INTERNAL COMBUSTION ENGINE FUNDAMENTALS

SECOND EDITION

JOHN B. HEYWOOD

Internal Combustion Engine Fundamentals

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Internal Combustion Engine Fundamentals

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This second edition of my text is dedicated to my family:
my wife Peggy and our sons Jamie, Stephen (who died from ALS in 2006),
and Ben. They, and their families, have been wonderfully
supportive of my efforts in this challenging endeavor.
For this, I am truly grateful.

A documentary film, So Much So Fast, *was made about*
ALS, Stephen, Jamie, and our life together. The Los Angeles film critic's
review said of our family: "And what a family; close-knit, loving, and fiercely loyal."
I treasure those words.

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About the Author

John B. Heywood has been a faculty member at the Massachusetts Institute of Technology since 1968, where he was Sun Jae Professor of Mechanical Engineering and Director of the Sloan Automotive Laboratory. His interests are focused on internal combustion engines and their fuels, and on broader studies of future transportation technology and policy, fuel supply options, and vehicular air pollutant and greenhouse gas emissions. He has published over 230 papers in the technical literature, and is the author of five books, including this text. Dr. Heywood is a Fellow of the American Society of Mechanical Engineers, the British Institution of Mechanical Engineers, and the Society of Automotive Engineers. He has received many awards for his work, including the 1996 U.S. Department of Transportation Award for the Advancement of Motor Vehicle Research and Development, the 1999 Soichiro Honda Medal from the American Society of Mechanical Engineers, and the 2008 Society of Automotive Engineers Award for Contributions to Automotive Policy. He is also a Member of the National Academy of Engineering and a Fellow of the American Academy of Arts and Sciences. Dr. Heywood has a Ph.D. in mechanical engineering from MIT (1965), a Sc.D. from Cambridge University for his research contributions (1983), an honorary Doctor of Technology degree from Chalmers University of Technology, Sweden (1999), and an honorary D.Sc. from City University, London (2004).

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Preface

There are about two billion internal combustion engines in use in the world today. These engines enable key areas of our daily lives, propelling our many vehicles, generating electricity, and providing mechanical power in a wide range of applications. Their origin dates back to 1876 when Nicolaus Otto first developed the spark-ignition engine, and 1892 when Rudolf Diesel invented the compression-ignition engine. Since then, the utility of spark-ignition and diesel engines has steadily improved as our understanding of engine processes has increased, new technologies have become available, and market and regulatory requirements have become more demanding. A sense of urgency, driven largely by our need to combat global climate change, now requires ever-faster development of better engines and fuels, and exploration of alternative approaches. Increasing engine power density and efficiency and reducing engine emissions are really important objectives. The availability and effective use of our expanding knowledge base on engines and fuels are therefore critical.

The 1988 edition of this book has served as an educational text and professional reference in response to that need for some thirty years. Since 1988, we have steadily improved engine performance, reduced engine fuel consumption, developed air pollutant control technologies that have reduced engine emissions, improved the quality of our mainstream petroleum-based fuels, and made a start on reducing transportation's greenhouse gas emissions. Obviously, over these past thirty years, much new engineering knowledge relevant to engines and fuels has been developed. The purpose of this second edition of my book is to incorporate this new material, update the existing knowledge base, and make available a modern, broader, and thus more useful engine text and reference.

There is a massive amount of material, both analytical and experimentally based, available on internal combustion engines, and no text can include it all. The emphasis here is on the key physical and chemical processes that govern engine operation and design. These include the thermodynamics of energy conversion in engines, the physics and chemistry that govern engine combustion, the engine's fuel requirements, the important fluid flow, heat transfer, friction, and lubrication processes in engines, and the engine's dynamic behavior. These all influence engine performance, efficiency, and emissions.

There are two main types of internal combustion engine: spark-ignition and diesel. The primary organizing approach for the material in this text is how the mixture of fuel and air formed inside the engine cylinder is ignited. From *method of ignition*—spark-ignited or compression-ignited—follows each type of engine's operating cycle, fuel requirements, mixture preparation approach, combustion process, combustion chamber configuration, method used to control load, air pollutant formation mechanisms and control approaches, performance and efficiency characteristics, and greenhouse gas emissions. While many engine processes are similar in both types of engines, the method of ignition is fundamentally different. The consequences of that difference underlie the overall organization of this book.

The book is arranged in four major sections. The first (Chaps. 1 to 5) provides an introduction to, and overview of, the operating characteristics of spark-ignition and compression-ignition (diesel) engines, defines the parameters used to characterize engine operation, and develops the thermodynamics and combustion theory required for a quantitative analysis of engine behavior. It concludes with an integrated treatment of the methods used for analyzing idealized models of internal combustion engine operating cycles.

The second section (Chaps. 6 to 8) focuses on engine flow phenomena. The details of the gas exchange process—intake and exhaust processes in four-stroke and scavenging in two-stroke cycle engines—and the various methods of boosting engines—turbocharging and supercharging—are discussed. The several fuel metering approaches used in spark-ignition engines are reviewed next. Then, the key features of the flows setup inside the engine cylinder are described. These flow processes (along with engine boost levels) control the amount of air the engine will induct, and thus its power, and largely govern the rate at which the fuel-air mixture in the cylinder will burn.

The third section of the book examines engine combustion phenomena. These chapters (9 to 11) are especially important to smooth and robust engine behavior. The combustion process releases the fuel's chemical energy within the engine's cylinders for eventual conversion to work. The fraction of the fuel's energy that is converted depends strongly on how that combustion process occurs. The spark-ignition and compression-ignition combustion processes (Chaps. 9 and 10, respectively) thus influence essentially all aspects of engine behavior. Air pollutants are undesirable byproducts of combustion. Our extensive knowledge of how air pollutants form inside the engine and how such emissions can be controlled is reviewed in Chap. 11.

The last section of the book focuses on the operating characteristics of these engines. First, the basics of engine heat transfer and friction, both of which degrade engine performance, are developed in Chaps. 12 and 13. Then, Chap. 14 describes the methodologies available for predicting details of key engine processes and overall engine behavior based on realistic models of engine flow and combustion phenomena. The various thermodynamic- and fluid-mechanic-based computer codes that have been developed over the past several decades are widely used in engine research, development, and design, so knowledge of their basic structure and capabilities is important. Chapter 15 then summarizes how the operating characteristics—power, efficiency, and emissions—of spark-ignition and diesel engines depend on the major engine design and operating variables. These final two chapters effectively integrate our analysis-based understanding and practical knowledge of individual engine processes together to describe and explain overall spark-ignition engine, and compression-ignition engine, behavior.

While this book contains much advanced material on engine design and operation intended for the practitioner, each major topic is developed from its beginnings and the more sophisticated chapters have introductory sections to facilitate their use in undergraduate courses. The chapters are extensively cross-referenced and indexed. Each chapter is fully illustrated and referenced, and includes problems for both undergraduate and graduate student courses.

John B. Heywood

Acknowledgments

Many individuals and organizations have assisted me as I have worked on this revision of my engine text over the past ten or so years. I am especially grateful to my colleagues in the Sloan Automotive Laboratory at MIT: Professors Wai Cheng, William Green, and James Keck, and Drs. Leslie Bromberg, Daniel Cohn, Tian Tian, and Victor Wong. They have helped create a stimulating and collegial environment within which we have all expanded our knowledge and understanding of engine and fuels phenomena. Many of the graduate students in our lab have made significant contributions to this text through their research; their names can often be found in the reference list at the end of each chapter.

I am indebted also to many companies in the automotive and energy industries that have supported the research that my colleagues and I have carried out on engine and fuels topics. The fellow professionals in these companies who have helped us in our work (much of it relevant to topics I cover in this book) have provided important practical knowledge and insights that significantly enhanced the value of what we have been able to do. Many engineering advances are a result of team efforts, and as a researcher I really appreciate the contributions of engine practitioners. I also thank those in the engine business who helped me by providing drawings of recent engines that help anchor engine theory in the real world. Their companies are acknowledged in the relevant figure captions.

I want to thank my colleagues and the leadership of our Mechanical Engineering Department at MIT for their encouragement and support in this major endeavor. My department provided administrative support for a part of this effort. My assistant, Karla Stryker-Currier, was responsible for typing and upgrading the many drafts of each chapter and the final versions submitted to the publisher. Her extensive and thorough efforts to help bring this new edition to fruition, over many years, are much appreciated.

I have been fortunate to find focused time for much of the writing, away from the disruptions of regular life, during several long stays at Montestigliano, a farm in Tuscany—the most beautiful place I have ever lived. I am most grateful to the people who manage this Azienda Agricola for their help in making these visits so productive.

Finally, my family has strongly supported me in this time-consuming endeavor. My wife Peggy and sons Jamie, Stephen (now deceased), and Ben continually urged me to "keep going" until the task was completed. As noted in my dedication, I am truly grateful for their encouragement.

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Commonly Used Symbols, Subscripts, and Abbreviations[a]

SYMBOLS

a	Acceleration
	Crank radius
	Sound speed
	Specific availability
A	Area
A_C	Valve curtain area
A_{ch}	Cylinder head area
A_e	Exhaust port area
A_E	Effective area of flow restriction
A_i	Inlet port area
A_p	Piston crown area
A_v	Valve open area
B	Cylinder bore
	Steady-flow availability
c	Distance of piston from TC position
	Specific heat
c_p	Specific heat at constant pressure
c_s	Soot concentration (mass/volume)
c_v	Specific heat at constant volume
C	Absolute gas velocity
C_D	Discharge coefficient
	Vehicle drag coefficient
C_s	Swirl coefficient
D	Diameter
	Diffusion coefficient
D_d	Droplet diameter
D_{SM}	Sauter mean droplet diameter
D_v	Valve diameter
e	Radiative emissive power
	Specific energy
E_A	Activation energy
f	Coefficient of friction
	Fuel mass fraction
F	Force
g	Gravitational acceleration
	Specific Gibbs free energy

[a]Nomenclature specific to a section or chapter is defined in that section or chapter.

G	Gibbs free energy
h	Clearance height
	Oil film thickness
	Specific enthalpy
h_c	Heat-transfer coefficient
h_p	Port open height
h_s	Sensible specific enthalpy
H	Enthalpy
I	Moment of inertia
J	Flux
k	Thermal conductivity
	Turbulent kinetic energy
k_i^+, k_i^-	Forward, backward rate constants for *i*th reaction
K	Constant
K_c	Equilibrium constant expressed in concentrations
K_p	Equilibrium constant expressed in partial pressures
l	Characteristic length scale
	Connecting rod length
l_T	Characteristic length scale of turbulent flame
L	Piston stroke
L_I	Sound intensity level
$\tilde{L}_{\mathrm{LO}}$	Normalized spray lift-off length
L_n	Fuel-injection-nozzle orifice length
L_v	Valve lift
m	Mass
$\dot{m}$	Mass flow rate
m_r	Mass of residual gas
M	Mach number
	Molecular weight
n	Number of moles
	Polytropic exponent
n_c	Number of cylinders
n_R	Number of crank revolutions per power stroke
N	Crankshaft rotational speed
	Soot particle number density
	Turbocharger shaft speed
p	Cylinder pressure
	Pressure
P	Power
$\dot{q}$	Heat-transfer rate per unit area
	Heat-transfer rate per unit mass of fluid
Q	Heat transfer
$\dot{Q}$	Heat-transfer rate
Q_{ch}	Fuel chemical energy release or gross heat release
Q_{HV}	Fuel heating value
Q_n	Net heat release
r	Radius
r_c	Compression ratio
R	Connecting rod length/crank radius
	Gas constant
	Radius

R^+, R^-	One-way reaction rates
R_s	Swirl ratio
s	Crank axis to piston pin distance
	Specific entropy
S	Entropy
	Spray penetration
S_b	Turbulent burning speed
S_L	Laminar flame speed
S_p	Piston speed
t	Time
T	Temperature
	Torque
u	Specific internal energy
	Velocity
u'	Turbulence intensity
u_s	Sensible specific internal energy
u_T	Characteristic turbulent velocity
U	Compressor/turbine impellor tangential velocity
	Fluid velocity
	Internal energy
v, υ	Specific volume
	Velocity
v_{ps}	Valve pseudo-flow velocity
v_{sq}	Squish velocity
V	Cylinder volume
	Volume
V_c	Clearance volume
V_d	Displaced cylinder volume
w	Relative gas velocity
	Soot surface oxidation rate
W	Work transfer
W_c	Work per cycle
W_p	Pumping work
x, y, z	Spatial coordinates
x	Mass fraction
$\tilde{x}$	Mole fraction
x_b	Burned mass fraction
x_r	Residual mass fraction
y	H/C ratio of fuel
	Volume fraction
Y_α	Concentration of species α per unit mass
z	Distance, piston crown to cylinder head
Z	Inlet Mach index
α	Angle
	Thermal diffusivity $k/(\rho c)$
β	Angle
γ	Specific heat ratio c_p/c_v
Γ_c	Angular momentum of charge
δ	Boundary-layer thickness
δ_L	Laminar flame thickness
$\Delta h^\circ_{f,i}$	Molal enthalpy of formation of species i

$\Delta\theta_b$	Rapid burning angle
$\Delta\theta_d$	Flame development angle
ε	$4/(4 + y)$: y = H/C ratio of fuel
	Turbulent kinetic energy dissipation rate
ζ	Percentage of stoichiometric air entrained into fuel spray
η_a	Availability conversion efficiency
η_c	Combustion efficiency
η_{ch}	Charging efficiency
η_f	Fuel conversion efficiency
η_T	Turbine isentropic efficiency
η_{tr}	Trapping efficiency
η_v	Volumetric efficiency
θ	Crank angle
λ	Relative air/fuel ratio
Λ	Delivery ratio
μ	Dynamic viscosity
μ_i	Chemical potential of species i
ν	Kinematic viscosity μ/ρ
ν_i	Stoichiometric coefficient of species i
ξ	Flow friction coefficient
ρ	Density
$\rho_{a,0}$, $\rho_{a,i}$	Air density at standard, inlet conditions
σ	Normal stress
	Standard deviation
	Stefan-Boltzmann constant
	Surface tension
τ	Characteristic time
	Induction time
	Shear stress
τ_{id}	Ignition delay time
ϕ	Fuel/air equivalence ratio
Φ	Flow compressibility function [Eq. (C.11)]
	Isentropic compression function [Eqs. (4.15*b*), (4.25*b*)]
ψ	Molar N/O ratio
Ψ	Isentropic compression function [Eqs. (4.15*a*), (4.25*a*)]
ω	Angular velocity
	Frequency

SUBSCRIPTS

a	Air
b	Burned gas
c	Coolant
	Cylinder
C	Compression stroke
	Compressor
cr	Crevice
e	Equilibrium
	Exhaust

E	Expansion stroke
f	Flame
	Friction
	Fuel
g	Gas
i	Indicated
	Intake
	Species i
ig	Gross indicated
in	Net indicated
l	Liquid
L	Laminar
p	Piston
	Port
P	Prechamber
r, θ, z	r, θ, z components
R	Reference value
s	Isentropic
T	Nozzle or orifice throat
	Turbine
	Turbulent
u	Unburned
v	Valve
w	Wall
x, y, z	x, y, z components
0	Reference value
	Stagnation value

NOTATION

Δ	Difference
$-$	Average or mean value
$\sim$	Value per mole
[ ]	Concentration, moles/vol
{ }	Mass fraction
$\cdot$	Rate of change with time
u	Bold type, vector (e.g., velocity)

ABBREVIATIONS

(A/F)	Air/fuel ratio
BC, ABC, BBC	Bottom-center crank position, after BC, before BC
bmep	Brake mean effective pressure
CN	Fuel cetane number
Da	Damköhler number τ_T/τ_L
EGR	Exhaust gas recycle
EI	Emission index
EPC, EPO	Exhaust port closing, opening

EVC, EVO	Exhaust valve closing, opening
(F/A)	Fuel/air ratio
(G/F)	Gas/fuel ratio
gimep	Gross indicated mean effective pressure
IPC, IPO	Inlet port closing, opening
IVC, IVO	Inlet valve closing, opening
mep	Mean effective pressure
nimep	Net indicated mean effective pressure
Nu	Nusselt number $h_c l/k$
ON	Fuel octane number
Re	Reynolds number $\rho u l/\mu$
sfc	Specific fuel consumption
TC, ATC, BTC	Top-center crank position, after TC, before TC
We	Weber number $\rho_l u^2 D/\sigma$

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CHAPTER 1

Engine Types and Their Operation

1.1 INTRODUCTION AND HISTORICAL PERSPECTIVE

The purpose of internal combustion engines is to produce mechanical power from the chemical energy contained in the fuel. In *internal* combustion engines, as distinct from *external* combustion engines, this energy is released by burning or oxidizing the fuel *inside* the engine. The fuel-air mixture before combustion and the burned products after combustion are the actual working fluids. The work transfers that provide the desired power output occur directly between these working fluids and the mechanical components of the engine. The internal combustion engines that are the subject of this book are spark-ignition (SI) engines (sometimes called Otto engines, or gasoline or petrol engines, though other fuels can be used) and compression-ignition (CI) or diesel engines.[a] Because of their simplicity, ruggedness, high power to weight ratio, efficiency, and low cost, these two types of engine have found wide application in transportation (land, sea, and air) and power generation. It is the fact that combustion takes place inside the work-producing part of these engines that makes their design and operating characteristics fundamentally different from those of other types of engine.

Power-producing engines have served human beings for over two and a half centuries. For the first 150 years, water, converted to steam, was interposed between the combustion gases produced by burning the fuel and the work-producing piston-in-cylinder expander. It was not until the 1860s that the internal combustion engine became a practical reality.[1,2] The early engines developed for commercial use burned coal-gas air mixtures at atmospheric pressure—there was no compression before combustion. J. J. E. Lenoir (1822–1900) developed the first marketable engine of this type. Gas and air were drawn into the cylinder during the first half of the piston stroke. The charge was then ignited with a spark, the pressure increased, and the burned gases then delivered power to the piston for the second half of the stroke. The cycle was completed with an exhaust stroke. Some 5000 of these engines were built between 1860 and 1865 in sizes up to six horsepower. Efficiency was at best about 5%.

A more successful development—an atmospheric engine introduced in 1867 by Nicolaus A. Otto (1832–1891) and Eugen Langen (1833–1895)—used the pressure rise resulting from combustion of the fuel-air charge early in the outward stroke to accelerate a free piston and rack assembly so its momentum would generate a vacuum in the cylinder. Atmospheric pressure then pushed the piston inward, with the rack engaged through a roller clutch to

[a]The gas turbine is also, by this definition, an "internal combustion engine." Conventionally, however, the term is used for spark-ignition and compression-ignition engines. The operating principles of gas turbines are fundamentally different, and they are not discussed in this book.

the output shaft. Production engines, of which about 5000 were built, obtained thermal efficiencies of up to 11%. A slide valve controlled intake, ignition by a gas flame, and exhaust.

To overcome this engine's shortcomings of low thermal efficiency and excessive size and weight, Otto proposed an engine cycle with four piston strokes: an intake stroke, then a compression stroke before ignition, an expansion or power stroke where work was delivered to the crankshaft, and finally an exhaust stroke. He also proposed incorporating a stratified-charge induction system, though this was not achieved in practice. His prototype four-stroke engine first ran in 1876. A comparison between the Otto engine and its atmospheric-type predecessor indicates the reason for its success (Table 1.1): the enormous reduction in engine weight and volume. This was the breakthrough that effectively founded the internal combustion engine industry. By 1890, almost 50,000 of these engines had been sold in Europe and the United States.

In 1884, an unpublished French patent issued in 1862 to Alphonse Beau de Rochas (1815–1893) was found that described the principles of the four-stroke cycle. This chance discovery cast doubt on the validity of Otto's own patent for this concept, and in Germany, it was declared invalid. Beau de Rochas also outlined the conditions under which maximum performance and efficiency in an internal combustion engine could be achieved. These were:

1. The largest possible cylinder volume with the minimum boundary surface
2. The greatest possible working speed
3. The greatest possible expansion ratio
4. The greatest possible pressure at the beginning of expansion

The first condition holds heat losses from the charge to a minimum. The second condition increases the power output from a given size engine. The third condition recognizes that the greater the expansion of the postcombustion gases, the greater the amount of work extracted. The fourth condition recognizes that higher initial pressures make greater expansion possible and give higher pressures throughout the process, both resulting in greater work transfer. Although Beau de Rochas' unpublished writings predate Otto's developments, he never reduced these ideas to practice. Thus Otto, in the broader sense, was the inventor of the modern internal combustion engine as we know it today.

Further developments followed fast once the full impact of what Otto had achieved became apparent. By the 1880s, several engineers (e.g., Dugald Clerk, 1854–1913, James Robson, 1833–1913, in England, and Karl Benz, 1844–1929, in Germany) had successfully developed two-stroke cycle internal combustion engines where the exhaust and intake processes occur during the end of the power stroke and the beginning of the compression

TABLE 1.1 Comparison of Otto's early four-stroke cycle and Otto-Langen's engines[2]

	Otto and Langen	Otto's four-stroke
Brake horsepower	2	2
Weight, lb, approx.	4000	1250
Piston displacement, in^3	4900	310
Power strokes per minute	28	80
Shaft speed, rev/min	90	160
Mechanical efficiency, %	68	84
Overall efficiency, %	11	14
Expansion ratio	10	2.5

stroke. James Atkinson (1846–1914) in England made an engine with a longer expansion than compression stroke, which had a high efficiency for the times but mechanical weaknesses. It was recognized that efficiency was a direct function of expansion ratio, yet compression ratios were limited to less than four if serious knock problems were to be avoided with the available fuels. Substantial carburetor and ignition system developments were required, and occurred, before high-speed gasoline engines suitable for automobiles became available in the late 1880s. Stationary engine progress also continued. By the late 1890s, large single-cylinder engines of 1.3-m bore fueled by low-energy blast furnace gas produced 600 bhp at 90 rev/min. In Britain, legal restrictions on volatile fuels turned their engine builders toward kerosene. Low compression ratio "oil" engines with heated external fuel vaporizers and electric ignition were developed with efficiencies comparable with those of gas engines (14 to 18%). The Hornsby-Ackroyd engine became the most popular oil engine in Britain, and was also built in large numbers in the United States.[2]

In 1892, the German engineer Rudolf Diesel (1858–1913) outlined in his patent a new form of internal combustion engine. His concept of initiating combustion by injecting a liquid fuel into the high-temperature air in the cylinder produced by compression permitted a doubling of efficiency over the other internal combustion engines then available. Much greater compression and expansion ratios, without detonation or knock, were now possible. However, even with the efforts of Diesel and the resources of M.A.N. in Ausburg combined, it took 5 years to develop a practical engine.

