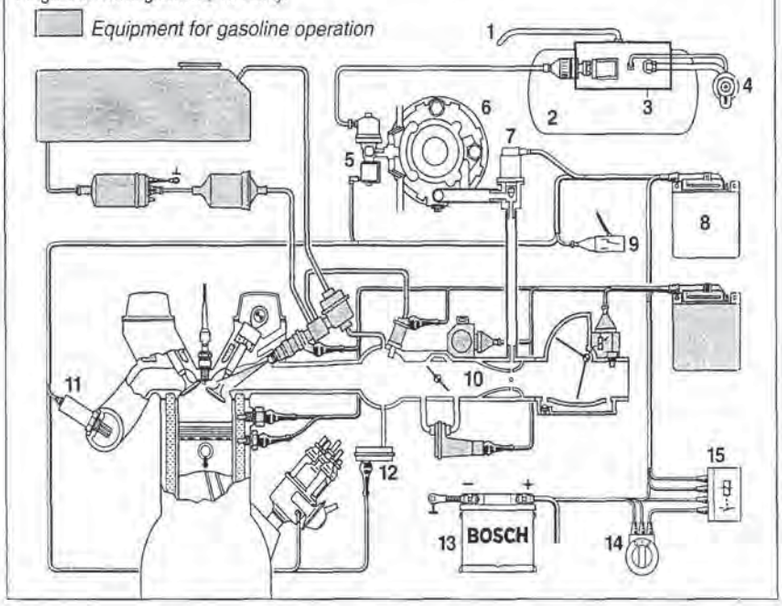
- Lower cruising ranges and increased fuel volumetric consumption compared to gasoline (although actual energy use is not higher than with gasoline, lowest with diesel engine).
- Special safety precautions are necessary, as LPG is pressurized.
- Pressurized gas cylinders require a lot of space, as the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



PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

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.

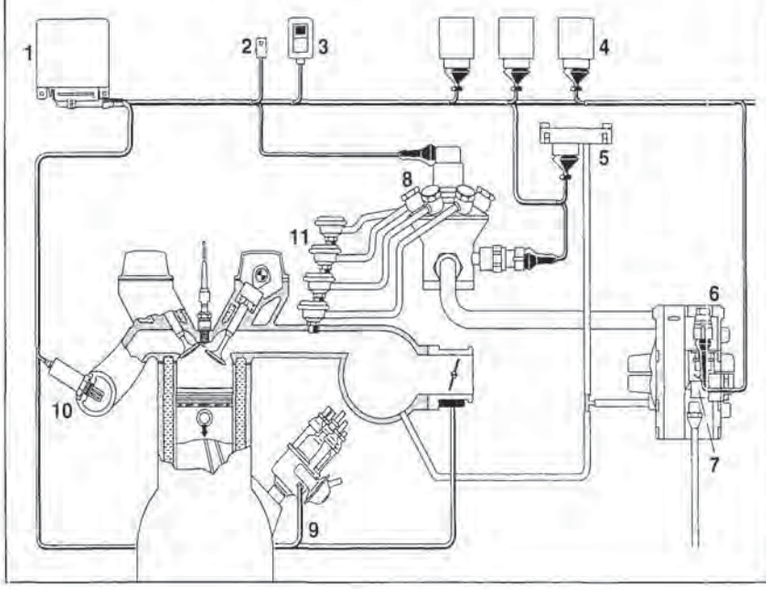
The flow-interrupt valve shuts immediately when the ignition is switched off. A gasoline/LPG switch installed in the instrument panel allows the operator to alternate between the two fuels.

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.



PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

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 primarily in passenger cars.

PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

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.

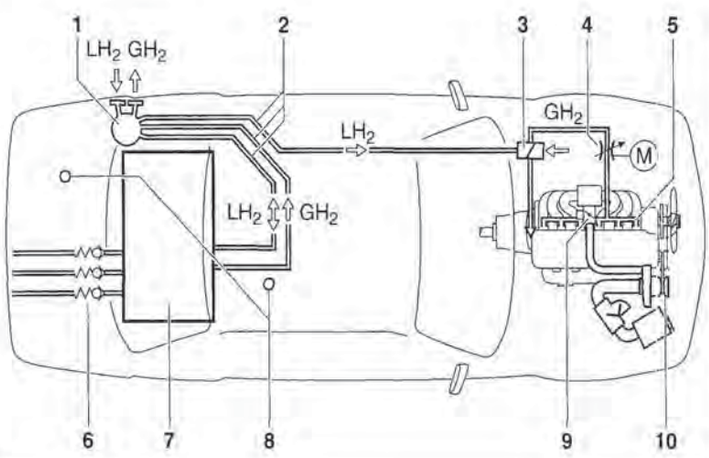
In the absence of comprehensive methanol-distribution networks to ensure universal availability, engines and engine-control systems must be designed for flexible dual-fuel operation (ranging from pure gasoline to max. 85% methanol). Alcohol places special, particularly critical demands upon engines and 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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US PATENT NO. 6,581,565

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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US PATENT NO. 6,581,565

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.