Engine developments, perhaps less fundamental but nonetheless important to the steadily widening internal combustion engine markets, have continued ever since.[2-4] There has always been an interest in engine geometries different from the standard reciprocating piston-in-cylinder, connecting rod, and crankshaft arrangement. Especially, there has been an interest in rotary internal combustion engines. Although a wide variety of experimental rotary engines have been proposed over the years,[5] the first practical rotary internal combustion engine, the Wankel, was not successfully tested until 1957. That engine, which evolved through many years of research and development, was based on the designs of the German inventor Felix Wankel.[6,7] While the Wankel engine has been used in niche markets, its advantages of compactness and smoother operation have not been sufficient to overcome its high manufacturing cost.

Fuels have also had a major impact on engine development. The earliest engines used for generating mechanical power burned gaseous fuels. Gasoline, and lighter fractions of crude oil, became available in the late 1800s, and various types of carburetors were developed to vaporize the fuel and mix it with air. Before about 1905, there were few issues with gasoline; though compression ratios had to be low (4 or less) to avoid knock, the highly volatile fuel made starting easy and gave good cold weather performance. However, a serious crude oil shortage developed, and to meet the fivefold increase in gasoline demand between 1907 and 1915, the yield from crude had to be raised. Through the work of William Burton (1865–1954) and his associates of Standard Oil of Indiana, a thermal cracking process was developed whereby heavier oils were heated under pressure and decomposed into less complex, more volatile compounds. These thermally cracked gasolines satisfied demand, but their higher boiling point range created cold weather starting problems. Fortunately, electrically driven starters, introduced in 1912, came along just in time.

On the farm, kerosene was the logical fuel for internal combustion engines since it was used for heat and light. Many early farm engines had heated carburetors or vaporizers to enable them to operate with such a fuel.

The period following World War I saw a tremendous advance in our understanding of how fuels affect combustion, and especially the problem of knock. The antiknock effect of tetraethyl lead was discovered at General Motors,[4] and it became commercially available as a gasoline additive in the United States in 1923. In the late 1930s, Eugene Houdry found that vaporized oils passed over an activated catalyst at 450 to 480°C were converted to high-quality gasoline in much higher yields than was possible with thermal cracking. These advances, and others, permitted fuels with ever better antiknock properties to be produced in large quantities; thus engine compression ratios steadily increased, improving power and efficiency.

During the past several decades, new factors for change have become important and now significantly affect engine design and operation. These factors are, first, the need to control the automotive contribution to urban air pollution and, second, the need to achieve significant improvements in automotive fuel consumption.

The automotive air-pollution problem became apparent in the 1940s in the Los Angeles basin. In 1952, it was demonstrated by Prof. A. J. Haagen-Smit that the smog problem there resulted from reactions between oxides of nitrogen and hydrocarbon compounds in the presence of sunlight.[8] In due course it became clear that the automobile was a major contributor to hydrocarbon and oxides of nitrogen emissions, as well as the prime cause of high carbon monoxide levels in urban areas. Diesel engines are a significant source of small soot or smoke particles, as well as hydrocarbons and oxides of nitrogen. Table 1.2 outlines the dimensions of the problem. As a result of these developments, emission standards for automobiles were introduced first in California, then nationwide in the United States, starting in the 1960s. Emission standards in Japan and Europe, and for other engine applications, have followed. Substantial reductions in emissions from spark-ignition and diesel engines have been achieved. Both the use of catalysts in SI engine exhaust systems for

TABLE 1.2 The automotive urban air-pollution problem: typical vehicle emissions*

			Automobile emissions, SI engines		Truck emissions, diesel engines,	
Pollutant	Impact	Mobil source emissions as % of total†	Precontrol vehicles, g/km‡	Current vehicles, g/km	Precontrol engines, g/kWh	Current engines, g/kWh
Oxides of nitrogen (NO and NO_2)	Reactant in photochemical smog; NO_2 is toxic	50–60	2.0	0.03	21	0.25
Carbon monoxide (CO)	Toxic	60	60	2	~ 20	low
Unburned hydrocarbons (HC, many hydrocarbon compounds)	Reactant in photochemical smog	25	10	0.05	~ 1	low
Particulates (soot, hydrocarbons, sulfates)	Some of HC compounds mutagenic; reduces visibility	5–10	0.5§	0.007§	1	0.02

*Varies from country to country. The United States, Canada, Western Europe, and Japan have standards with different degrees of severity. The United States, Europe, and Japan have different test procedures. Standards are strictest in the United States and Japan.
†Depends on type of urban area and source mix. Approximate percentages.
‡Average values for pre-1968 automobiles that had no emission controls, determined by U.S. test procedure that simulates typical urban and highway driving. Exhaust emissions, except for HC where 55% are exhaust emissions, 20% are evaporative emissions from fuel tank and carburetor, and 25% are crankcase blowby gases.
§Diesel engine automobiles only. Particulate emissions from spark-ignition engines are relatively low.

emissions control and concern over the toxicity of lead antiknock additives have resulted in the reappearance of unleaded gasoline as the dominant part of the automotive fuels market. These emission-control requirements and fuel developments have produced significant changes in the way internal combustion engines are now designed and operated.

Internal combustion engines are also an important source of noise. There are several sources of engine noise: the exhaust system, the intake system, the fan used for cooling, and the engine block surface. The noise may be generated by aerodynamic effects, may be due to forces that result from the combustion process, or may result from mechanical excitation by rotating or reciprocating engine components. Vehicle noise legislation to reduce this impact on the ambient environment (and thus on people) was first introduced in the early 1970s.

During the 1970s, the price of crude petroleum rose rapidly to several times its cost (in real terms) in 1970. In the 1980s, the price of crude oil fell, and then fluctuated at relatively low levels until the early 2000s when it rose to close to its late 1970s values. The price then fell rapidly, and then rose again. Currently, the growth in oil demand in the developing world, the uncertainty in future extraction from established fields and discovery of new sources of oil, and the nonuniform concentration of petroleum reserves in a few nations, suggest that the balance between global oil production and transportation fuel demand will be tight over the next few decades. This uncertainty regarding the longer-term availability of adequate supplies of petroleum-based fuels is creating substantial pressures for significant improvements in internal combustion engine efficiency (in all the engine's many applications). Much work is being done to develop the supply and use of alternative fuels to gasoline and diesel. Of the nonpetroleum-based fuels, natural gas, methanol (methyl alcohol), and biomass-derived fuels such as ethanol (ethyl alcohol) and biodiesel are receiving significant attention. Synthetic gasoline and diesel are being made from tar (oil) sands, and could be produced from shale oil or coal. Hydrogen is being considered as a longer-term zero carbon containing possibility.

The growing consumption of fossil fuels has raised the concern that the greenhouse gas (GHG) emissions from our energy supply and use are causing global warming that could lead to changes in our climate. Emissions of carbon dioxide, along with other GHGs—methane, nitrous oxide, three groups of fluorinated gases (sulfur hexafluoride, hydrofluorocarbons, and perfluorocarbons), ozone—will need to be significantly reduced over the next several decades. Thus, internal combustion engines will need to become more efficient, and low GHG emitting sources of energy will need to be developed so that consumption of petroleum-based fuels—gasoline and diesel—can be significantly reduced. Transportation is estimated to be the source of about one-quarter of the world's GHG emissions.

Table 1.3 lists the CO_2 emissions of various fuels and other sources of energy that might be used in transportation. Emissions from the various fossil fuels listed vary by about a factor of two. Emissions from biofuel production are generally lower (and could be significantly lower), depending on the biomass feedstock, the choice of fuel produced, and the process used to produce that fuel.[b]

The lower value given for hydrogen (which contains no carbon) is based on the current industrial hydrogen production process—steam reforming of natural gas. The electricity carbon dioxide-emissions intensity value depends on the mix of coal, natural gas, nuclear, hydro, wind (and solar) used to generate the electricity. While this electricity generating mix varies country to country, the major roles of coal and natural gas are common to most regions.

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[b]The conversion of the carbon in the biomass source to CO_2 is often regarded as "carbon neutral" since that carbon came from CO_2 in the atmosphere. This topic is the subject of ongoing research.

TABLE 1.3 CO_2 emissions per unit chemical energy from various fuels or energy sources[9]

	gCO_2/MJ
Gasoline	93
Diesel (fuel oil)	99
Natural gas	74
Liquid petroleum gas	86
Ethanol*	34–73
Biodiesel	45–73
Hydrogen†	100–200
Electricity‡	90–160

*Varies with biomass feedstock and process used.
†From steam reforming of natural gas (low end) or from electrolysis (high end).
‡Depends on electricity generating system source mix (especially the fraction from coal).

What would such fuel changes mean for internal combustion engines? With appropriate changes in engine design and operation, natural gas and the liquid fuels listed in Table 1.3 can be effectively utilized; indeed engines using these fuels are in use today. While the potential for hydrogen as a major transportation energy source (actually an *energy storage* medium) is partly based on large-scale use of highly efficient fuel cell technology, it can be used effectively in suitably designed SI engines. Vehicle propulsion system electrification is already occurring through the use of hybrid electric vehicle (HEV) technology—a combination of a battery, electric motor, internal combustion engine, and generator. The next step in vehicle electrification is to expand the battery's energy storage capacity and recharge (in part) from the electricity supply grid: deployment of this plug-in hybrid (PHEV) technology is occurring. HEV and PHEV propulsion systems require an internal combustion engine, albeit with specific characteristics that improve its efficiency (see Sec. 1.7.2). Some view the pure battery electric vehicle is the final step in this electrification process. Whether, and how far into the future complete electrification might occur is currently unclear.

This brings us back to internal combustion engines. It might be thought that after over a century of development, the internal combustion engine has reached its peak and little potential for further improvement remains. Such is not the case. As spark-ignition and diesel engine technology evolves, these engines continue to show substantial improvements in efficiency, power density, degree of emission control, and operational capacity. Changes in engine operation and design are steadily improving engine performance in its broadest sense. New materials becoming available and more knowledge-based design offer the potential for continuing to reduce engine weight, size, and cost, for a given power output, and for different and more efficient internal combustion engine concepts. Emissions control technologies, in both the engine and the exhaust system, are becoming more effective and robust. Variable valve control is replacing fixed valve control approaches, with performance and efficiency benefits. Direct-injection gasoline engines, which offer improved dynamic engine control relative to port fuel injection, are now in large-scale production. These technologies are enabling increasing deployment of more highly boosted turbocharged gasoline and diesel engines.[10] Looking ahead, the engine development opportunities of the future are many and substantial. While they present a formidable challenge to automotive engineers, they will be made possible in large part by the enormous expansion of our knowledge of engine processes that the last several decades have witnessed.

1.2 ENGINE CLASSIFICATIONS

There are many different types of internal combustion engines. They can be classified by:[11]

1. *Application.* Automobile, truck, bus, locomotive, light aircraft, marine, portable power system, power generation
2. *Basic engine configuration.* Reciprocating engines (in turn subdivided by arrangement of cylinders: e.g., in-line, V, radial, opposed), rotary engines (Wankel and other geometries)
3. *Working cycle.* Four-stroke cycle: naturally-aspirated (admitting atmospheric air), supercharged (admitting precompressed air), and turbocharged (admitting air compressed in a compressor driven by an exhaust turbine). Two-stroke cycle: crankcase scavenged, supercharged, and turbocharged
4. *Valve or port design and location.* Four-stroke cycle: Overhead (or I-head) valves, underhead (or L-head) valves, with two, three, or four valves per cylinder, and fixed or variable valve control (timing, opening and closing points, and lift), rotary valves. Two-stroke cycle: cross-scavenged porting (inlet and exhaust ports on opposite sides of cylinder at one end), loop-scavenged porting (inlet and exhaust ports on same side of cylinder at one end), through- or uniflow-scavenged (inlet and exhaust ports or valves at different ends of cylinder)
5. *Fuel.* Gasoline (or petrol), fuel oil (or diesel fuel), natural gas, liquid petroleum gas (LPG), alcohols (methanol, ethanol), hydrogen, dual fuel
6. *Method of mixture preparation.* Carburetion or single-point fuel injection upstream of the throttle, fuel injection into the intake ports, fuel injection directly into the engine cylinder
7. *Method of ignition.* Spark ignition in engines where the in-cylinder fuel-air mixture is uniform and in stratified-charge engines where the mixture is nonuniform; compression ignition locally of the evolving in-cylinder fuel-air mixture in diesel engines, as well as ignition in natural gas engines by pilot injection of fuel oil)
8. *Combustion chamber design.* Open chamber (many designs: e.g., disc, wedge, hemisphere, pent-roof, bowl-in-piston), divided chamber (small and large auxiliary chambers; many designs: e.g., swirl chambers, prechambers)
9. *Method of load control.* Varying fuel and air flow together so mixture composition is essentially unchanged, control of fuel flow alone, a combination of these
10. *Method of cooling.* Water cooled, air cooled, uncooled (other than by natural convection and radiation)

All these distinctions are important and they illustrate the breadth of engine designs available. Because this book approaches the operating and emissions characteristics of internal combustion engines from a fundamental point of view, method of ignition has been selected as the primary classifying feature. From the method of ignition—SI or CI[c]—follow the important characteristics of the fuel used, method of mixture preparation, method of load control, combustion chamber design, details of the combustion process, engine emissions, and operating characteristics. Some of the other classifications are used as subcategories within this basic classification. The engine operating cycle—four-stroke or two-stroke is next in importance; the principles of these two cycles are described in the following section.

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[c]In the remainder of the book, these terms will often be abbreviated by SI and CI, respectively.

1.3 ENGINE OPERATING CYCLES

Most of this book is about *reciprocating engines*, where each piston moves back and forth in a cylinder and transmits power from the high-pressure and temperature burned gases inside the cylinder through the piston and the connecting rod and crank mechanism to the drive shaft as shown in Fig. 1.1. The rotation of the crank produces a cyclical piston motion. The piston comes to rest at the top-center (TC) crank position and bottom-center (BC) crank position when the cylinder volume is a minimum or maximum, respectively.[d] The minimum cylinder volume is called the clearance volume V_c. The volume swept out by the piston, the difference between the maximum or total volume V_t and the clearance volume, is called the displaced or swept volume V_d. The ratio of maximum volume to minimum volume is the compression ratio r_c. Values of r_c are 8 to 12 for SI engines and typically in the ranges of 14 to 22 for CI engines.

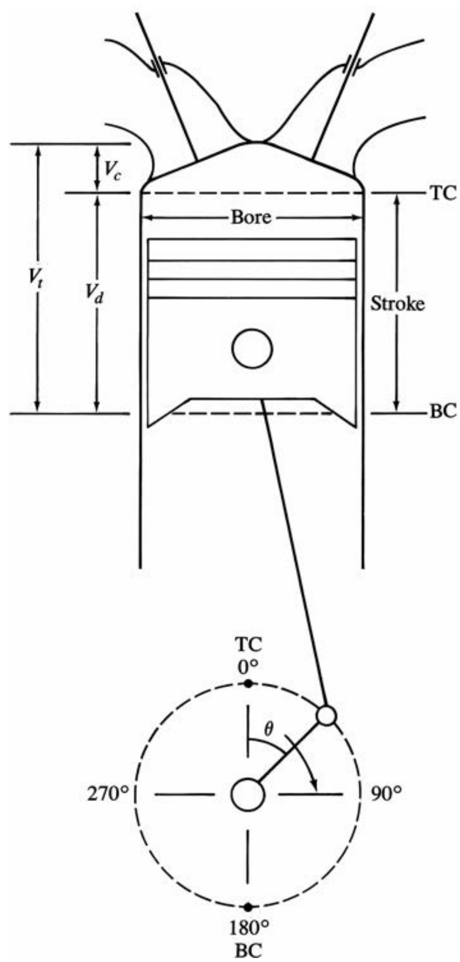


Figure 1.1 Basic geometry of the reciprocating internal combustion engine. V_c, V_d, and V_t indicate clearance, displaced, and total cylinder volumes.

[d]These crank positions are also referred to as top-dead-center (TDC) and bottom-dead-center (BDC).

The majority of reciprocating engines operate on what is known as the *four-stroke cycle*. Each cylinder requires four strokes of its piston—two revolutions of the crankshaft—to complete the sequence of events that produces one power stroke. Both SI and CI engines use this cycle that comprises (Fig. 1.2):

1. *An intake stroke*, which starts with the piston at TC and ends with the piston at BC, which draws fresh air or fuel-air mixture into the cylinder. To increase the mass inducted, the inlet valve opens shortly before the stroke starts and closes after it ends.
2. *A compression stroke*, which starts with the piston at BC and ends at TC, when the mixture inside the cylinder is compressed to a small fraction of its initial volume. Toward the end of the compression stroke, combustion is initiated and the cylinder pressure rises more rapidly.
3. *A power stroke*, or expansion stroke, which starts with the piston at TC and ends at BC as the high-temperature, high-pressure gases push the piston down and force the crank to rotate. About five times as much work is done on the piston during the power stroke as the piston had to do during compression. As the piston approaches BC, the exhaust valve opens to initiate the exhaust process and drop the cylinder pressure to close to the exhaust system pressure.
4. *An exhaust stroke*, where, as the piston moves from BC to TC, the remaining burned gases exit the cylinder: first, because the cylinder pressure may be significantly higher than the exhaust pressure; then as these gases are swept out by the piston as it moves toward TC. As the piston approaches TC the inlet valve opens and just after TC the exhaust valve closes. The cycle then starts again.

Though often called the Otto cycle after its inventor, Nicolaus Otto, who built the first engine operating on these principles in 1876, the more descriptive four-stroke nomenclature is preferred.

The four-stroke cycle requires, for each engine cylinder, two crankshaft revolutions for each power stroke. To obtain a higher power output from a given engine size, and a simpler valve design, the *two-stroke* cycle was developed. The two-stroke cycle is applicable to both SI and CI engines.

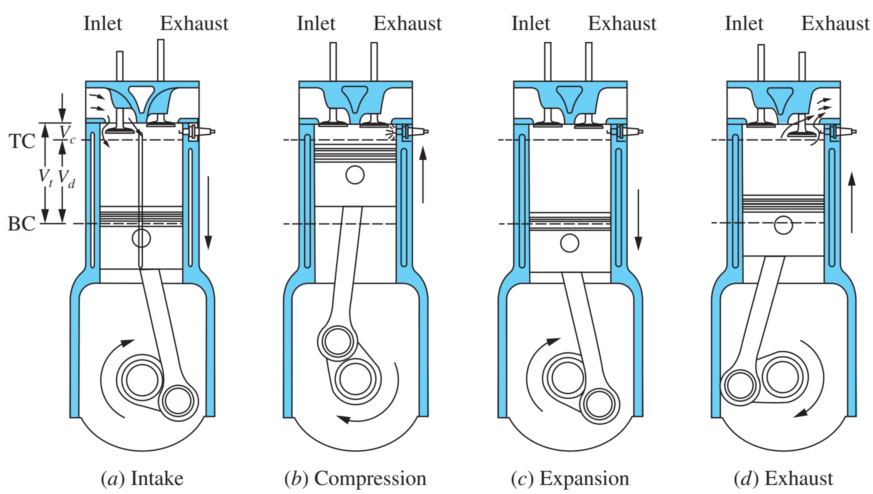


Figure 1.2 The four-stroke operating cycle.[12]

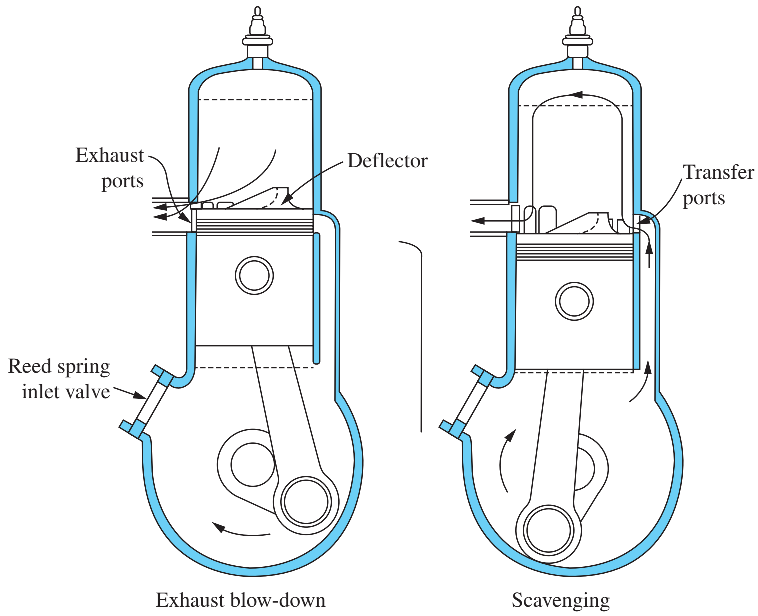


Figure 1.3 The two-stroke operating cycle. A crankcase-scavenged engine is shown.[12]

Figure 1.3 shows one of the simplest types of two-stroke engine designs. Ports in the cylinder liner, opened and closed by the piston motion, control the exhaust flow out of the cylinder and the fresh charge flow into the cylinder, while the piston is close to BC. The two strokes are:

1. *A compression stroke*, which starts with the closing of the fresh charge transfer ports and then the exhaust ports, and compresses the cylinder contents as the piston moves up the cylinder, and also draws fresh charge into the crankcase through the inlet Reed valve. As the piston approaches TC, combustion is initiated.
2. *A power or expansion stroke*, similar to that in the four-stroke cycle until the piston approaches BC, when first the exhaust ports and then the transfer ports are uncovered (Fig. 1.3). Most of the burnt gases exit the cylinder in an exhaust blowdown process. When the transfer ports are uncovered, the fresh charge that has been compressed in the crankcase flows into the cylinder. The piston and the ports are generally shaped to deflect the incoming charge from flowing directly into the exhaust ports, and to achieve effective scavenging of the residual in-cylinder burned gases by this fresh charge.

Each engine cycle with one power stroke is completed in one crankshaft revolution. However, it is difficult to fill completely the displaced volume with fresh charge, and some of the fresh mixture flows directly out of the cylinder during the scavenging process.[e] The example shown is a *cross-scavenged* design; other approaches use *loop-scavenging* or *uniflow* gas exchange processes (see Sec. 6.6).

[e]It is primarily for this reason that two-stroke SI engines are at a disadvantage because the lost fresh charge contains fuel and air.

1.4 ENGINE COMPONENTS

Cutaway drawings of a four-stroke spark-ignition (SI) engine and a diesel (CI) engine are shown in Figs. 1.4 and 1.5, respectively. The SI engine is a four-cylinder in-line automobile engine. The major components are labeled. The diesel is a six-cylinder in-line heavy-duty truck engine. The function of the major components of these engines and their construction materials will now be reviewed.

The engine cylinders are contained in the engine block. The block has traditionally been made of gray cast iron because of its good wear resistance and low cost, but is often now made of aluminum. Passages for the cooling water are cast into the block. Heavy-duty and truck engines often use removable cylinder sleeves pressed into the block that can be replaced when worn. These are called *wet liners* or *dry liners* depending on whether the sleeve is in direct contact with the cooling water. Aluminum is used in automotive SI engine blocks to reduce engine weight. Iron cylinder liners may be inserted at the casting stage, or later on in the machining and assembly process. The crankcase is often integral with the cylinder block.

The crankshaft has traditionally been a steel forging; nodular cast iron crankshafts are also accepted practice in automotive engines. The crankshaft is supported in main bearings. The number of crankshaft bearings depends largely on the engine's loading and maximum speed. The maximum number of main bearings is one more than the number of cylinders; there may be less. The crank has eccentric portions (crank throws); the connecting rod big-end bearings attach to the crank pin on each throw. Both main and connecting rod bearings use steel-backed precision inserts with bronze, babbit, or aluminum as the bearing materials. The crankcase is sealed at the bottom with a pressed-steel or cast aluminum oil pan, which acts as an oil reservoir for the lubricating system.

Pistons are made of aluminum in smaller engines or cast iron in larger slower-speed engines. The piston both seals the cylinder and transmits the combustion-generated gas pressure to the crank pin via the connecting rod. The connecting rod, usually a steel or alloy forging (though sometimes aluminum), is fastened to the piston by means of a steel piston pin through the rod upper end. The piston pin is usually hollow to reduce its weight.

The oscillating motion of the connecting rod exerts an oscillating force on the cylinder walls via the piston skirt (the region below the piston rings). The piston skirt is usually shaped to provide appropriate thrust surfaces. The piston is fitted with rings that ride in grooves cut in the piston head to seal against gas leakage and control oil flow. The upper ring is the compression ring that is forced outward against the cylinder wall and downward onto the groove face. The lower rings scrape the surplus oil from the cylinder wall to reduce exposure to the hot burned gases, and return it to the crankcase. The crankcase must be ventilated to remove gases that blow by the piston rings, to prevent pressure buildup. The crankcase gases are recycled to the engine intake.

The cylinder head (or heads in V engines) seals off the cylinders and is made of aluminum or cast iron. It must be strong and rigid to distribute the gas forces acting on the head as uniformly as possible through the engine block. The cylinder head contains the spark plug (for an SI engine) or fuel injector (for a CI or direct-injection engine), and, in overhead valve engines, parts of the valve mechanism.

The valves shown in Fig. 1.4 are poppet valves, the valve type normally used in four-stroke engines. The engine shown has one intake and one exhaust valve: most modern engines have four valves per cylinder (two intake and two exhaust valves), or three valves (two intake and one exhaust). Valves are made from forged alloy steel; the cooling of the exhaust valve, which operates at up to about 700°C, may be enhanced by using a hollow stem partially filled with sodium, which through evaporation and condensation carries heat

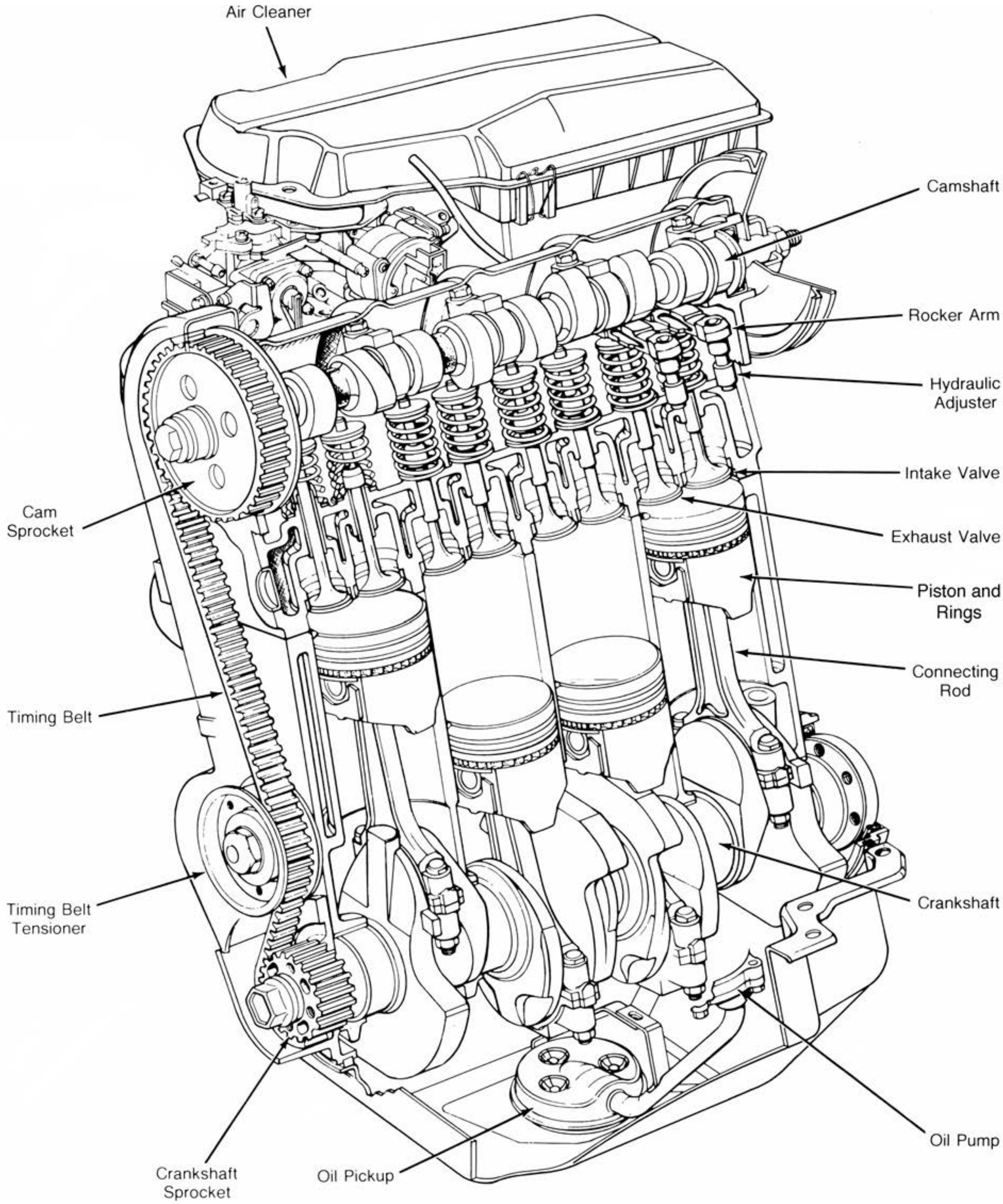


Figure 1.4 Cutaway drawing of 2.2-liter displacement four-cylinder spark-ignition engine. Bore 87.5 mm, stroke 92 mm, compression ratio 8.9.

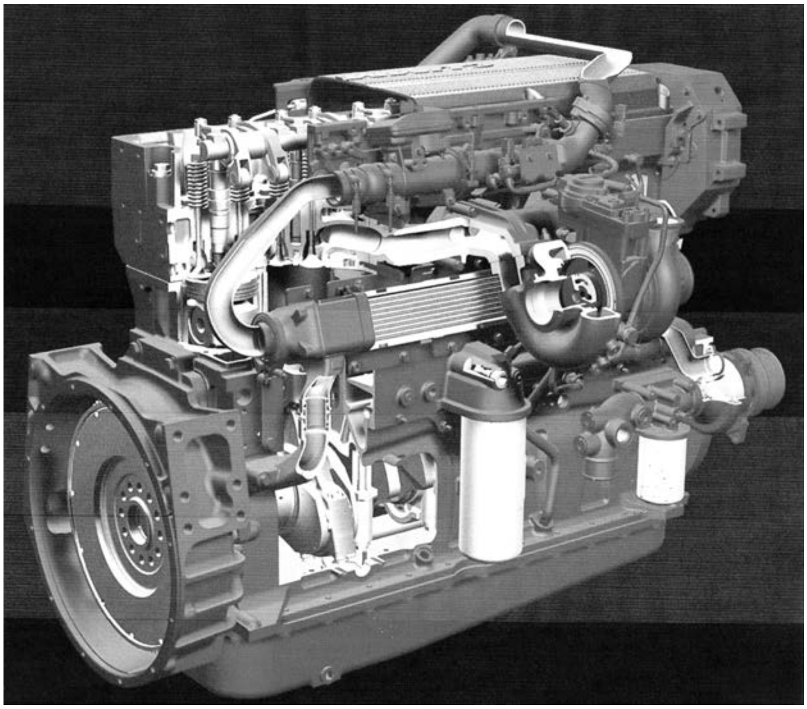


Figure 1.5 Cross-section drawing of a four-stroke cycle 6.7-liter in-line six-cylinder turbocharged diesel engine. Bore 107 mm, stroke 124 mm, compression ratio 17.3, maximum torque 1200 N·m at 1600 rev/min, maximum power 285 kW at 2800 rev/min. (*Courtesy Cummins Engines.*)

from the hot valve head to the cooler stem. Most modern SI engines have overhead valve locations (sometimes called valve-in-head or I-head configurations) as shown in Fig. 1.4. This geometry leads to a compact combustion chamber with minimum heat losses and flame travel time, and improves the breathing capacity. Older geometries such as the L head where valves are to one side of the cylinder are now only used in small low-cost engines.

The valve stem moves in a valve guide, which can be an integral part of the cylinder head (or engine block for L-head engines), or may be a separate unit pressed into the head (or block). The valve seats may be cut in the head or block metal (if cast iron) or hard steel inserts may be pressed into the head or block. A valve spring, attached to the valve stem with a spring washer and split keeper, holds the valve closed. A valve rotator turns the valves a few degrees on opening to wipe the valve seat, avoid local hot spots, and prevent deposits building up in the valve guide.

A camshaft made of cast iron or forged steel with one cam per valve (or pair of valves in four valves per cylinder engines) is used to open and close the valves. The cam surfaces are hardened to obtain adequate life. In four-stroke cycle engines, camshafts turn at one-half the crankshaft speed. Mechanical or hydraulic lifters or tappets slide in the block and ride or roll on the cam. Depending on valve and camshaft location, additional members are required to transmit the tappet motion to the valve stem; for example, in in-head valve engines with the camshaft at the side, a push rod and rocker arm are used. A trend in high-speed automotive engines is to mount the camshaft over the head with the cams acting either directly or through a pivoted follower on the valve. Also, variable control of valve opening and closing as a function of engine operating conditions, in its simplest form using a camshaft phasing device, is replacing fixed valve timing engine designs. Camshafts are gear, belt, or chain driven from the crankshaft.

An intake manifold (aluminum, cast iron, or plastic) and an exhaust manifold (generally of cast iron) complete the SI engine assembly. Other engine components specific to SI engines—fuel injectors, ignition systems—are described in more detail in the remaining sections in this chapter.

The four-stroke cycle diesel engine shown in Fig. 1.5 is an in-line six-cylinder design commonly used for large trucks. The engine is turbocharged to increase the amount of air that enters the cylinder each cycle. The turbocharger consists of a centrifugal compressor (which compresses the air prior to entry into the cylinder on the same shaft as the exhaust-gas-driven turbine that powers the compressor). In diesel engines, the fuel injectors are mounted in the cylinder head. Diesel fuel-injection systems are discussed in more detail in Sec. 1.8.

1.5 MULTICYLINDER ENGINES

Small engines are used in many applications: for example, lawn mowers, chain saws, in portable power generation, as outboard motorboat engines, and in motorcycles. These are often single-cylinder engines. In the above applications, simplicity and low cost in relation to the power generated are the most important characteristics; fuel consumption, engine vibration, high power to weight or volume ratio, and engine durability are usually less important. A single-cylinder engine gives only one power stroke per revolution (two-stroke cycle) or two revolutions (four-stroke cycle). Hence, the individual cycle torque pulses are widely spaced, and engine vibration and smoothness are significant issues.

Multicylinder configurations are invariably used in practice in all but the smallest engines. As rated power increases, the advantages of smaller cylinders in regard to bulk size, weight, improved engine performance, and engine balance and smoothness all point toward increasing the number of cylinders so the engine's total displaced volume is spread out amongst several smaller cylinders. The increased frequency of power strokes with smaller and increasing number of cylinders produces more frequent and smaller torque pulses, and thus smoother output. The forces in each component are smaller, so structural design requirements are reduced. Multicylinder engines can also achieve a much better state of balance than single-cylinder engines. A force must be applied to each piston to accelerate it during the first half of its travel from BC or TC. The piston then exerts a force on the crankshaft as it decelerates during the second part of the stroke. It is desirable to cancel these inertia forces through the choice of number and arrangement of cylinders to achieve a *primary* balance. Note, however, that the motion of the piston is more rapid during the upper half of its stroke than during the lower half (a consequence of the connecting rod and crank mechanism evident from Fig. 1.1; see also Sec. 2.2). The resulting inequality in acceleration and deceleration of pairs of pistons (one moving up and one moving down) produces corresponding differences in inertia forces generated. Certain combinations of cylinder number and arrangement balance out these *secondary* inertia force effects.

For a given engine displaced volume, the larger the number of cylinders, the higher the engine's maximum power. The reciprocating speed of an engine's pistons is limited by the airflow into each cylinder. Once the flow through the intake valve becomes sonic—reaches the speed of sound—higher piston speeds do not increase airflow. For a given engine displacement, increasing the number of cylinders, and thus reducing their size, raises the crankshaft rotational speed at which this sonic airflow limit is reached. Since engine power is proportional to the engine's rotational (crankshaft) speed, maximum performance is improved.

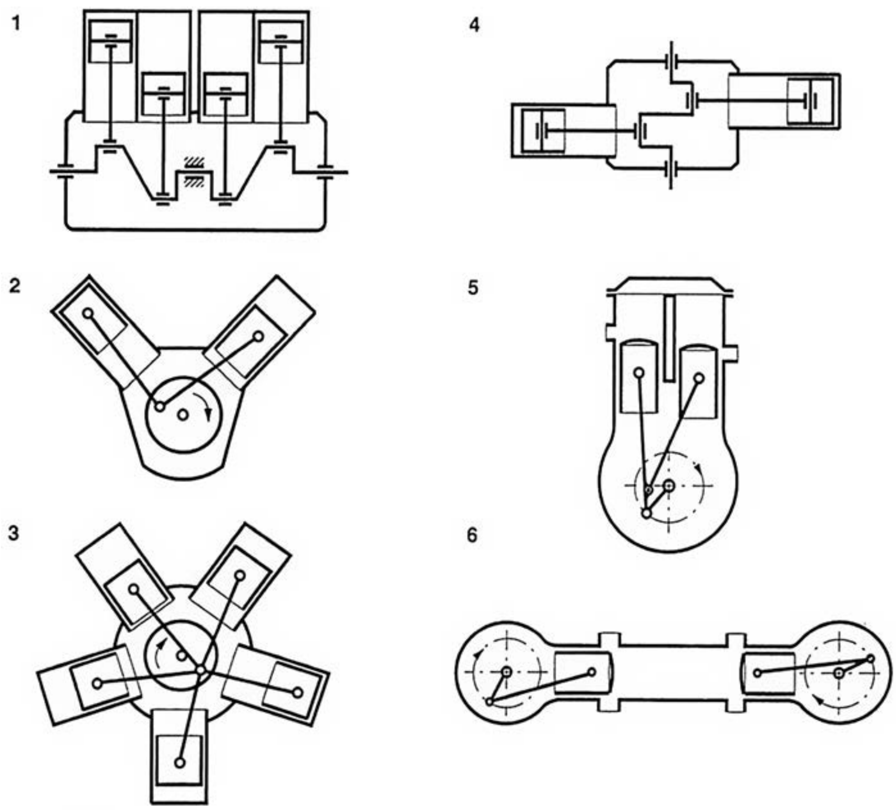


Figure 1.6 Multicylinder engine configurations: (1) In-line engine; (2) V-engine; (3) Radial engine; (4) Opposed-cylinder engine; (5) U-engine; (6) Opposed-piston engine.[13] (*Courtesy Robert Bosch GmbH and SAE.*)

Other operational issues are affected by cylinder size. The relative importance of heat losses from the in-cylinder gases depends on the relative importance of the combustion chamber surface area to its volume. The SI engine compression-ratio limiting phenomenon called knock is adversely affected by the flame travel distance (spark plug gap to farthest combustion chamber wall).

Common four-stroke multicylinder configurations are shown in Fig. 1.6.[13] These multicylinder configurations normally use equal crankshaft rotation firing intervals between cylinders. In *in-line engines*, the cylinders are arranged in a single plane. Three-, four-, five-, and six-cylinder in-line configurations are used. Four-cylinder in-line engines are the most common arrangement for automobile engines from 1.2 to about 2.5-liter displacement. An example of this in-line arrangement is shown in Fig. 1.4. It is compact—an important consideration for small passenger cars. It provides two torque pulses per revolution of the crankshaft, and primary inertia forces (though not secondary forces) are balanced. Six-cylinder in-line diesel engines are commonly used in the truck market with up to 12-liter displacement.

The vee (V) arrangement, with two banks of cylinders set at an angle to each other, provides a compact engine block and is used extensively for larger displacement automotive engines. Vee six, eight, ten, and twelve configurations are used. In a V-6 engine, the six cylinders are arranged in two banks of three, usually with a 60° angle between their axis.

Six cylinders are normally used in gasoline SI engines in the 2.4- to 3.6-liter displacement range. Six-cylinder engines provide smoother operation with three torque pulses per revolution. The in-line arrangement is fully balanced. However, it gives rise to crankshaft torsional vibration, and also makes even distribution of air to each cylinder more difficult. The V-6 arrangement is more compact than an in-line 6, and provides primary balance of the reciprocating components. With the V-engine, however, a rocking moment is imposed on the crankshaft due to the secondary inertia forces, which results in the engine being less well balanced than the in-line version. The V-8 arrangement, in sizes between 3.2 and 6 or more liters, is commonly used to provide compact, smooth, low-vibration, larger-displacement, SI engines, as are V-10 and V-12 designs.

The radial engine configuration, with cylinders arranged in one or more radial planes, as shown, was common in larger piston-driven aircraft engines. Opposed cylinder engine designs are occasionally used. As Fig. 1.6 indicates, the motion of pairs of pistons with this design is fully balanced. The U-cylinder configuration, where the pistons move in the same direction, and the opposed-piston configuration have been used in special purpose two-stroke engine concepts.

1.6 SPARK-IGNITION ENGINE OPERATION

In SI engines, the fuel must be vaporized and well mixed with the air inducted into the cylinder, prior to combustion. Historically, the fuel flow was metered with a carburetor or single-point fuel-injection system (Fig. 1.7*a*) upstream of the throttle, which controls the airflow. This approach has been superceded by intake-port fuel injection (Fig. 1.7*b*) where a pulsed liquid fuel spray is directed toward the intake valve. Injection of gasoline directly into each cylinder (Fig. 1.7*c*) is now in large-scale production. Moving the point of fuel injection closer to the cylinder enables better dynamic response during engine transients.

Figure 1.8 shows the layout of a modern SI engine management system. The airflow, fuel flow, exhaust gas characteristics, and engine operating state are all monitored and controlled as shown to provide the desired engine performance with good combustion characteristics, high efficiency, and low exhaust air pollutant emissions. The ratio of mass flow of air to mass flow of fuel must be held approximately constant at about 15 to ensure reliable

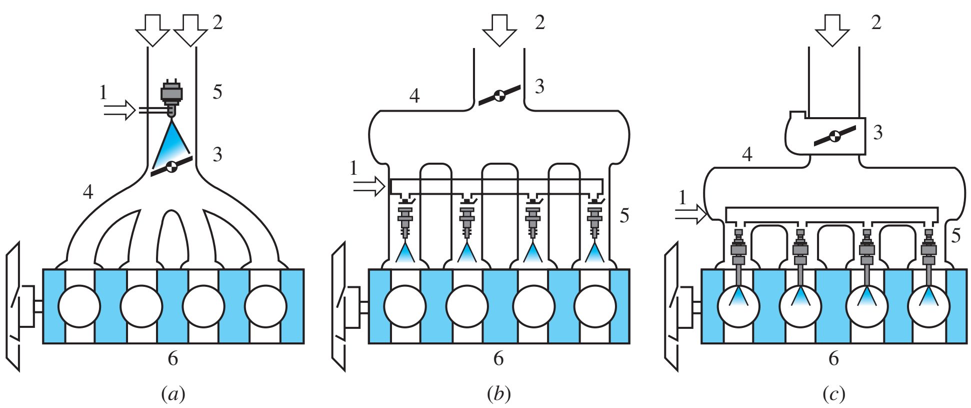


Figure 1.7 Different fuel-injection approaches for gasoline spark-ignition engines. (*a*) Single-point injection; (*b*) Multipoint port injection; (*c*) Direct in-cylinder injection. (1) Fuel supply; (2) Air supply; (3) Throttle valve; (4) Intake manifold; (5) Injectors; (6) Engine.[13] (*Courtesy Robert Bosch GmbH and SAE.*)

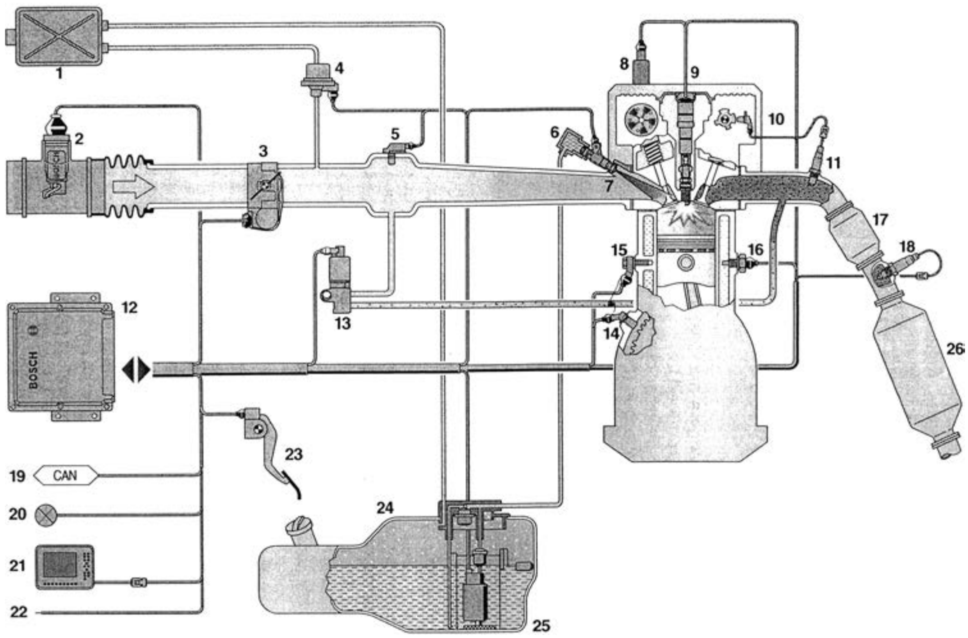


Figure 1.8 Schematic of modern port-injected engine management system (Bosch ME-Motronic system). (1) Carbon fuel-vapor absorbing canister; (2) Hot-film air-mass meter with integrated temperature sensor; (3) Throttle device; (4) Canister-purge valve; (5) Intake-manifold pressure sensor; (6) Fuel-distribution pipe; (7) Injector; (8) Actuators and sensors for variable valve timing; (9) Ignition coil with attached spark plug; (10) Camshaft phase sensor; (11) Lambda oxygen sensor upstream of primary catalytic converter; (12) Engine control unit; (13) Exhaust-gas recirculation valve; (14) Speed sensor; (15) Knock sensor; (16) Engine-temperature sensor; (17) Primary three-way catalytic converter; (18) Lambda oxygen sensor downstream of primary catalytic converter: (19) CAN interface; (20) Fault lamp; (21) Diagnosis interface; (22) Interface to immobilizer control unit; (23) Accelerator-pedal module with pedal-travel sensor; (24) Fuel tank; (25) In-tank unit with electric fuel pump, fuel filter, and fuel-pressure regulator; (26) Main three-way catalytic converter.[13] (*Courtesy Robert Bosch GmbH and SAE.*)

combustion and facilitate exhaust emissions control. The appropriate fuel flow is determined for the engine airflow in the following manner. The airflow into the intake system is measured with an air mass-flow meter. A throttle valve or plate, which can be opened or closed, controls the airflow. The appropriate amount of fuel required per cylinder per cycle to generate the desired engine output is then determined by the engine control unit. In naturally-aspirated engines, the intake airflow is reduced by throttling to below atmospheric pressure by reducing the flow area when the power required (at any engine speed) is below the maximum, which is obtained when the throttle is wide open.