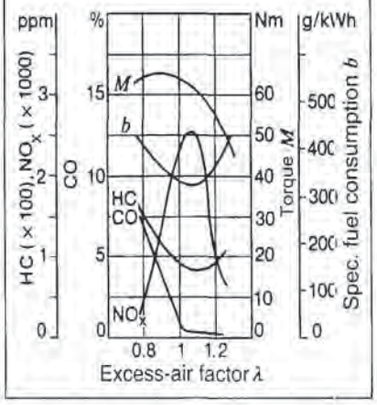
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US PATENT NO. 6,581,565

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.

PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

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

PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

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-

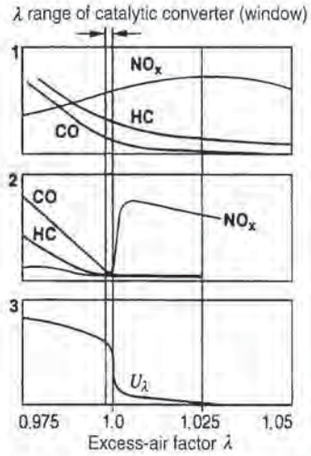
PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

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.

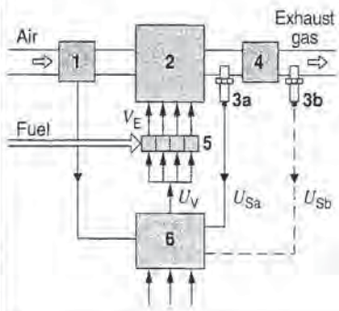
The reduction- and oxidation-type catalytic converters can also be combined in series to produce a dual-bed catalytic converter. This device uses air injection between the two converters to reduce not only NO_X emissions but also HC and CO levels. Its disadvantages include a more complex design (two 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

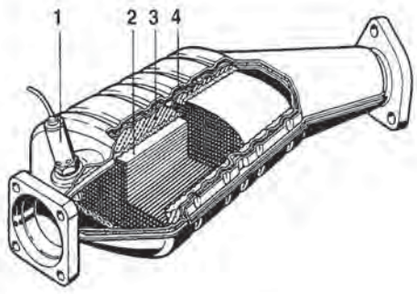
In the US, Europe and Japan, catalytic aftertreatment of the exhaust gas using a closed-loop three-way catalytic converter has proven to be a highly effective means of meeting the current low emission limits for CO, NO_X and HC.

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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US PATENT NO. 6,581,565

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 $\lambda > 1$ (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 ($\lambda = 1$)

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.

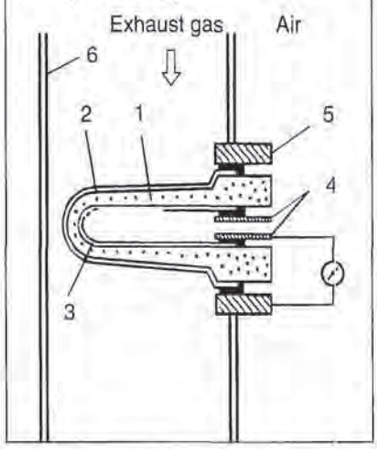
An electrical voltage is generated across the platinum-cermet electrodes in accordance with the Nernst principle.

$$U_s = R \cdot \frac{T}{4F} \cdot \ln\left(\frac{p_{O_2}''}{p_{O_2}}\right)$$

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.



PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

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.

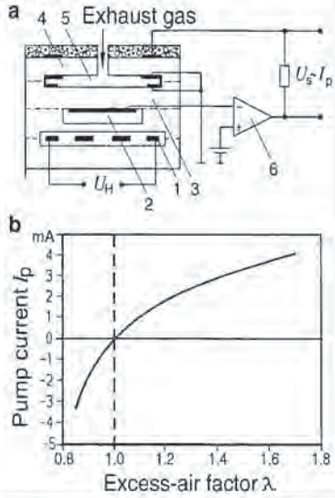
Broadband sensor

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.



PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

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

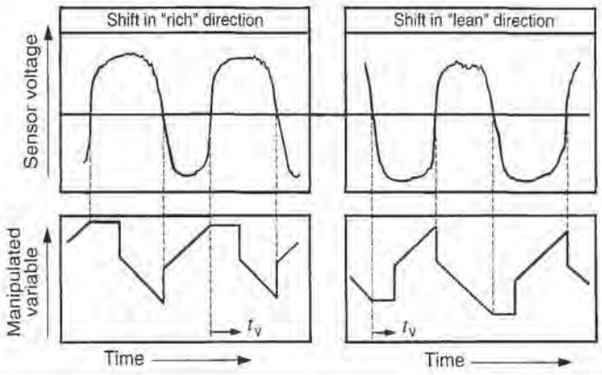
The effect of the disturbance on the accuracy at the voltage-jump point at $\lambda = 1$, as described above, has been minimized by a modified surface coating. Nevertheless, aging and environmental influences (poisoning) still have an effect. A sensor downstream of the catalytic converter is subjected 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.



PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

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-

PETITIONER'S EXHIBIT 1002
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