The sequence of events that take place inside the engine cylinder is illustrated in Fig. 1.9. Several variables are plotted against crank angle through the entire four-stroke SI engine cycle. Crank angle is a useful independent variable because the various engine processes occupy almost constant crank angle intervals over a wide range of engine-operating conditions. The figure shows the valve opening and closing angles, and volume relationship, for a typical fixed valve-timing automotive SI engine. To maintain high mixture flows at high engine speeds (and thus high power outputs) the inlet valve, which opens before TC, closes substantially after BC. During intake, the inducted fuel and air mix in the cylinder

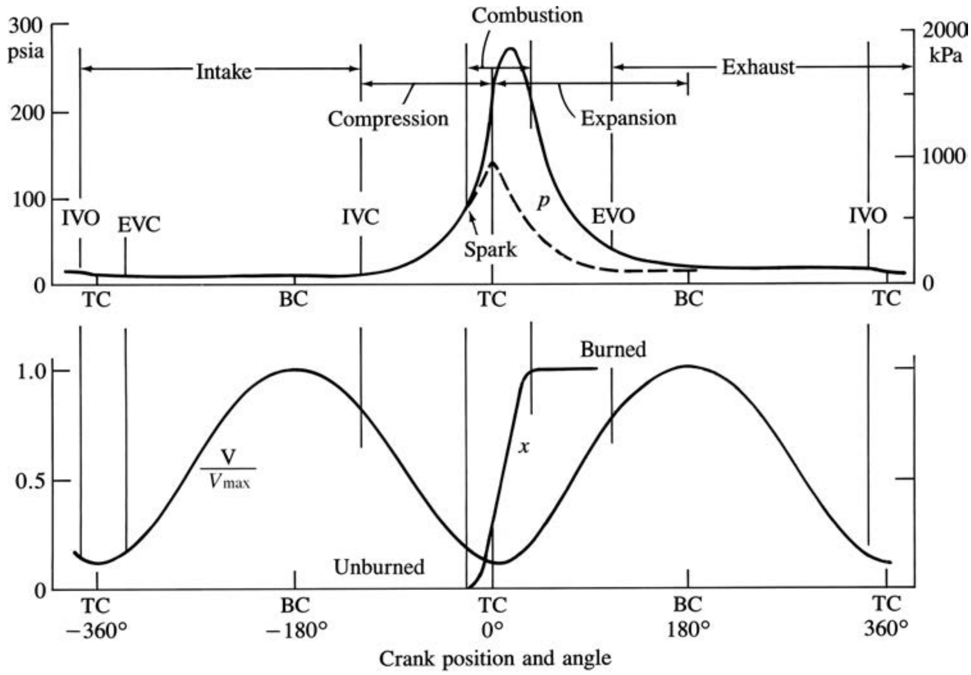


Figure 1.9 Sequence of events in four-stroke spark-ignition engine-operating cycle. Cylinder pressure p (solid line, firing cycle; dashed line, motored cycle), cylinder volume V/V_{max}, and mass fraction burned x_b are plotted against crank angle.

with the *residual* burned gases remaining from the previous cycle. After the intake valve closes, the cylinder contents are compressed to above atmospheric pressure and temperature as the cylinder volume is reduced. Some heat transfer between the in-cylinder gases and the piston, cylinder head, and cylinder walls occur—first a heating of the gases, then a cooling, but the effect on unburned gas properties is modest.

Between about 10 and 40 crank angle degrees before TC, an electrical discharge across the spark plug starts the combustion process. Before the desired ignition point, the ignition driver switches a current to the primary circuit of the ignition coil. At the ignition point, the primary winding is interrupted, generating in the secondary ignition coil winding that is connected to the spark plug, a high voltage across the plug electrodes as the magnetic field collapses. This switching is done electronically. A flame develops from the spark discharge, propagates through the mixture of air, fuel, and residual gas in the cylinder, and extinguishes at the combustion chamber walls. The duration of this burning process varies with engine design and operation, but is typically 40 to 60 crank angle degrees, as shown in Fig. 1.9. As fuel-air mixture burns in the flame, the cylinder pressure (solid line in Fig. 1.9) rises above the level due to compression alone (dashed line). This latter curve—called the *motored* cylinder pressure—is the pressure trace obtained from a motored or nonfiring engine.[f] Note that due to differences in the flow pattern and mixture composition between cylinders and within each cylinder, cycle-by-cycle, the development of each combustion process differs somewhat. As a result, the shape of the pressure versus crank angle curve in each cylinder, and cycle-by-cycle, is not exactly the same.

[f] In practice, the intake and compression processes of a firing engine and a motored engine are not the same due to the presence of burned gases from the previous cycle under firing conditions.

There is an optimum spark timing which, for a given mass of fuel, air, and residual inside the cylinder, gives maximum torque. More advanced (earlier) timing or retarded (later) timing than this optimum gives lower output. Called *maximum brake-torque* (MBT) timing,[g] this optimum timing is an empirical compromise between starting combustion too early in the compression stroke (when the work transfer is *to* the cylinder gases) and completing combustion too late in the expansion stroke (and so lowering peak expansion stroke pressures).

About two-thirds of the way through the expansion stroke, the exhaust valve starts to open. The cylinder pressure is significantly higher than the exhaust manifold pressure and a *blowdown* process occurs. The burned gases flow through the valve into the exhaust port and manifold until the cylinder pressure and exhaust pressure equilibrate. The duration of this process depends on the pressure level in the cylinder. The piston then *displaces* most of the remaining burned gases from the cylinder into the manifold during the exhaust stroke. The exhaust valve opens before the end of the expansion stroke to ensure that the blowdown process does not last too far into the exhaust stroke when the piston travels upwards. The actual timing is a compromise that balances reduced work transfer *to the piston* before BC against reduced work transfer *to the cylinder contents* after BC.

The exhaust valve remains open until just after TC; the intake opens just before TC. The valves are opened and closed slowly to avoid noise and excessive cam wear. To ensure the valves are fully open when piston velocities are at their highest, the valve open periods usually overlap somewhat. If the intake flow is throttled to below exhaust manifold pressure, then backflow of burned gases from the cylinder into the intake manifold occurs when the intake valve is first opened. During the valve overlap period, backflow of burned gas from the exhaust port into the cylinder occurs.

With variable valve control, the trade-offs that fixed valve timing requires can be relaxed. The simplest approach varies intake and exhaust valve timing by rotating the camshafts to change their phasing relative to the crankshaft. More complex systems vary valve lift as well as varying the valve opening and closing angles. Variable valve control is attractive because it improves maximum engine power (at high speed) and maximum torque at lower speeds, and can improve part-load engine efficiency. It can also be used to control the mass of burned residual gas, and fresh air, trapped in the engine cylinder.

1.7 DIFFERENT TYPES OF FOUR-STROKE SI ENGINES

A variety of SI engines are used in practice, depending on the application. Small SI engines are used in many applications: in the home (e.g., lawn mowers, chain saws), in portable power generation, as outboard motorboat engines, and in motorcycles. These are often single-cylinder engines producing a few kW of power. In the above applications, light weight, small bulk, and low cost in relation to the power generated are the most important characteristics; fuel consumption, engine vibration, and engine durability are less important. A single-cylinder engine gives only one power stroke per crank revolution (two-stroke cycle) or two revolutions (four-stroke cycle). Hence, the torque pulses are widely spaced, and engine vibration and smoothness are significant problems.

Multicylinder engines are invariably used in automotive practice. As rated power increases, the advantages of smaller cylinders in regard to size, weight, power density, improved engine balance, and smoothness point toward increasing the number of cylinders per engine: see Sec. 1.5. Multicylinder SI engines (in the power range 25 to 400 kW) are used

[g]MBT timing was traditionally defined as the minimum spark advance for best torque. Since the torque first increases and then decreases as spark timing is advanced, the definition used here is more precise.

in cars, light trucks, vans, light-duty commercial vehicles, and in stationary applications to produce mechanical and electrical power. Many of these markets are shared with diesel (CI) engines. Low engine emissions and high operating efficiency are important, especially in these transportation applications. Precise control of fuel and airflow is critical to achieving these objectives. What used to be the dominant fuel metering device—the carburetor—has been superceded by electrically controlled fuel injection into each intake port. Now, injection of the gasoline directly into each engine cylinder is coming into production (Fig. 1.7). Each of these technology steps improves control of the amount of fuel entering each cylinder per cycle, and thus the dynamic response of the engine to changes in load (engine output) and speed.

The work transfer per cycle to each piston depends on the amount of fuel burned per cylinder per cycle, which depends on the amount of fresh air inducted each cycle. Variable valve control over the engine's speed range can be used to increase the mass of air inducted into each cylinder in four-stroke SI engines (especially at low and high speed), and thus increase the wide-open-throttle torque and power. Engine output from a given displacement engine can be increased by boosting—increasing the density of the air supplied to the engine intake by compressing atmospheric air. Thus, compressing the air prior to entry into the cylinder with a supercharger or a turbocharger increases the output from a given displacement engine.

Examples of various types of SI engines in practical use follow to provide the context for reviewing critical engine processes, a primary objective of this text.

1.7.1 Spark-Ignition Engines with Port Fuel Injection

Spark-ignition (SI) engines have traditionally been operated with a premixed fuel vapor/air mixture inside the cylinder, prepared by feeding liquid gasoline into the engine intake. Carburetors were used to meter the fuel flow in proportion to the airflow. This technology has largely been replaced by port fuel injection (see Fig. 1.8) where an injector in each cylinder's intake port or manifold injects a pulsed fuel spray toward the intake valve, once per cycle. The hot valve surface and warm intake port (once the engine has warmed-up) promote rapid evaporation of the liquid fuel, and the airflow through the port(s) past the intake valve(s) and into the cylinder, coupled with the in-cylinder flow and mixing with the hot residual gas, produces a nearly homogeneous mixture by the time combustion starts. Here, we show some examples of SI engines with this method of mixture preparation.

Figure 1.10 shows a small single-cylinder air-cooled SI engine with a displaced volume of 149 cm^3 and power output of 2.8 kW (3.9 hp). The objective of such simple construction SI engines is to produce modest power levels at low cost. A primary benefit of air-cooled, as compared to the water-cooled, engines is lower engine weight. The fins on the cylinder block and head are necessary to increase the external heat-transfer surface area to achieve the required heat rejection. In small engines, such as in Fig. 1.10, natural convection promotes adequate airflow around the outside of the engine. In larger engines, an air blower provides forced air convection over the block. The blower is driven off the driveshaft.

Figure 1.11 shows a turbocharged automobile engine that incorporates many of the features now used to improve engine performance and efficiency. The in-line arrangement with four cylinders provides a compact block, and when turbocharged, increases the power per unit engine displaced volume significantly. This engine features all-aluminum construction, four valves per cylinder, dual overhead camshafts, friction reducing roller finger followers in the valve train, variable phasing on each cam to control the relative phasing of intake and exhaust valves, and piston-cooling oil jets to control piston temperatures in this high-performance engine. Other performance enhancing features now being designed

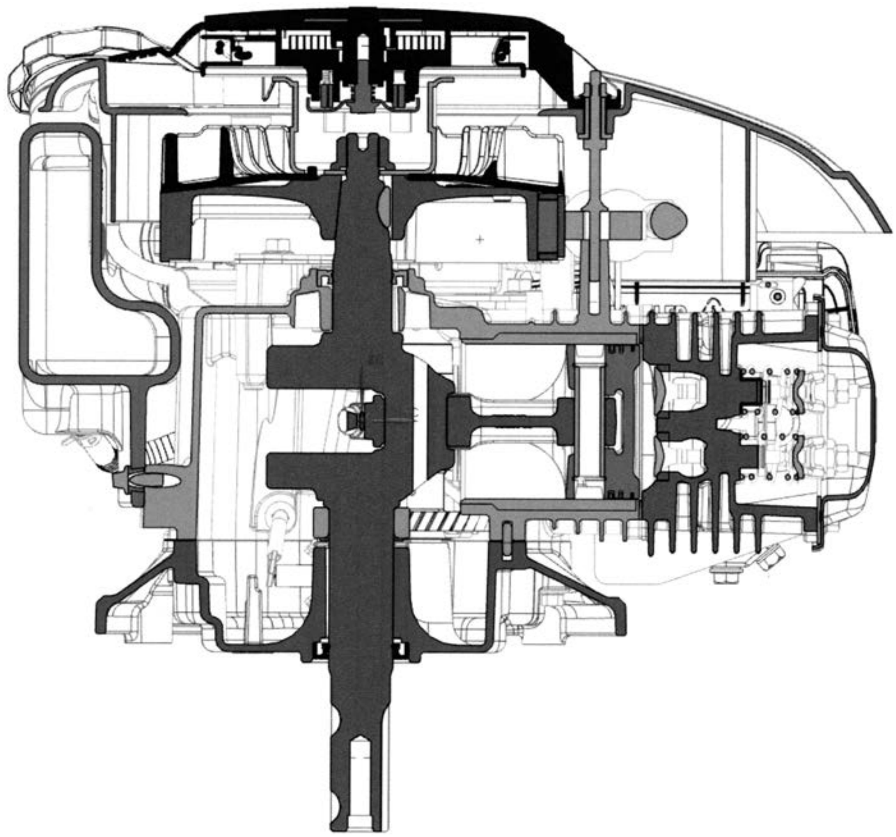


Figure 1.10 Cutaway drawing of single-cylinder air-cooled spark-ignition engine. Displacement 149 cm^3, bore 65 mm, stroke 45 mm, compression ratio 9.2, maximum power 2.8 kW at 3000 rev/min. (*Courtesy Kohler Co.*)

into such automobile engines are cylinder cut out (or displacement on demand) where, for example, in a V-8 engine, at the lighter loads, the valves in half the cylinders are deactivated so only four cylinders provide torque. This reduces the pumping work over the exhaust and intake strokes and thereby improves engine fuel consumption.

Variable valve control improves engine performance, efficiency, and emissions (see Sec. 6.3.3). The simpler systems used vary the relative phasing of the intake and exhaust valve opening and closing by rotating the camshafts relative to the crankshaft. Valve lift profiles and open duration remain fixed. This is the approach used in the engine shown in Fig. 1.11. (The cam-phasing system is apparent upper center of the engine drawing.) More sophisticated approaches vary valve timing, lift profile, and open duration (e.g., BMW's Valvetronic system[15]). This technology can eliminate the need for throttle valves by accurately controlling the cylinder charging process by intake valve control.

1.7.2 SI Engines for Hybrid Electric Vehicles

The use of internal combustion engines in automotive hybrid propulsion systems is prompting additional SI engine developments. In such a hybrid system, an internal combustion engine, a generator, battery, and electric motor are combined. Figure 1.12 shows three categories of hybrid systems: a *parallel* hybrid, a *series* hybrid, and a *power split* hybrid. In the

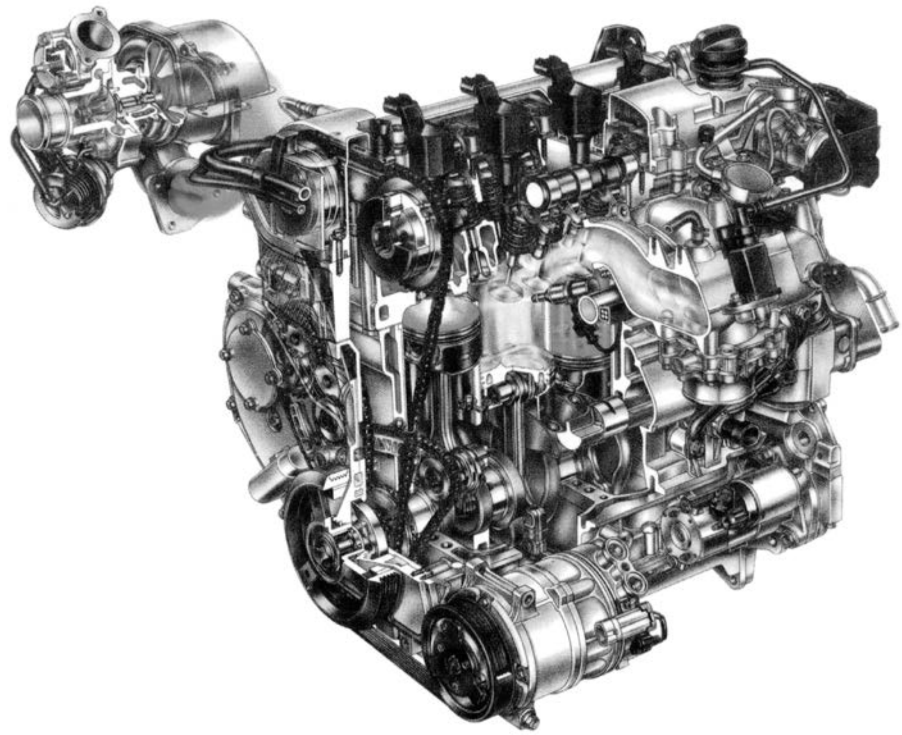


Figure 1.11 Cutaway drawing of General Motors four-cylinder turbocharged DI Ecotec gasoline spark-ignition engine. Displacement 2.0 liters, bore 86 mm, stroke 86 mm, compression ratio 9.2, maximum power 187 kW (250 hp) at 5300 rev/min, maximum torque 353 N·m (260 lb·ft) at 2000 rev/min.[14] (*Courtesy General Motors Corporation.*)

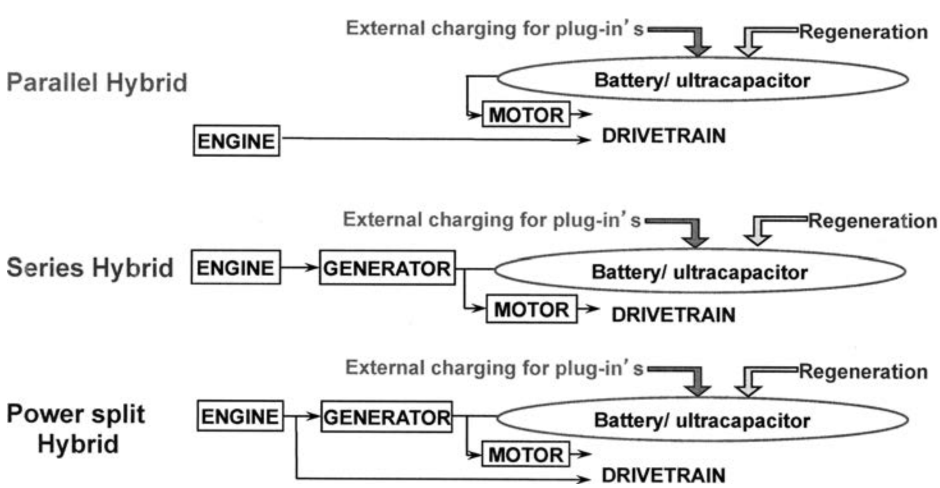


Figure 1.12 Diagrams of parallel, series, and power split hybrid electric vehicle propulsion systems.

parallel approach, the engine can drive the wheels directly, the battery can drive via the electric motor, or both can be combined to drive the wheels to obtain a high overall propulsion system efficiency at all loads and speeds.

In the series approach, the electric motor drives the vehicle's wheels. The engine can drive through the generator and motor, or recharge the battery via the generator. Since the vehicle is propelled solely by electrical energy, the engine is not coupled to the wheels. Thus its operating conditions are not dependent on the vehicle's operation so it can be operated in its higher efficiency modes. In the power-split system, a planetary gear set is used to transmit power from the engine. This arrangement allows both parallel and series-type operation to be combined. Power from the engine can flow directly to the wheels via the ring of the planetary gear system. Engine power can also flow through the generator, producing electrical power that can drive the wheels through the electric motor.

These hybrid propulsion systems provide increased vehicle drive efficiency relative to direct internal combustion engine drive for three basic reasons. First, *regenerative braking*—applying a braking torque by connecting the generator to the vehicle's wheels is then used to recharge the battery—converts a substantial fraction of the vehicle's kinetic energy as the vehicle slows down to store electrical energy. Second, when the engine is being used, it can operate much of the time at a higher efficiency than would be the case with a stand-alone engine vehicle propulsion system. Third, the battery electric drive mode allows the engine to be shut down when the vehicle is decelerating or idling.

Figure 1.13 shows an SI engine designed specifically for this application. The engine employs a modified version of the four-stroke cycle called the *Atkinson cycle*, where the volume ratio used for expansion is higher than the volume ratio for compression. The engine shown has a displaced volume of 1.5 liters, a geometric (TC to BC) compression/expansion ratio of 13:1, and uses variable valve timing with late intake valve closing during compression and late exhaust valve opening during expansion to achieve a higher

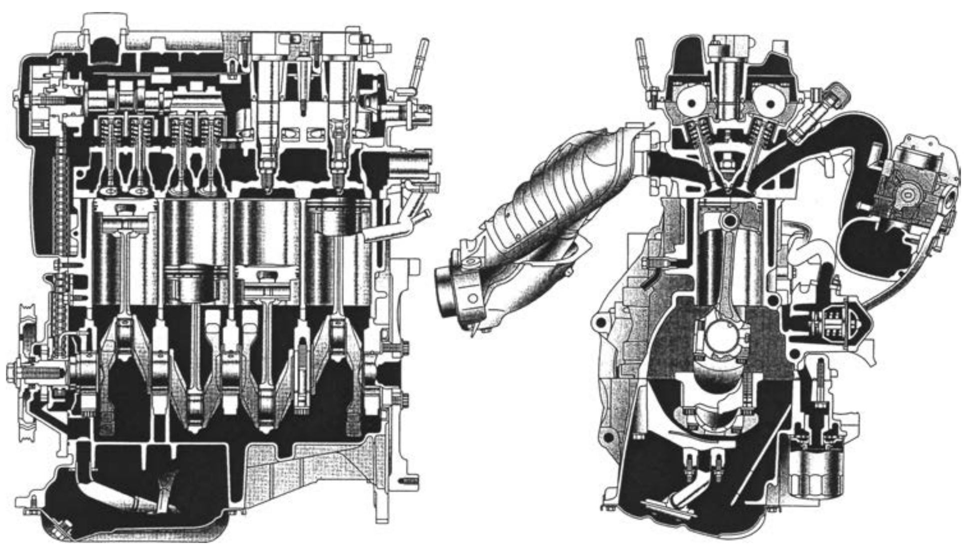


Figure 1.13 Four-cylinder Toyota spark-ignition engine designed for a hybrid electric automobile propulsion system.[16] This 1.5-liter (bore = 75 mm, stroke = 84.7 mm), four valves per cylinder, variable valve timing engine uses the Atkinson cycle with a geometric compression/expansion ratio of 13:1. Maximum power is 57 kW (76 hp) at 5000 rev/min. Valve timings are: intake opening 18 to –15° BTC, closing 72 to 105° ABC; exhaust opening 34° BBC, closing 2° ATC.

effective expansion than compression. This increases engine efficiency. Maximum engine speed is held to 5000 rev/min to minimize the pumping penalties of this Atkinson cycle approach.

An alternative to this hybrid electric vehicle (HEV) system, which overall is powered solely by a fuel such as gasoline, is the plug-in hybrid (PHEV) system. Here a larger battery, with some 10 to 30 mile (15 to 50 km) all electric driving range rather than the electric range of a few miles of the HEV system, is used that can be recharged from the electrical grid. Thus the PHEV can be driven with electricity or with a hydrocarbon fuel similarly to an HEV. There is an important but different role for SI engines (and potentially diesels) to play as a key component of these more efficient hybrid systems: the engine preserves the driving flexibility that vehicles require, as the electrification of propulsion systems continues to evolve.

1.7.3 Boosted SI Engines

The work transfer to each piston per cycle that can be obtained from a given displacement engine determines the amount of torque the engine can deliver. This work transfer depends on the amount of fuel that can be burned in each cylinder each cycle. This depends on the amount of fresh air that is inducted into each cylinder each cycle. Increasing the air density prior to its entry into each cylinder thus increases the maximum torque that an engine of a given displacement can deliver. This can be done with a supercharger, a compressor mechanically driven by the engine. More often it is done with a turbocharger, a compressor-turbine combination, which uses the energy available in the engine exhaust stream to provide via the turbine the power required to compress the intake air.

Figure 1.14 shows a cutaway drawing of a turbocharged automobile SI engine, which illustrates how the turbocharger connects with the engine's cylinders. The airflow passes through an air filter (1) into a centrifugal compressor (2) where the radially outward flowing air is compressed by the rotating varies. Next the air flows through an intercooler (3) to reduce the compressed air temperature (further increasing its density), through the intake manifold (4) into the intake port where the fuel is injected, past the intake valve (5), and into the cylinder (6). When the exhaust valve (7) opens, the hot and higher-than-atmospheric pressure exhaust gas flows through the valve and exhaust manifold (8) into the turbine (9). The exhaust gas is directed radially inward and circumferentially at high velocity by vanes (nozzles) onto the turbine wheel's blades where some of the exhaust gas energy is extracted as work or power. The turbine drives the compressor. A wastegate (valve) just upstream of the turbine bypasses some of the exhaust gas flow when necessary to prevent the boost pressure becoming too high. The wastegate linkage (11) is controlled by a boost pressure regulator (12). Figure 1.15 shows a cutaway drawing of a small automotive turbocharger. The arrangement of the compressor and turbine rotors connected via the central shaft and of the turbine and compressor flow passages are evident.

Increasing the intake air density, through boosting, increases the mass of air trapped within the cylinder, the mass of fuel burned, and thus the torque a given size engine can protrude. Torque increases of more than a factor of two can be realized. Turbocharging of SI engines is made difficult by the SI engine's knock constraint. The onset of knock (the rapid spontaneous ignition of a fraction of the in-cylinder fuel-air mixture) during the latter part of combustion) depends on the maximum mixture temperature and pressure reached inside the engine cylinder, and boosting raises both these variables. Special measures such as reducing the compression ratio, higher octane—better knock resisting—fuels have to be used to control knock. Direct fuel injection into the cylinder (see following section), with its charge-cooling effect, eases this problem.

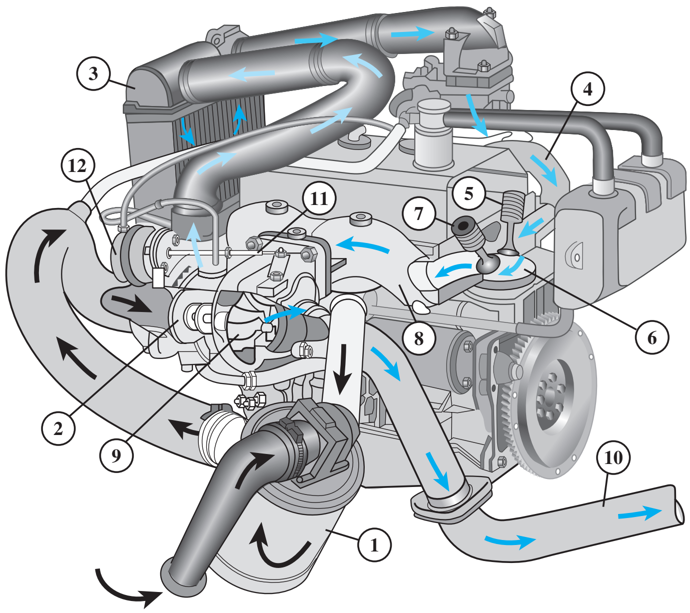


Figure 1.14 Drawing of turbocharger system connected to four-cylinder automobile spark-ignition engine. See text for details. (*Courtesy Regie Nationale des Usines.*)

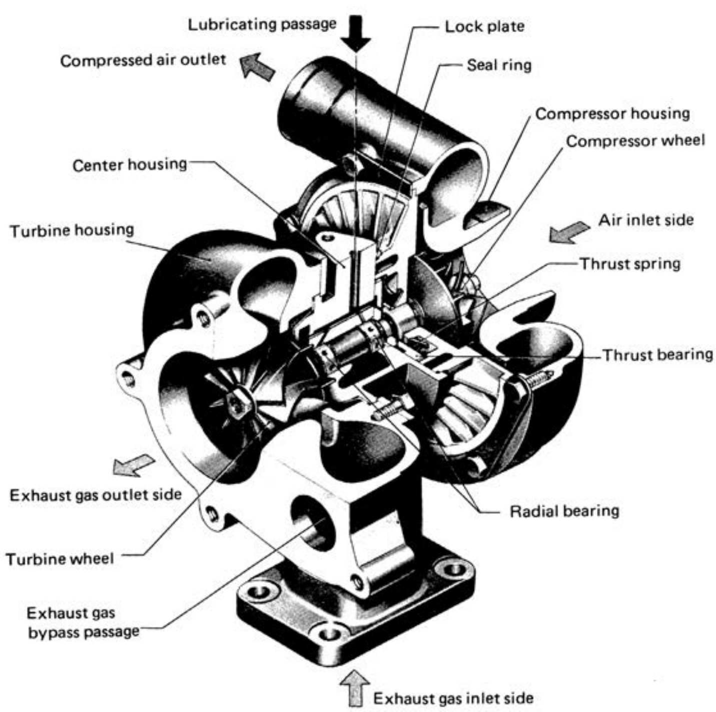


Figure 1.15 Cutaway view of small automotive SI engine turbocharger. (*Courtesy Nissan Motor Co. Ltd.*)

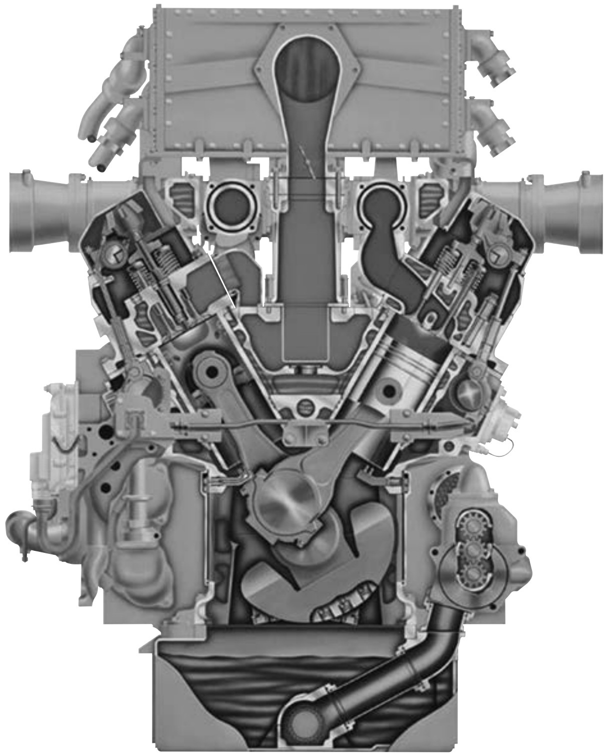


Figure 1.16 Large natural-gas-fueled boosted SI engine used in electric power generation. Bore 170 mm, stroke 190 mm, displaced volume per cylinder 4.3 liters, compression ratio 12:1, power (eight cylinders) 965 kW at 1500 rev/min. (*Courtesy Caterpillar, Inc.*)

A different type of boosted SI engine is large natural-gas fueled engines. These are used in electric power generation, propulsion, and marine applications. An example is shown in Fig. 1.16. It uses an encapsulated spark plug with orifices to improve ignition.

1.7.4 Direct-Injection SI Engines

Since the 1920s, attempts have been made to develop internal combustion engines that combine the best features of the SI engine and the diesel. By injecting the gasoline fuel directly into each cylinder of the engine, better control of the fuel's behavior can be achieved, improving the engine's dynamic performance, permitting use of higher compression ratios, and reducing the losses resulting from throttling the airflow in the standard port-injected SI engine. Diesels are more efficient because they operate close to the optimum compression ratio (14 to 18), operate fuel lean (with excess air), and control engine output by varying the fuel flow rate while leaving the airflow unthrottled. Historically, direct-injection SI engines have often been called *stratified-charge engines* since to realize all these benefits, the mixing process between the evaporating fuel jet and the air in the cylinder must produce a "stratified" or nonuniform fuel-air mixture, with an easily ignitable composition at the spark plug at the time of ignition, and with excess air surrounding the fuel-containing spray.

Over the years, many different types of stratified-charge engine have been proposed; some are now being used in practice.[17] The operating principles of three of these early

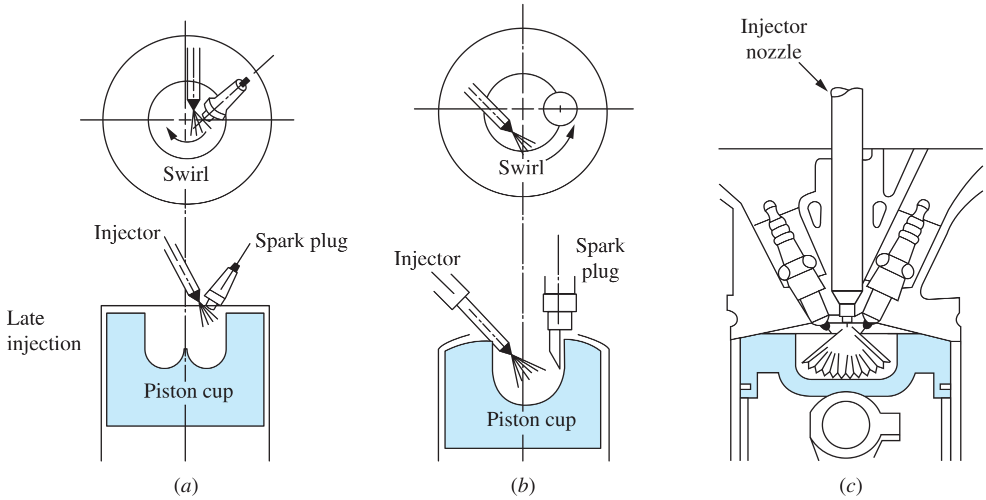


Figure 1.17 Three historical stratified-charge engines that were developed for production: (*a*) Texaco Controlled Combustion System (TCCS);[18] (*b*) M.A.N.-FM Combustion System;[19] (*c*) Ford PROCO Combustion System.[20]

designs are shown in Fig. 1.17. The combustion chambers are bowl-in-piston designs, and a high degree of air swirl (rotation about the cylinder axis) is created during intake and enhanced in the piston bowl during compression to achieve rapid fuel-air mixing. With the Texaco[18] and MAN[19] systems (Figs. 1.17*a* and *b*), fuel is injected into the cylinder in tangentially into the bowl during the latter stages of compression. A long-duration spark discharge ignites the fuel-air jet as it passes the spark plug; the flame spreads downstream, and consumes the fuel-air mixture. Figure 1.17*c* shows the Ford PROCO system[20] with its centrally located injector and hollow cone spray injected earlier in the compression stroke to get more complete fuel vapor/air mixing, so that high air utilization could be achieved to obtain high outputs.

Modern direct-injection SI engines are often divided in so-called spray-guided, wall-guided, and air-guided categories: see Fig. 1.18. This classification is based on the primary mechanism used to control the development of the fuel spray. In practice, mixture stratification is achieved through a combination of these mechanisms. The Texaco TCCS system in Fig. 1.17*a* and the PROCO system in Fig. 1.17*c* are examples of the former. The MAN system, Fig. 1.17*b*, is primarily wall guided (with air swirl also playing an important role). The Texaco system, Fig. 1.17*a*, is also air guided, with high air swirl generated during intake and augmented by the bowl-in-piston combustion chamber during compression. Many systems with significantly different geometric details are now being developed and employed in production:[17] see Sec. 7.7.2. Generally, spray-guided approaches require a closer spacing between the injector and spark plug electrode location, as shown in Fig. 1.18, to limit the dispersion of the fuel spray and provide substantial mixture stratification. Wider spacing allows more time for fuel-air mixing, produces a more uniform composition spray, but then requires a specific combination of charge motion and wall guiding to achieve the desired spray behavior, and combustion, and emissions characteristics.

Figure 1.19 shows a production example of a direct-injected (DI) wall-guided system. This Mitsubishi gasoline DI engine used a spherically shaped cavity in the piston crown and an upright intake port to generate a reverse tumbling airflow in the cylinder during intake,

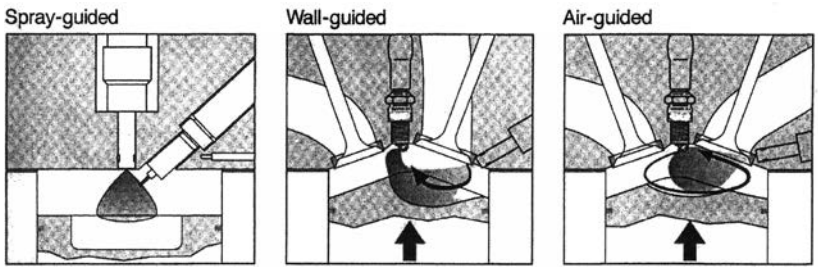


Figure 1.18 Illustrations of spray-guided, wall-guided, and air-guided direct-injection SI combustion systems.[17]

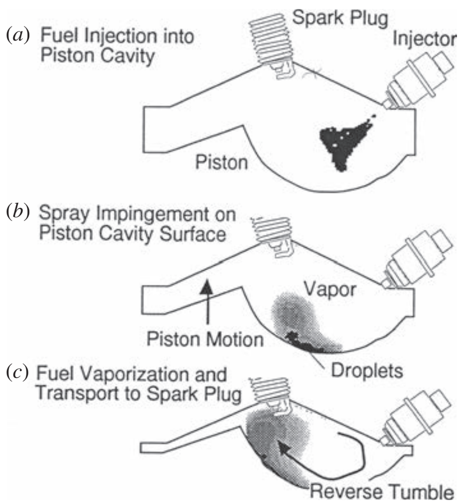


Figure 1.19 Mitsubishi gasoline direct-injection SI engine design. It uses a wide spacing between injector and spark plug; the spray is guided by the hemispherical piston cavity, and the reverse tumble produced by the upright intake port. In this 1.83-liter, four-cylinder, 12:1 compression ratio engine, the bore is 81 mm and the stroke is 89 mm. The fuel system uses an electromagnetic-controlled high pressure (5 MPa) swirl injector.[21]

to “guide” the developing spray toward the spark plug in the center of the cylinder head. Figure 1.20 shows a Toyota direct-injection engine design that uses a fan-shaped fuel spray directed into a shell-shaped bowl in the piston crown to provide rapid air-fuel mixing and fuel vaporization, and by the in-cylinder flow set-up by the straight intake port, to guide the spray so it reaches the spark plug location at the appropriate point in the cycle. To achieve high engine outputs, both these concepts transition from *late injection* (i.e., injection during the latter half of the compression stroke) when stratified operation is desired, to *early injection* (injection during the intake stroke) when essentially complete mixing of the injected fuel with *all* the air in the cylinder is required. This latter mode is called *homogeneous-charge* operation, as distinct from stratified operation.

Homogeneous-charge operation at all engine loads and speeds is a viable direct-injection SI engine approach, and is used in production engines. While the efficiency benefit of stratified operation with excess air (which the diesel enjoys) is lost, the in-cylinder charge cooling due to liquid fuel vaporization that increases the amount of air inducted and reduces the propensity of the engine to knock, and the more accurate control of fuel flow during engine transients, are retained. Homogeneous direct-injection engine concepts

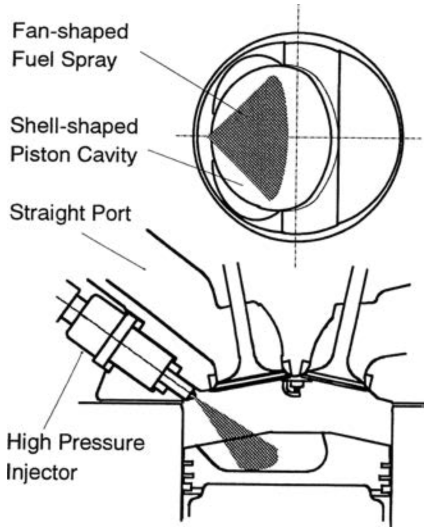


Figure 1.20 Toyota gasoline direct-injection SI engine design uses a wide-angle fan-shaped fine-atomization fuel spray injected into a shell-shaped piston cavity to achieve a stratified mixture with late injection, and homogeneous mixture with early (during intake stroke) injection.[22] This concept is also used in Toyota's homogeneous-charge direct-injection engines.

thus can increase compression ratio and efficiency, increase maximum power, and benefit from the effective emission-control technology that has been developed for port-injected SI engines.

Most production designs of direct-injection engines have used the four-stroke cycle. Direct-injection is, however, especially helpful in controlling fuel carry through in two-stroke cycle engines (see Sec. 1.10). Direct fuel is especially attractive with turbocharged engines to increase their power density. The charge cooling, which evaporation of the in-cylinder fuel spray produces, reduces the engine's propensity to knock.

1.7.5 Prechamber SI Engines

An alternative to these open-chamber SI engines described above is a prechamber engine concept, which has been mass produced. It uses a small *prechamber* fed during intake with an auxiliary fuel system to obtain an easily ignitable mixture around the spark plug. This concept, first proposed by Ricardo in the 1920s and extensively developed in the Soviet Union and Japan, is often called a *jet-ignition* or *torch-ignition* stratified-charge engine. Its operating principles are illustrated in Fig. 1.21, which shows a three-valve carbureted version of the concept.[23] A separate carburetor and intake manifold feed a fuel-rich mixture (which contains fuel beyond the amount that can be burned with the available air) through a separate small intake valve into the prechamber that contains the spark plug. At the same time, a lean mixture (which contains excess air beyond that required to burn the fuel completely) is fed to the main combustion chamber through the normal intake manifold. During intake, the rich prechamber flow fully purges the prechamber volume. After intake valve closing, lean mixture from the main chamber is compressed into the prechamber bringing the mixture at the spark plug to an easily ignitable, slightly rich composition. After combustion starts in the prechamber, rich burning mixture issues as a jet (or series of jets) through one or more orifices into the main chamber, entraining and igniting the lean main chamber charge. This engine is really a jet-ignition concept whose primary function is to extend the operating limit of conventionally ignited SI engines to mixtures leaner than could normally

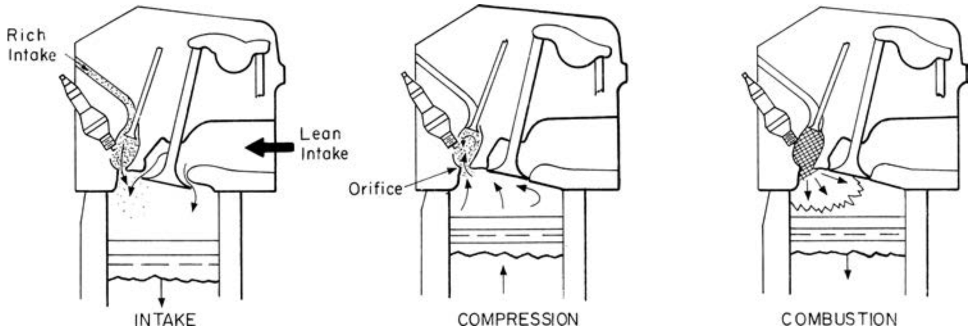


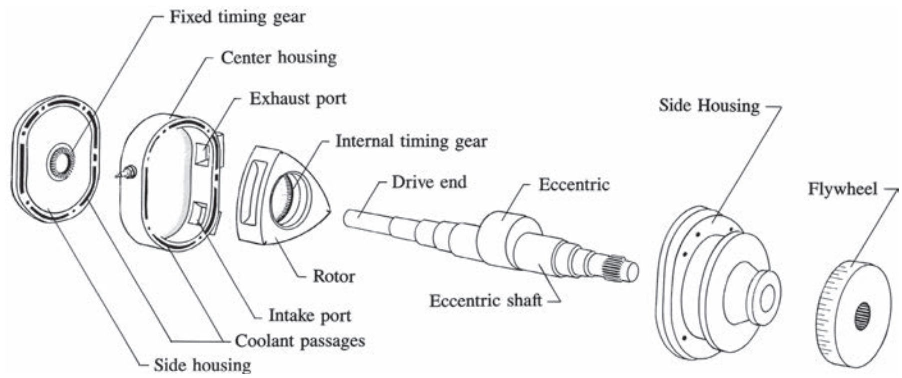
Figure 1.21 Schematic of three-valve torch-ignition stratified-charge spark-ignition engine.[23]

be burned. This approach has been used in large natural gas engines to provide rapid initiation of the combustion process.

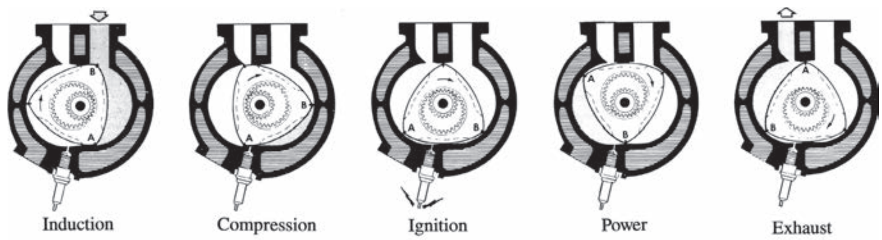
1.7.6 Rotary Engines

The reciprocating engine geometry discussed so far dominates the practical world of internal combustion engines. However, motivated by the fact that engine power is delivered through a rotating drive shaft, over the years many rotary engine designs have been proposed.[5] One of these, the Wankel rotary engine[6, 7] has and continues to be used in limited production. Its attractive features are its compactness and higher engine speed (which result in high power/weight and power/volume ratios), and its inherent balance and smoothness. These benefits are, however, offset by its higher heat transfer, and the engine's gas sealing and leakage problems.

Figure 1.22 shows the major mechanical parts of a simple single-rotor Wankel engine and illustrates its geometry and operation. There are two rotating parts: the triangular-shaped rotor and the output shaft with its integral eccentric. The rotor revolves directly on the eccentric. The rotor has an internal timing gear, which meshes with the fixed timing gear on one side housing to maintain the correct phase relationship between the rotor and eccentric shaft rotations. Thus the rotor rotates and orbits around the shaft axis. Breathing is through ports in the center housing (and sometimes the side housings). The combustion chamber lies between the center housing and rotor surface and is sealed by seals at each apex of the rotor and around the perimeters of the rotor sides. Figure 1.22 also shows how the Wankel rotary geometry operates with the four-stroke cycle. The figure shows the induction, compression, power, and exhaust processes of the four-stroke cycle for the chamber defined by rotor surface AB. The remaining two chambers defined by the other rotor surfaces undergo exactly the same sequence. As the rotor makes one complete rotation, during which the eccentric shaft rotates through three revolutions, each chamber produces one power "stroke." Three power pulses occur, therefore, for each rotor revolution; thus for each eccentric (output) shaft revolution, there is one torque pulse. Figure 1.23 shows a cutaway drawing of an intake-port injected two-rotor automobile Wankel engine. The two rotors are out of phase to provide a greater number of torque pulses per shaft revolution. Note the combustion chamber cut out in each rotor face, and the rotor apex and side seals. Two spark plugs per firing chamber are often used to obtain a faster combustion process. Large area side intake and exhaust ports are used to increase airflow, and improve burned gas outflow.

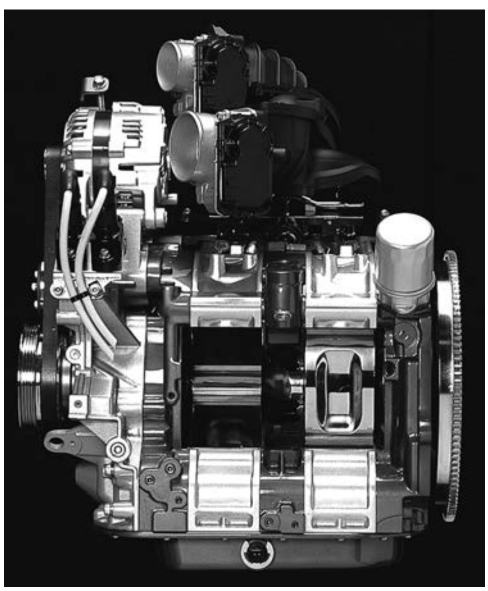


(*a*)



(*b*)

Figure 1.22 (*a*) Major components of the Wankel rotary engine. (*b*) Induction, compression, power, and exhaust processes of the four-stroke cycle for the chamber defined by rotor surface AB. (*From Mobil Technical Bulletin*, Rotary Engines, © *Mobil Oil Corporation, 1971*)



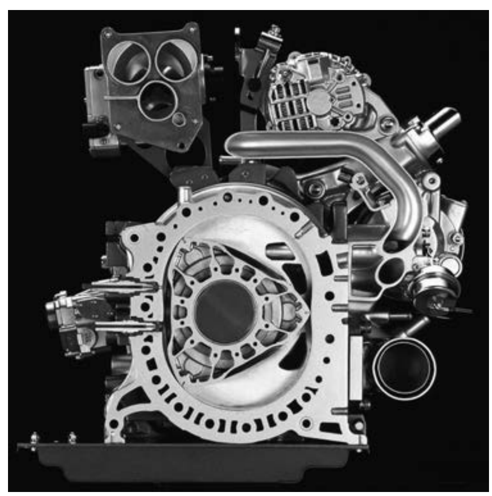


Figure 1.23 Mazda 1.3-liter RENESIS two-rotor Wankel rotary engine. Compression ratio 10:1, 15-mm eccentricity (offset between eccentric shaft axis and rotor centerlines), 105-mm generating radius (distance between rotor centerline and apex), trochaic (rotor) chamber width is 80 mm, giving 654 cm^3 displaced volume for each rotor chamber (1308 cm^3 total). Produces 184 kW at 8200 rev/min and 216 N·m of torque at 5500 rev/min.[24] (*Courtesy Mazda Motor Co.*)

1.8 COMPRESSION-IGNITION ENGINE OPERATION

In compression-ignition or diesel engines, air alone is drawn into the cylinder during intake. The fuel (in most applications a light fuel oil, though heated residual fuel is used in large marine and power-generating diesels) is injected directly into each cylinder just before the combustion process is required to start. Load control is achieved by varying the amount of fuel injected each cycle; the airflow at a given engine speed is not directly controlled, and in naturally-aspirated engines is essentially unchanged. There are a great variety of CI engine designs in use in a wide range of applications—automobile, truck, locomotive, marine, power generation. Both naturally-aspirated engines where atmospheric air is inducted, and boosted engines where the inlet air is compressed by a turbocharger—an exhaust-driven turbine-compressor combination—are common. Turbocharging increases engine output by increasing the air mass flow per unit displaced volume, thereby allowing an increase in fuel flow. These devices are used to reduce engine size and weight for a given power output. In large engine sizes, the two-stroke cycle is competitive with the four-stroke cycle because in these large low-speed engines, the cylinder can be scavenged more effectively and, with the diesel's direct fuel injection, only air is lost in the scavenging process.

The operation of a typical four-stroke naturally-aspirated CI engine is illustrated in Fig. 1.24. The compression ratio of diesels is much higher than typical SI engine values, and is in the range 14 to 22, depending on the type of diesel engine and whether the engine is naturally aspirated or turbocharged. The valve timings used are similar to those of SI engines. In a naturally-aspirated engine, air at close-to-atmospheric pressure is inducted during the intake stroke and then compressed to a pressure of about 5 MPa (50 atm) and temperature of about 900 K (600°C) during the compression stroke. At about 20 crank angle degrees before TC, fuel injection into the engine cylinder commences; a typical rate of injection profile is shown in Fig. 1.24*b*. Usually there are four to eight or more liquid fuel jets; each jet exiting the injector nozzle atomizes into drops and entrains air to form a set of sprays that penetrate into the bowl-in-piston combustion chamber, as shown in Fig. 1.25. In each spray, the liquid fuel drops evaporate, and the fuel vapor then mixes with the entrained air. The fuel-vapor air mixture temperature and pressure are above the ignition point where fuel oxidation chemistry can occur. Thus, after a short *delay period*, spontaneous ignition (autoignition) of parts of the nonuniform fuel-air mixture within these sprays initiates the combustion process, and the cylinder pressure (solid line in Fig. 1.24*c*) rises above the nonfiring engine level as fuel chemical energy is released. A diffusion flame then spreads rapidly to surround each fuel spray, with partly reacted fuel-air mixture in the spray on the inside of the flame, and the additional air in the cylinder on the outside. As the expansion process proceeds, mixing between fuel vapor, air, and burning gases continues, accompanied by further combustion (Fig. 1.24*d*). At full load, the mass of fuel injected is about 5% of the mass of air in the cylinder. At higher fueling levels, increasing amounts of black smoke in the exhaust limit the quantity of fuel that can be burned efficiently. The exhaust process is similar to that of the four-stroke SI engine: a rapid outflow or blowdown of burned gases as soon as the exhaust valves start opening, followed by displacement of the remaining burned gases from the cylinder during the exhaust stroke. At the end of the exhaust stroke, the cycle starts again.

In this diesel combustion process, the fuel is injected directly into the engine cylinder at a pressure of between several hundred and more than 2000 bar.[25] The diesel fuel-injection system consists of a low-pressure pump, and a high-pressure injection pump, delivery pipes, and fuel injector nozzles. Several different types of injection pumps and nozzles are used.

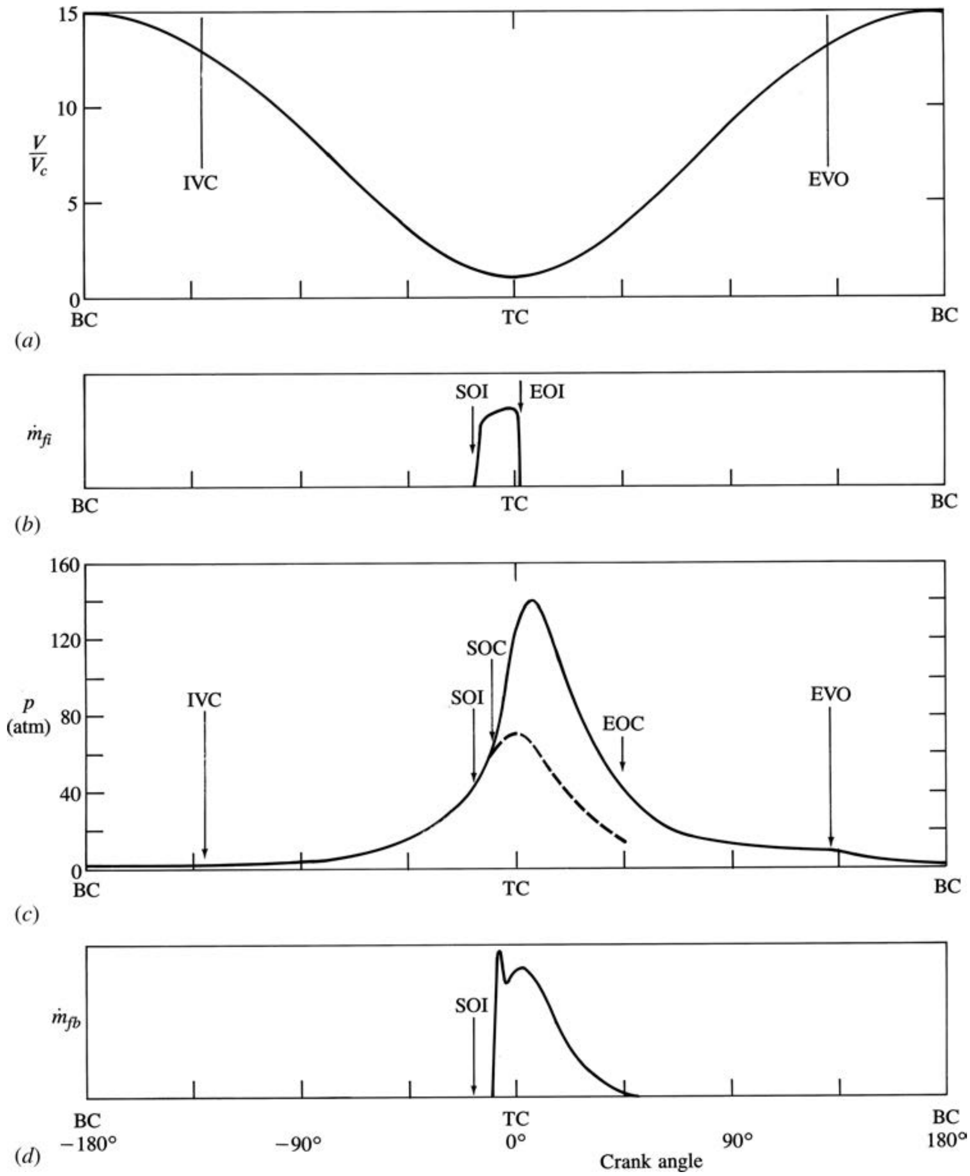


Figure 1.24 Sequence of events during compression, combustion, and expansion processes of a naturally-aspirated compression-ignition engine operating cycle. (*a*) Cylinder volume/clearance volume V/V_c, (*b*) rate of fuel injection $\dot{m}_{fi}$, (*c*) cylinder pressure p (solid line, firing cycle; dashed line, motored cycle), and (*d*) rate of fuel burning (or fuel chemical energy release rate) $\dot{m}_{fb}$ are plotted against crank angle.

In one common system shown in Fig. 1.26, an in-line pump containing a set of cam-driven plungers (one for each cylinder) operate in closely fitting barrels. Early in the stroke of the plunger, the inlet port is closed and the fuel trapped above the plunger is forced through a check valve into the injection line. The injection nozzle (Fig. 1.27) has one or more holes through which the fuel sprays into the cylinder. A spring-loaded valve closes these holes until the pressure in the injection line, acting on part of the valve surface, overcomes the

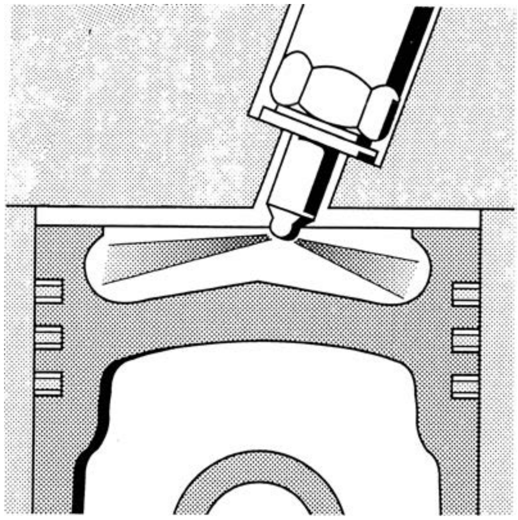


Figure 1.25 Schematic of diesel engine fuel sprays, formed from liquid fuel jets injected at high pressure through individual injection nozzle holes, penetrating the diesel bowl-in-piston combustion chamber.

spring force and opens the valve. Injection starts shortly after the line pressure begins to rise. Thus, the phasing of the pump camshaft relative to the engine crankshaft controls the start of injection. Injection is stopped when the inlet port of the pump is uncovered by a helical groove in the pump plunger, because the high pressure above the plunger is then released (Fig. 1.27, bottom). The amount of fuel injected (which controls the load) is determined by the injection pump cam design and the position of the helical groove. Thus for a given cam design, rotating the plunger and its helical groove varies the load.

In smaller diesel engines, distributor-type fuel pumps are often used. These have one pump plunger and barrel, which meters and distributes the fuel to all the injection nozzles. The unit contains a high-pressure injection pump, an overspeed governor, and an injection timer. High pressure is generated by the plunger, which is made to describe a combined rotary and stroke movement. This rotary motion distributes the fuel to the individual injection nozzles. Distributor pumps can operate at higher speed and take up less space than in-line pumps. They are normally used on smaller three- to six-cylinder diesel engines. In-line pumps are used in larger, midsize engines.

An alternative fuel-injection approach uses individual single-barrel injection pumps, close mounted to each cylinder with an external drive. These unit injector systems (UIS) combine the pump and injector into a single unit. Figure 1.28 illustrates how this system operates. The unit is driven by the engine camshaft, and the start of injection and injected fuel quantity are controlled by a solenoid valve in the injector. With this type of system, very high injection pressures (some 2000 bar) are achieved along with precise control of the amount and timing of injection.

Common rail (or fuel accumulation) injection systems allow greater freedom to control the fuel-injection process, and thus combustion. The functions of fuel pressure generation and fuel injection are separated by an accumulator or common rail. Figure 1.29 shows the system layout. The high-pressure pump feeds the common rail, which feeds each of the injectors. Control of this system can readily be integrated with other engine parameters

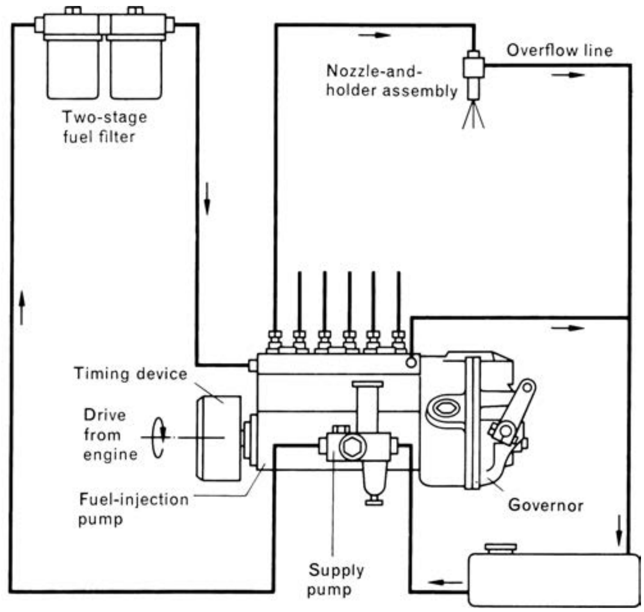


Figure 1.26 Diesel fuel system with in-line fuel-injection pump.[25] (*Courtesy Robert Bosch GmbH and SAE.*)

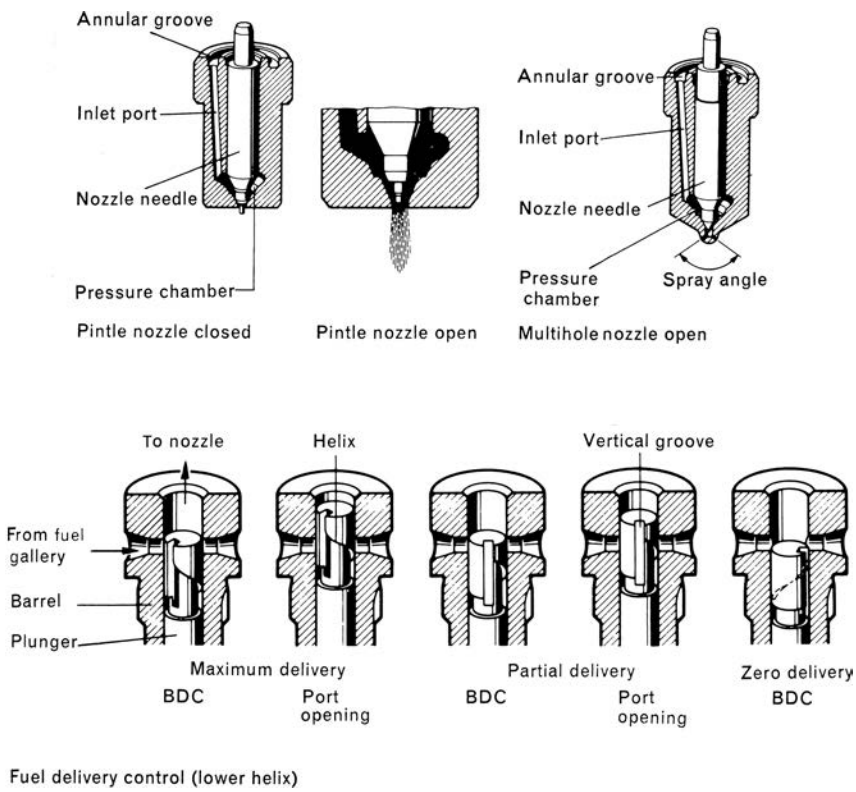


Figure 1.27 Details of fuel-injection nozzles and fuel-delivery control.[26] (*Courtesy Robert Bosch GmbH and SAE.*)

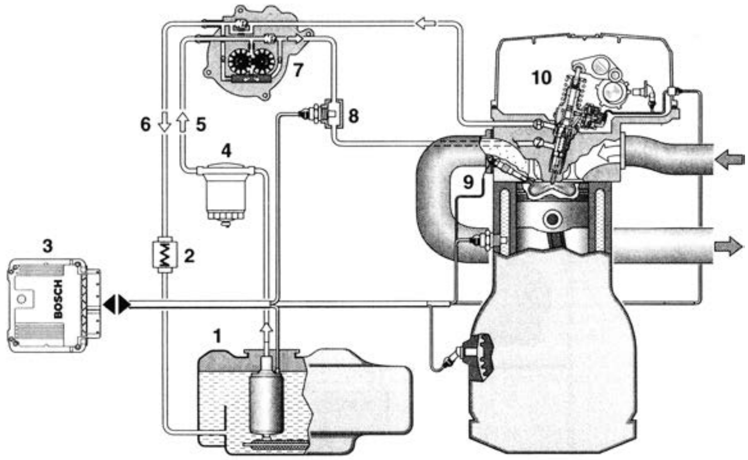


Figure 1.28 Diesel fuel-injection system with unit injector (for passenger cars). (1) Fuel tank with fuel supply pump; (2) Fuel cooler; (3) Electronic control unit; (4) Fuel filter; (5) Fuel feed line; (6) Fuel return line; (7) Tandem pump; (8) Fuel-temperature sensors; (9) Glow plug; (10) Injector.[26] (*Courtesy Robert Bosch GmbH and SAE.*)

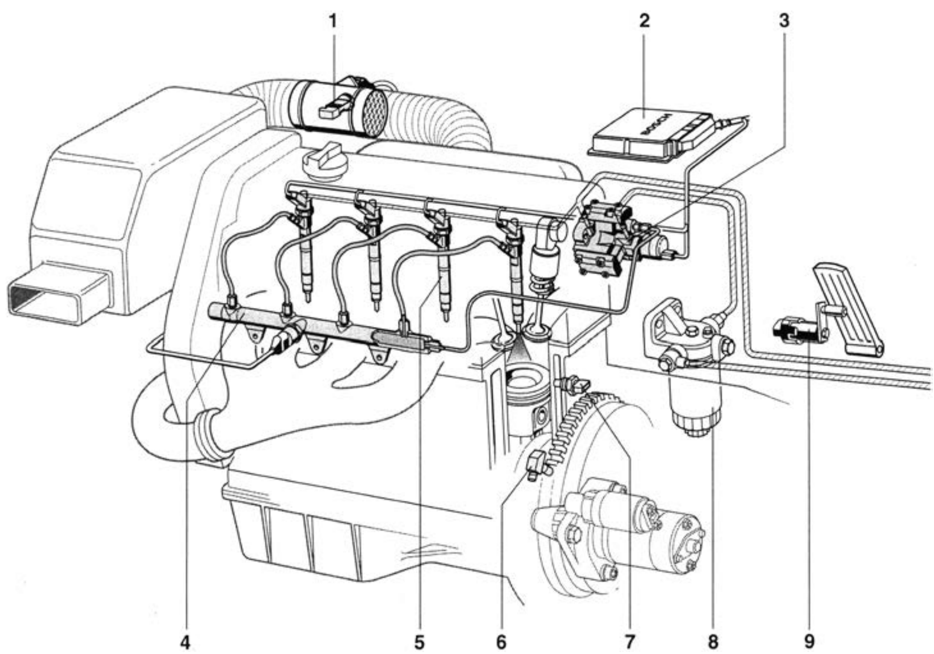


Figure 1.29 Common rail accumulator fuel-injection system on a four-cylinder automobile diesel.[25] (1) Air-mass flow meter; (2) Electronic control unit; (3) High-pressure pump; (4) High-pressure accumulator (rail); (5) Injectors; (6) Crankshaft speed sensor; (7) Coolant temperature sensor; (8) Fuel filter; (9) Accelerator-pedal sensor. (*Courtesy Robert Bosch GmbH and SAE.*)

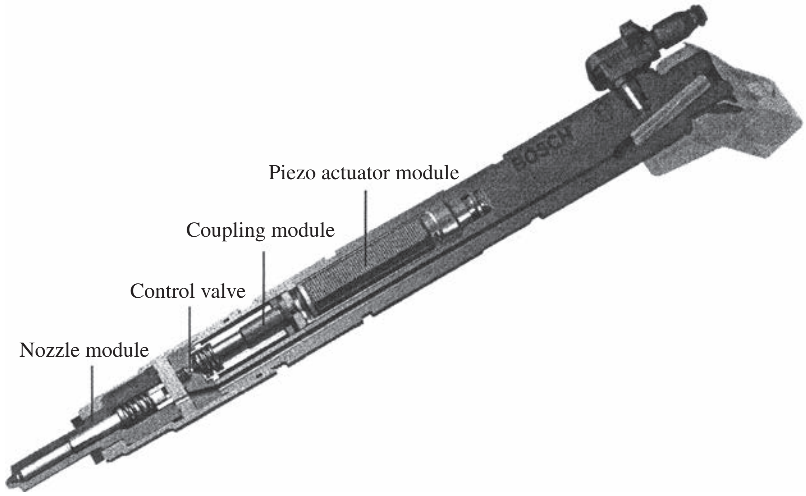


Figure 1.30 Bosch piezoelectric fuel injector that provides more precise fuel injection control on opening and closing, and finer atomization.[27] (*Courtesy Robert Bosch GmbH and SAE.*)

as indicated. Injection pressures of 1400 to 1800 bar can be achieved. By repeated activation of the fast-acting solenoid valve within the injector, multiple injection pulses in each injection cycle can be realized. It is often advantageous to use a short *pilot injection* before the main injection, to initiate combustion with a small amount of fuel, and reduce engine noise. Multiple pulse main injections can help control emissions. Both solenoid-controlled injectors and piezo-actuated injectors are used. Figure 1.30 shows a piezoelectric injector that provides more precise control of injection pulses and improves fuel atomization within the cylinder.

1.9 DIFFERENT TYPES OF DIESEL ENGINES

A large number of diesel engine configurations and designs are in common use. The largest marine and stationary power-generating diesels are two-stroke cycle engines and are discussed in the next section, 1.10. Small- and medium-size diesel engines use the four-stroke cycle. Because air capacity is an important constraint on the amount of fuel that can be burned in the diesel engine, and therefore on power, turbocharging is used extensively. Larger engines are almost always turbocharged. Small low-cost diesels are not usually turbocharged. The details of the engine design also vary significantly over the diesel size range. In particular, different combustion chamber geometries and fuel-injection characteristics are required to deal effectively with a major diesel engine design problem: achieving sufficiently rapid fuel-air mixing rates to complete the fuel-burning process in the short time available. Smaller engines run at higher maximum speeds. Thus, a wide variety of inlet port geometries, cylinder head and piston cavity or bowl shapes, and fuel-injection patterns are used to achieve the airflows and fuel flows needed to accomplish fast enough combustion over the diesel size range.

Figure 1.31 shows a diesel engine typical of the heavy-duty truck application. The design shown is a six-cylinder in-line engine. The drawing indicates that diesel engines are generally substantially more rugged and heavier than SI engines because stress levels are higher due to the significantly higher pressure levels of the diesel cycle. The engine shown

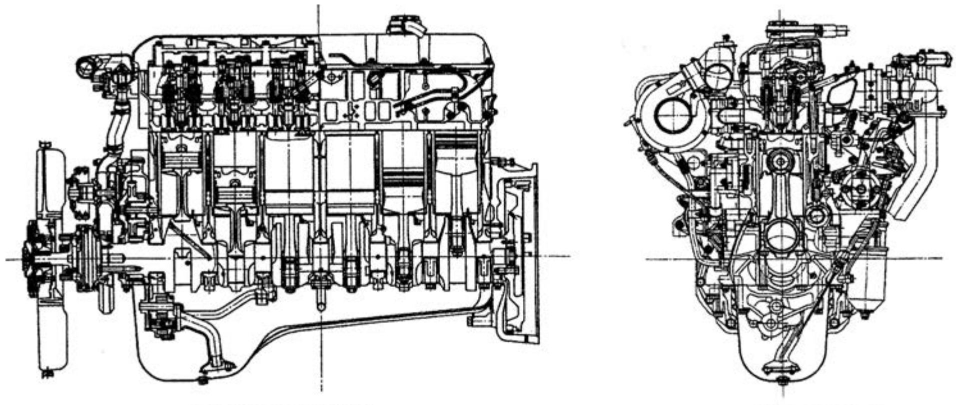


Figure 1.31 Six-cylinder 12.9-liter turbocharged direct-injection truck diesel engine: transverse and longitudinal sections. Bore 135 mm, stroke 150 mm, compression ratio 16:1, governed speed 2100 rev/min, maximum output 294 kW, maximum torque 1667 N·m at 1300 rev/min.[28]

has a displacement of 12.9 liters, a compression ratio of 16.5, and is turbocharged. This type of diesel is called a *direct-injection* diesel since the fuel is injected directly into a combustion chamber above the piston crown. The combustion chamber shown is a “bowl-in-piston” design, which puts most of the clearance volume into a compact cavity in the piston crown. With this type of diesel engine, it is often necessary to use a swirling airflow rotating about the cylinder axis, which is created by suitable design of the inlet port and valve, to achieve fast enough fuel-air mixing and fuel-burning rates. The fuel injector, shown on the cylinder axis in the drawing, has a multihole nozzle. It typically has four to six holes in this application. The fuel jets move out radially from the nozzle holes close to the center of the piston bowl into the (swirling) airflow.

Figure 1.32 shows a 2.2-liter four-cylinder high-speed direct-injection (HSDI) diesel typical of those used in automobiles. It has four valves per cylinder, with the fuel injector and bowl-in-piston combustion chamber centered on each cylinder axis. These types of engines are highly boosted to give high torque at low- to mid-engine speeds per unit displaced cylinder volume. The compressor exit air, cooled in a heat exchanger, enters the cylinder via intake ports that generate swirl about the cylinder axis. During the compression stroke, the flow of this swirling air into the reentrant bowl-in-piston significantly enhances the swirl, thereby achieving high rates of mixing of air with the injected fuel sprays close to top center. Maximum rated speeds (4500 to 5000 rev/min) are lower than maximum speeds of gasoline SI engines due to fuel-air mixing rate limitations. The compression ratio of these HSDI engines is 18 to 20:1, somewhat above the value that gives maximum engine efficiency. This is done to obtain higher air compression temperatures to enable cold engine starting at low cranking speeds. An electrically heated ceramic *glow plug* is usually inserted through the cylinder head in these size engines to assist the cold starting process.

The smallest diesels operate at higher engine speeds than do larger engines: hence the time available for burning the fuel is less and the fuel-injection and combustion systems must achieve faster fuel-air mixing rates. Figure 1.32 shows how this rapid combustion is realized in HSDI diesels. Historically, in the smallest sizes diesel engines, this can be accomplished by using an *indirect-injection* or prechamber type of diesel. Fuel is injected into an auxiliary combustion chamber that is separated from the main combustion chamber above

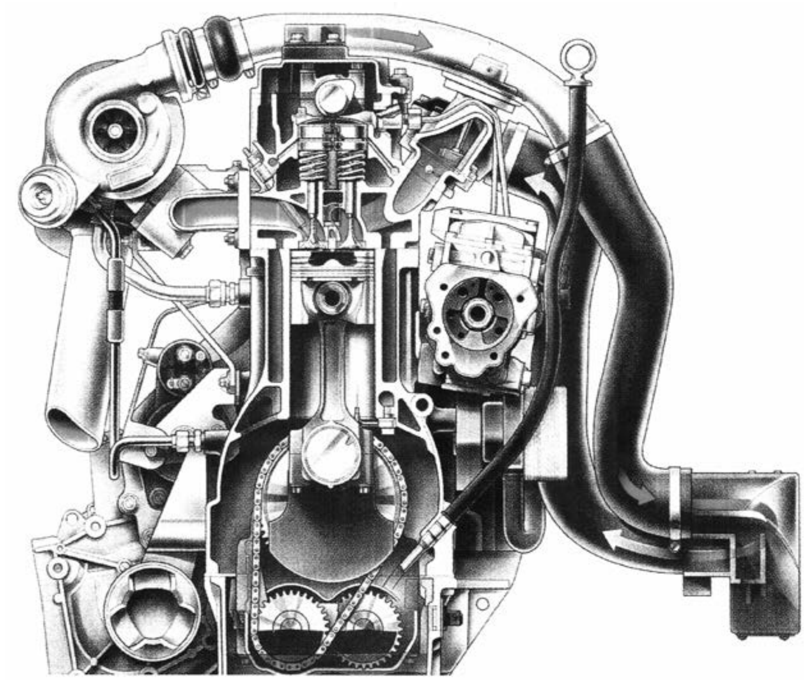


Figure 1.32 Small 2.2-liter four-cylinder high-speed direct-injection automobile diesel engine: maximum power 93 kW; maximum torque 285 N·m at 1750 rev/min; maximum boost 0.9 bar (gauge); compression ratio 18.5. (*Courtesy SAE and Saab.*)

the piston by a passageway or nozzle. During the latter stages of the compression process, air is forced through this nozzle from the cylinder into the prechamber at high velocity. Fuel is injected into the highly turbulent and often rapidly swirling flow in this prechamber, and very high fuel-air mixing rates are achieved. Combustion starts in the prechamber, and the resulting pressure rise in the prechamber forces burning gases, fuel, and air into the main chamber. Since this outflow is also extremely vigorous, rapid mixing then occurs in the main chamber as the burning jet exiting the prechamber mixes with the remaining air and combustion is completed. A glow plug is also shown in the auxiliary chamber; this plug is electrically heated during cold engine start-up to raise the temperature of the air charge and the fuel sufficiently to achieve autoignition. The compression ratio of this engine is high—around 20. Indirect-injection diesel engines require higher compression ratios than direct-injection engines to start adequately when cold.

1.10 TWO-STROKE CYCLE ENGINE OPERATION

The two-stroke engine is used at the small-size and very large-size ends of the engine market. In small sizes, the two-stroke cycle SI engine is cheap, compact and light, simple, and robust. This is the basis of its market appeal in mopeds, scooters, motorcycles, and snowmobiles, in portable devices such as chain-saws and bush cutters, in agricultural and construction devices such as lawn mowers, disc saws, and snow blowers, in the outboard marine engine arena, and in light and in remotely piloted aircraft. The very large diesel engines used in marine and power-generation applications are also two-stroke cycle engines.

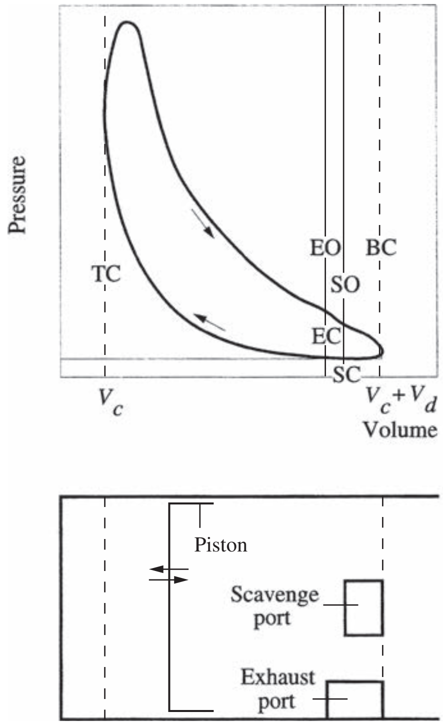


Figure 1.33 Cylinder-pressure versus cylinder-volume trace for a two-stroke cycle engine cylinder. Exhaust and transfer or scavenge ports are uncovered by the piston as shown.

These large internal combustion engines are the most efficient and cost effective prime movers currently available. The two-stroke diesel has also been used in the locomotive and in parts of the truck market.[29,30] The passenger-car and truck engine markets are now, however, dominated by four-stroke cycle engines.

Key operating features of the two-stroke cycle are its power stroke every crankshaft revolution and its scavenging of the burned gases from the engine cylinder with fresh charge. Doubling the number of power strokes per unit time relative to the four-stroke cycle increases the power output per unit displaced volume. It does not, however, increase by a factor of two. The outputs of two-stroke engines range from only 20% to 60% above those of equivalent-size four-stroke units. This lower increase in practice is a result of the poorer than ideal charging efficiency, that is, incomplete filling of the cylinder volume with fresh air due to incomplete scavenging of the residual burned gases. Doubling the number of power strokes per unit time also halves the intervals between combustion-generated pressure impulses. This results in a smoother crankshaft torque versus time profile.

The two-stroke cycle's process of scavenging the burned gases from the engine cylinder with fresh charge—its gas exchange process—has several consequences. First, charging losses are inevitable. Under higher load conditions, in a typical small two-stroke engine, some 20% or more of the fresh charge that enters the cylinder is lost due to short-circuiting to the exhaust. When the fuel is mixed with air prior to cylinder entry, this process results in very high hydrocarbon emissions and poor fuel consumption compared with the four-stroke cycle engine. However, as both exhaust and charging occur around BC, the exhaust and intake ports can be situated near the bottom end of the cylinder and can be covered and uncovered by a long-skirt piston (see Fig. 1.33). This simple geometric two-stroke cycle configuration obviates the need for valves and their actuating gear. It also substantially simplifies the engine structure and the production process, and significantly reduces engine cost.

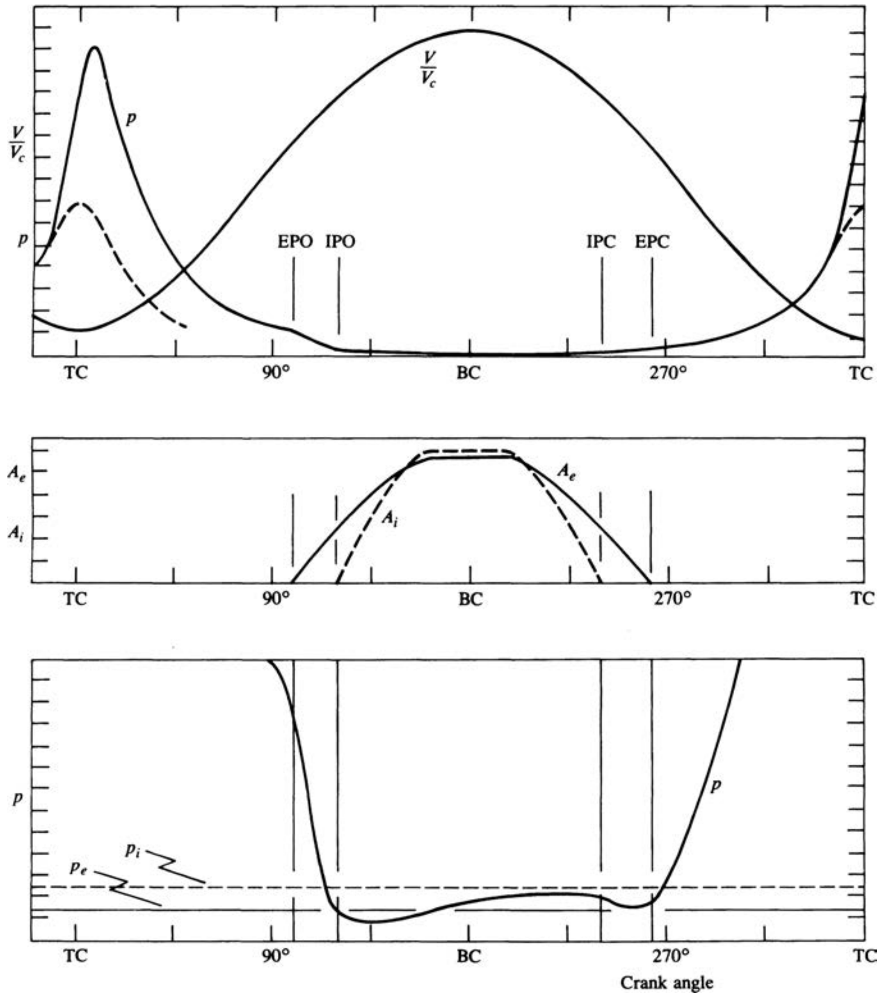


Figure 1.34 Sequence of events during expansion, gas exchange, and compression processes in a loop-scavenged two-stroke cycle compression-ignition engine. Cylinder volume/clearance volume V/V_c, cylinder pressure p, exhaust port open area A_e, and intake port open area A_i are plotted against crank angle.

In the two-stroke engine cycle, the compression, combustion, and expansion processes are similar to the equivalent four-stroke cycle processes; it is the intake and exhaust processes that are different (Fig. 1.33). The sequence of events in a port-scavenged two-stroke engine is illustrated in Fig. 1.34. In such engines, both exhaust and the scavenging (or transfer) ports are at the same end of the cylinder and are uncovered as the piston approaches BC (Fig. 1.34). After the exhaust ports open, the cylinder pressure falls rapidly as burned gases flow out of the cylinder into the exhaust system, in a blowdown process as shown. The scavenging or transfer ports then open, and once the cylinder pressure p falls below the scavenging pressure p_i, fresh charge flows into the cylinder. Burned gases, displaced by this fresh charge, continue to flow out of the exhaust port (along with some of the fresh charge). Once the ports close as the piston starts the compression stroke, compression, fuel-injection and fuel-air mixing in direct-injection engines, combustion, and expansion processes

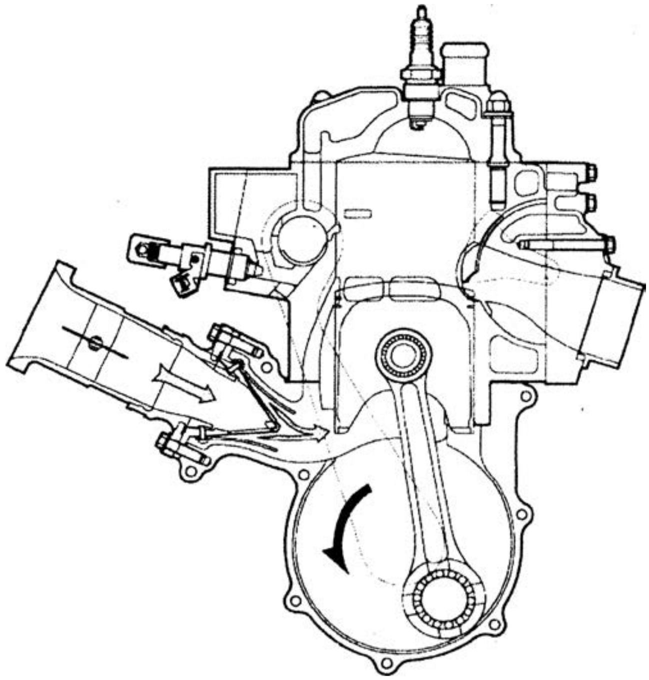


Figure 1.35 Honda experimental two-stroke cycle loop-scavenged engine with pneumatic direct in-cylinder fuel injection, employing an activated radical combustion process—controlled autoignition of the in-cylinder fuel-vapor, air, burned residual mixture just before top center. 402 cm^3 displacement, bore and stroke 80 mm, trapped compression ratio 6:1. Generates 33 kW at 6900 rev/min.[31]

proceed as in the equivalent four-stroke engine cycle.[29, 30] (Fresh charge is fuel vapor and air in engines where fuel and air are "premixed" before entry into the cylinder: the fresh charge is air if direct fuel injection is used.)

Figure 1.35 shows how this two-stroke cycle is realized in a small gasoline engine.[31] This is an experimental single-cylinder crankcase-scavenged engine, with a pneumatic direct fuel injection system to avoid high hydrocarbon emissions and achieve good fuel consumption. Air flows into the crankcase, through Reed valves, as the piston travels up the cylinder. This air is compressed as the piston travels down. Air flows into the cylinder once the transfer or scavenging ports, which connect the crankcase to the cylinder, are uncovered. Fuel is injected after the transfer ports have been closed off. This engine employs a novel combustion approach—often called *activated radical combustion*. By retaining the appropriate amount of burned residual gas within the cylinder by restricting the flow out of the exhaust ports with a throttle valve, the in-cylinder unburned gas temperature at the end of compression can be raised and controlled. This in-cylinder mixture can then be made to autoignite at the appropriate point in the cycle—at the end of compression, just before TC. This spontaneous autoignition of the complete well-mixed fuel vapor, air, and burned residual in-cylinder charge is an alternative combustion process to the standard spark-ignited or diesel combustion processes. It is often called *homogeneous charge compression ignition* (HCCI) or *controlled autoignition* (CAI). This HCCI combustion concept is also

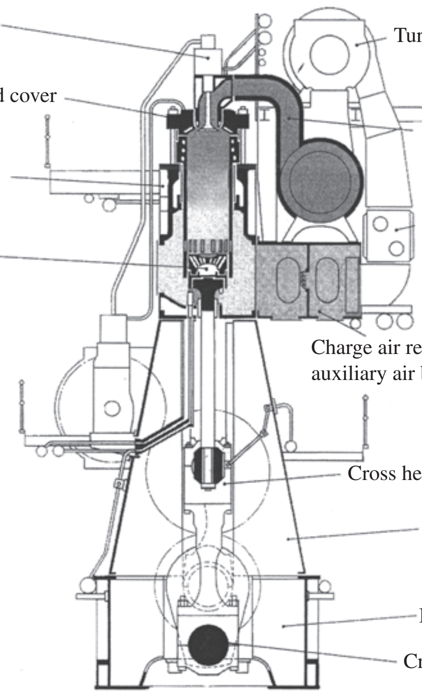


Figure 1.36 Cross-section of an IHI-Sulzer uniflow-scavenged large two-stroke cycle turbocharged diesel engine, developing 1590 kW per cylinder at 127 rev/min. Stroke to bore ratio about 2:5.[30]

being developed for four-stroke cycle engines. At high outputs, when effective scavenging of the burned gases becomes important, this two-stroke cycle engine reverts to the normal spark-ignition engine flame propagation process.

Diesels, in the very large size engines, used for marine propulsion and electrical power generation, also operate using the two-stroke cycle. Figure 1.36 shows such a two-stroke cycle marine engine, available with 4 to 12 cylinders, with a maximum bore of 0.6 to 0.9 m and stroke of 2 to 3 m, which operates at speeds of about 100 rev/min. These engines are normally of the crosshead type shown to reduce piston side forces on the cylinder. The gas exchange process is initiated by opening the exhaust valve in the cylinder head, followed by the piston uncovering the transfer ports at the lower end of the cylinder liner. The expanding exhaust gases leave the cylinder via the exhaust valve and manifold, and pass through the turbocharger turbine. Compressed air enters the cylinder via the transfer ports, continuing the scavenging process; the air is supplied from the turbocharger compressor and intercooler. At part load, electrically driven blowers cut in to compress the scavenge air. Because these large engines operate at low speed, the motion induced by the injected fuel jets is sufficient to mix the fuel with air and burn it in the time available. Thus, a simple open combustion chamber shape can be used, which achieves efficient combustion even with the low-quality heavy fuels used with these types of engines. The pistons are water cooled in these very large engines. The splash or jetted oil piston cooling used in small- and medium-size diesels is not adequate.

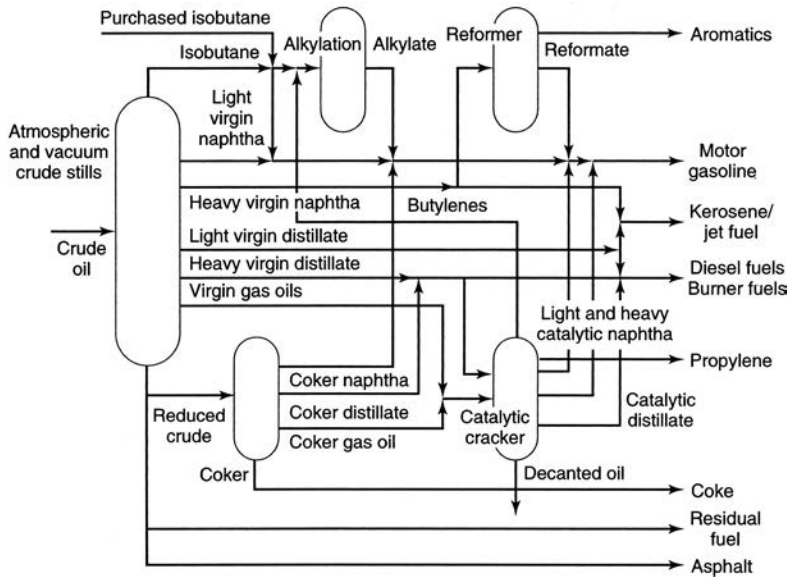


Figure 1.37 Schematic layout (simplified) of crude oil refinery process.[33]

1.11 FUELS

1.11.1 Gasoline and Diesel

Crude oil is the primary source or *feedstock* used today to produce transportation fuels, accounting for more than 95% of transport energy. The gasoline (or petrol) and diesel fuel produced from crude oil are the overwhelmingly dominant *energy carriers* used in SI and CI (or diesel) engines, respectively. Crude oil contains a large number of different hydrocarbons, and these compounds range from gases to liquids to waxes. In a refinery, the crude oil is physically separated by distillation into various fractions. Portions of these fractions are then chemically processed into fuels and other products.[32] Figure 1.37 shows the layout of a typical refinery.[33] The crude oil is first separated into various fractions (each with a higher boiling point range) referred to as *naphtha, distillate, gas oil*, and *residual oil*. The terms *light*, *middle*, or *heavy* break these out further by volatility. The terms *virgin* or *straight run* indicate that no chemical processing has been done to the fraction. As shown, virgin naphtha can be used directly as gasoline. Figure 1.37 also shows the different chemical processes typically used: alkylation, reforming, coking, and catalytic cracking. The refinery products are gasoline, kerosene and jet fuel, diesel, heating oil, burner fuel, residual fuel oil, coke, chemical feedstocks, and asphalt. On average, a refinery will refine 25 to 45% of the input crude oil into gasoline, 25 to 40% into diesel, jet fuel, and heating oil, 5 to 20% into heavy fuel oils, and the remaining 20% into other products.

Gasoline is a blend of hydrocarbons with boiling points ranging from about 25 to 200°C; diesel fuel is a blend of hydrocarbons with boiling points ranging from about 160 to 350°C. Chemical processing is used to convert one fraction into another to upgrade a given fraction so the refinery output meets each fuels' specifications. Alkylation is used to increase the fuel's molecular weight and knock resistance or octane number,[h] by adding alkyl radicals to gaseous hydrocarbon molecules to create branched paraffinic compounds

[h]See Sec. 9.6.3.

(alkanes). Light olefin gases are reacted with isobutene in the presence of a catalyst. This process consumes relatively little energy. Catalytic cracking breaks up heavy oil molecules to produce a range of lighter products including aromatics and olefins for use in gasoline. The products of catalytic cracking go into high octane gasolines. The catalytic cracking reactions occur at high temperatures so considerable energy is consumed in this process. The reformer changes the molecular structure of specific streams such as naphtha to yield higher octane gasoline (e.g., conversion of paraffins into aromatic hydrocarbons, and straight chain hydrocarbons into branched hydrocarbons) in the presence of a catalyst. Considerable hydrogen is produced that is used elsewhere in the refinery. The coker converts the heavy reduced crude fraction to naphtha and distillate fractions. The reduced crude is heated in an oven where the molecules undergo pyrolytic decomposition and recombination. While the average molecular weight of the fraction remains the same, a greater spectrum of components is produced. The heaviest component is coke, a solid largely carbonaceous material.[34]

Gasoline, or petrol, is the dominant SI engine fuel. It is a blend of light distillate hydrocarbons (paraffins, naphthenes, olefins, aromatics). Like all liquid hydrocarbons, it has a very high chemical energy density—energy per unit mass or volume. This chemical energy is stored in the carbon-hydrogen and carbon-carbon bonds and is released as thermal energy when these elements are oxidized by burning the fuel with air. In the United States, fuel requirements for SI engines are defined by the American Society for Testing and Materials in ASTM 4814.[35] The equivalent European Standard is EN 228.[36] The average molecular composition of a typical gasoline is close to C_7H_{13}; it includes hydrocarbon molecules containing between four and ten carbon atoms. The C:H atom ratio in commercial gasolines varies from about 1.6 to 2.2.

In terms of SI engine operation, the most critical gasoline properties are its volatility and its resistance to autoignition (or knock) during the latter part of the combustion process. A gasoline's volatility is characterized by its distillation curve, the volume fraction evaporated at atmospheric pressure as a function of temperature. Typically 10% of gasoline vaporizes below about 50°C (T_{10}), 50% vaporizes below 100°C (T_{50}), and 90% below 170°C (T_{90}). Winter gasolines are more volatile than summer gasolines. The fuel must contain enough highly volatile components to ensure rapid and low-emission cold starts. The lower temperature end of the distillation curve affects the fraction of the fuel injected into the port that vaporizes under these conditions. It also affects evaporative hydrocarbon emissions from the vehicle's fuel system. The upper end of the distillation curve is controlled to reduce hydrocarbon emissions and lubricating oil dilution with fuel.

A fuel's resistance to knock is defined by its octane rating. Two standard tests (see Sec. 9.6.3) define the *research octane number* and *motor octane number* of a fuel. The number displayed on gasoline pumps in the United States, the antiknock index, is the average of these. A typical regular (standard) gasoline would have a research octane number of 92, a motor octane number of 82, with an antiknock index of 87. Historically, this antiknock index correlated the knock resistance behavior of fuels in engines in vehicles on the road. Modern engine antiknock requirements are better correlated by the research octane number, which is the fuel's octane number used in Europe. At least two quality gasolines are usually marketed: regular and one or more higher octane grades. Premium gasolines have higher antiknock ratings (up to 98 RON) and often higher levels of additives that are used to control deposit build-up tendencies in various parts of the engine and fuel system. Lead alkyl additives were once widely used to improve the antiknock resistance of gasolines. Since lead is highly toxic, and the catalytic converters used in engines' exhaust systems to reduce emissions (and the oxygen exhaust sensors used for modern engine control) are poisoned by lead, unleaded gasolines are now required in the developed world. While leaded gasoline is still available in some countries, their number is steadily decreasing.

Environmental authorities are imposing increasingly stringent regulations for fuels (gasoline and diesel) to ensure low hydrocarbon evaporative emissions and exhaust pollutant emissions. Thus *reformulated fuels* are now widely used. The main characteristics of reformulated gasolines are reduced vapor pressure (i.e., reduced low-end volatility), lower concentrations of aromatic compounds and benzene, and a lower final boiling point (T_{90}—temperature at which 90% of fuel is evaporated). In the United States, additives that inhibit deposit formation in the engine intake are also required. These reformulated gasolines usually contain oxygenated organic compounds [historically, methyl tertiary-butyl ether (MBTE), now–ethyl alcohol]. In some parts of the United States, these oxygenates are required in winter gasolines (at least 2.7% oxygen) to reduce engine carbon monoxide emissions. Year round, to encourage the use of biofuels, gasolines with up to 5, 10, or 15% ethanol can be marketed (amount depending on country or region). Both MBTE and ethanol have higher octane numbers than the base gasoline so they improve the fuels knock resistance. However, since MTBE can contaminate drinking water supplies, its use in gasolines has been discontinued. The sulfur levels in these clean unleaded gasolines are being reduced from historical levels [some 300 parts per million (ppm) by mass] to levels approaching 10 ppm. Sulfur is a catalyst poison that degrades the effectiveness of exhaust emission-control catalyst systems.

Diesel fuels contain many individual hydrocarbon compounds with boiling points ranging from about 180 to 370°C. As shown in Fig. 1.37, they are a primary product of the crude oil distillation process. To meet the growing demand for diesel fuel, refineries are adding increasing amounts of conversion products through cracking and coking of heavy oil. Diesel fuel is a mixture of paraffins, napthas, olefins, and aromatics; these have higher molecular weights (and different proportions) than these types of hydrocarbons in gasolines. The average molecular composition is $(CH_{1.8})_n$, and average molecular size is $C_{13}H_{24}$. The molecular weight range of the diesel hydrocarbons is about 170 to 200. The density of diesel fuel is important because fuel-injection systems are designed to deliver a specified *volume* of fuel whereas the combustion characteristics depend on the fuel/air *mass* ratio. The average diesel fuel density in Europe and the United States is about 0.84 kg/liter; the density, in other major geographic regions, varies between 0.81 and 0.86 kg/liter. Because the density of diesel fuel is higher than gasoline, the chemical energy content (heating value) *per unit volume* of fuel is about 10% higher. Other properties related to density are volatility and viscosity. Fuel volatility is defined by the distillation curve and the flash point. The distillation curve effectively defines the vaporization of the fuel inside the engine cylinder. The flash point is the minimum temperature to which the fuel must be heated to produce vapor that ignites in the presence of a flame. The viscosity of diesel fuel affects the fuel's performance in the fuel-injection system; it affects both the fuel pump and injector behavior, and the atomization process in the fuel injector nozzle holes. The low temperature flow characteristics of the fuel in the fuel system (especially the fuel filter) are a critical issue in regions with low winter temperatures where waxes can precipitate out.

Diesel fuel must have appropriate autoignition characteristics—that is, spontaneously ignite fast enough within the developing fuel sprays (see Sec. 1.8) to initiate combustion at the desired point in the engine cycle. This characteristic of diesel fuels is defined by the *cetane number*, which compares the autoignition characteristics of a diesel fuel with those of defined reference fuels (see Sec. 10.5.2). Typical cetane numbers are in the 40 to 55 range; the higher the cetane number, the easier (more rapid) is autoignition in the engine at around top center, just after injection starts. Fuel composition is adjusted to provide the appropriate cetane number; it can be enhanced by ignition improving additives—active oxidizers such as alkyl nitrates. Diesel fuel property requirements are defined in ASTM D975.[37]

1.11.2 Alternative Fuels

Alternative fuels to the gasolines and diesel fuels produced from petroleum are in limited use in IC engines, and are being explored for expanded use in the future. Important driving forces are the need to reduce transportation's dependence on petroleum, and to reduce emissions of GHGs—especially CO_2. LPG and natural gas are in limited use as SI engine fuels in specific applications. Ethanol, produced from biomass, to date is used primarily as a gasoline blending agent. Alternative diesel fuels—for example, biodiesel (methyl esters made from rape seed or soybeans); also dimethyl ether (DME), Fischer-Tropsch gas-to-liquids fuels made from natural gas (or from coal)—are being produced and explored. Hydrogen is being examined as a possible longer-term SI engine fuel largely for its potential, once on the vehicles, to be a noncarbon-emitting energy carrier.

The more important properties of these alternative fuels, along with those of gasoline and diesel, are tabulated in App. D. An important property of these fuels relevant to their use in engines is their chemical energy content or heating value per unit volume of fuel-vapor/air mixture, which has just enough air for complete combustion (i.e., just enough for full oxidation of the fuel's carbon and hydrogen to CO_2 and H_2O—the stoichiometric mixture ratio). The basic objective of the SI engine's intake process is to fill each cylinder with such a mixture. In the diesel, this objective is met locally as the fuel is combusted. While the heating value of these different fuels *per unit mass of fuel* varies substantially, the differences in the composition of these fuels (and thus the amount of air each fuel requires for complete combustion) bring their heating values *per unit volume* of stoichiometric (chemically correct) mixture to surprisingly similar values. Thus SI engine outputs (i.e., maximum power per unit displaced volume with other parameters held fixed) over a wide range of fuels are closely comparable.

Note also that the combustion characteristics of these fuels when mixed with air in engines are similar (with some important differences in details—especially their anti-knock ratings). The exception is hydrogen which has a much higher flame speed than the hydrocarbon fuels.

Liquid petroleum gas, which consists of C_3 and C_4 hydrocarbons, is produced either by its removal from natural gas during the gas extraction process, or from the refining of crude oil. The fraction of LPG used in transportation varies widely (from a percent or so in the United States to significantly higher levels in some other countries), as does the composition of LPG. The dominant component is propane (C_3H_8): this can range from 30 to over 90%. Propylene, butanes, and ethane make up the remainder. In vehicles, LPG is stored as a compressed liquid at pressures between about 10 and 15 bar, in cylindrical on-board tanks, at ambient temperature. In most LPG fueling systems, the fuel is injected into the intake manifold in similar manner to multipoint injection gasoline systems. A common fuel rail and one injector valve for each cylinder inject the LPG (in liquid or gaseous form) into each intake manifold, continuously or intermittently.[13]

The components of LPG have higher octane ratings or knock resistance than gasoline, so higher engine compression ratios can be used. However, since the fuel is a gas with a lower molecular weight than gasoline, when mixed with air, the volume occupied by the fuel becomes more significant. LPG fuel therefore displaces more air than does gasoline vapor, and results in a modest loss in power. Air pollutant emissions from a LPG-fueled SI engine (prior to any catalyst in the exhaust) are usually lower than from gasoline-fueled engines.

Natural gas is a primary energy source and carrier. It is largely (80 to 98%) methane, CH_4. The remainder is ethane (1 to 8%), propane (up to 2%), with varying amounts of nitrogen (from a few to 10%). Thus it has a higher molar hydrogen:carbon ratio, so its CO_2 emissions on complete combustion are lower than those of gasoline and diesel by about 25%. However, methane itself is a GHG with a global warming potential about 20 times (per molecule) higher than CO_2. Natural gas leakage and unburned methane emissions would need to be carefully controlled if natural gas is used in transportation on a much larger scale.

Methane has a high octane number, or antiknock rating, though the other hydrocarbon species in natural gas reduce this somewhat depending on their relative amounts. Its combustion characteristics are comparable to those of gasoline.

Natural gas is, of course, a gas and thus storing adequate quantities storage on board the vehicle is a challenging problem. Storage as high-pressure compressed gas (35 MPa) is expensive and requires a substantial volume even for limited vehicle range. At these conditions, the volumetric energy density (MJ/m^3) of compressed natural gas is about one-quarter that of gasoline. Also, because it is a gas with low molecular weight (methane's is 16), when natural gas is mixed with air to combustible proportions, it displaces about 10% of the air (at fixed pressure and temperature) thus reducing the torque that can be produced per unit displaced cylinder volume. Similarly to LPG, natural gas is delivered to the engine intake manifold or intake port via a pressure regulator low pressure common rail, and injectors that inject natural gas intermittently into the intake. Because the fuel is a gas, mixture formation is easier to control since there are no liquid fuel drops and films to vaporize.

Alcohols, such as **ethanol** C_2H_6O and **methanol** CH_4O, can be used as SI engine fuels. Ethanol can be produced from biomass; methanol can be produced form natural gas. These alcohols contain oxygen, so their chemical energy content or heating value per unit mass is less than that of hydrocarbons (0.45 for methanol and 0.61 for ethanol). Heating values per unit mass of stoichiometric mixture are 5% less than gasoline.

These alcohols can be used as stand-alone fuels, or blended with gasoline at high or low fractions. These are designated E100 or M100 for ethanol or methanol alone. Blends with about 15% gasoline (e.g., E85 or M85) are used to broaden the volatility or vaporization characteristics of the single compound alcohol. Ethanol is the more practical of these two fuels. A disadvantage of methanol is its toxicity. Ethanol is used extensively as a transportation fuel in Brazil, where it is made from sugar cane. It is also made from corn (and can be more efficiently made from cellulosic biomass sources such as switchgrass and fast growing bushes and trees). Gasolines blended with up to 10% ethanol are used in the United States and elsewhere to both meet government requirements for oxygen content in "clean" lower-emission reformulated gasolines, and as an outlet for ethanol fuel.

Both ethanol and methanol have significantly higher antiknock or octane ratings than gasoline; the value depends on whether the fuels are tested or used as stand-alone fuels or in gasoline blends (see Table D4, App. D).

Hydrogen can be produced from natural gas, coal, biomass, and by electrolyzing water. Hydrogen for industrial use is currently largely made from steam reforming of natural gas. Storage of hydrogen on board vehicles is a major challenge. As compressed gas, at 70 MPa (10,000 psi), about the maximum practical storage pressure, hydrogen has about one-third the energy density per unit volume of natural gas. Liquid hydrogen, at 20 K, has about one-fourth the energy density per unit volume of a hydrocarbon fuel. When mixed with air to stoichiometric proportions, the hydrogen in the fuel-air mixture occupies about 30% of the mixture volume, compared with less than 2% for gasoline vapor-air mixtures, reducing the power per unit displaced cylinder volume at constant conditions.

The combustion characteristics of hydrogen are substantially different to those of hydrocarbons, and thus gasoline. The laminar flame speed (a fundamental combustion characteristics of fuel-air mixtures) in hydrogen-air mixtures is several times higher than that of equivalent hydrocarbon-air mixtures due to the much faster diffusion of hydrogen species within the flame. Its flammability limits are broad (4 to 75% by volume at standard atmospheric conditions) and its self-ignition temperature is lower than that of gasoline. However, it has a higher knock resistance and octane number than do gasolines.

Several other fuel's pathways are potential alternatives to petroleum-based diesel fuel. Fuels made via the Fischer-Tropsch process from natural gas are being produced from natural gas reserves that are difficult to transport to the major natural gas markets. Fischer-Tropsch F-T fuels can also be made from coal. The F-T process catalytically combines CO and hydrogen, usually produced by reforming natural gas, to form CH_2 radicals that are combined to form larger hydrocarbon chains. These fuels are completely paraffinic, have very low levels of sulfur, and excellent autoignition characteristics (a high cetane number). They would thus be excellent diesel engine fuels.

Other potential diesel fuels are DME C_2H_6O, and biodiesel. DME is a gas at ambient pressure and temperature; it can be liquefied at about 5 bar pressure. It burns easily in a diesel engine, and its oxygen component helps inhibit the formation of soot and diesel smoke. Vegetable oils, such as rapeseed methyl ester and soybean methyl ester, can be processed to produce biodiesel fuels. Their potential attractiveness is that they are produced from biomass and thus are a possible low GHG emitting source of diesel fuel.

PROBLEMS

1.1. Describe the major functions of the following reciprocating engine components: piston, connecting rod, crankshaft, cams and camshaft, valves, intake, and exhaust manifolds.

1.2. Indicate on an appropriate sketch the different forces that act on the piston, and the direction of these forces, during the engine's expansion stroke with the piston, connecting rod, and crank in the positions shown in Fig. 1.1. Explain how the reciprocating engine geometry generates torque. Illustrate on a sketch.

1.3. List six important differences between the design and operating characteristics of spark-ignition and compression-ignition (diesel) engines. Start with the more fundamental of these.

1.4. Indicate the approximate crank angle at which the following events in the four-stroke and two-stroke internal combustion engine cycles occur on a line representing the full cycle (720° for the four-stroke cycle; 360° for the two-stroke cycle): bottom- and top-center crank positions, inlet and exhaust valve or port opening and closing, start of combustion process, end of combustion process, maximum cylinder pressure.

1.5. Explain briefly the following differences between a standard automobile spark-ignition engine and a truck diesel engine:
(a) Where the fuel is injected and why
(b) How the load is varied at fixed speed
(c) How the combustion process starts, develops, and ends
(d) How the fuels are different and why
(e) How the in-cylinder pressure varies as a function of crank angle (draw qualitatively the pressure traces for both engines in the same graph showing their relative magnitudes and when approximately the combustion starts and ends in each case).

1.6. The two-stroke cycle has twice as many power strokes per crank revolution as the four-stroke cycle. However, two-stroke cycle engine power outputs per unit displaced volume are less than twice the power output of an equivalent four-stroke cycle engine at the same engine speed. Suggest reasons why this potential advantage of the two-stroke cycle is offset in practice.

1.7. Suggest reasons why multicylinder engines prove more attractive than single-cylinder engines once the total engine displaced volume exceeds a few hundred cubic centimeters.

1.8. The gasoline spark-ignition engine dominates the light-duty vehicle market in the United States. Briefly discuss the following:
(a) For light-duty vehicles, what other competing "prime mover" technologies might be attractive? Think both in terms of fuel used and the means by which the fuel's energy is converted to useful work.

(b) In what ways is each of these fuels or technologies better (or worse) than a gasoline spark-ignition engine? Think both in terms of a single vehicle's performance and the "big picture" resulting from having many such vehicles. You do not need to be quantitative.

(c) Now imagine that you are ready to buy your first car. Which of the factors you listed above are most important to you? Which "engine" and energy source would you choose?

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